

Reduction of the Compression Ratio on a HSDI Diesel Engine : Combustion Design Evolution for Compliance the Future Emission Standards

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ABSTRACT

Environment protection issues regarding CO₂ emissions as well as customers requirements for fun-to-drive and fuel economy explain the strong increase of Diesel engine on European market share in all passenger car segments. To comply future purposes of emission regulations, particularly dramatic decrease in NO_x emissions, technology need to keep upgrading ; the reduction of the volumetric compression ratio (VCR) is one of the most promising research ways to allow a simultaneous increase in power at full load and NO_x / PM trade-off improvement at part load.

This study describes the combustion effects of the reduction of compression ratio and quantifies improvements obtained at full load and part load running conditions on a HSDI Common Rail engine out performance (power, fuel consumption, emissions and noise). Potential and limitations of a reduced compression ratio from 18:1 to 14:1 are underlined. Moreover, this paper aims to define initially the head guidelines for future combustion system designs in focusing particularly on the specific evolution required by the reduction of the compression ratio, including width to depth ratio of piston bowl, number of nozzle holes and swirl.

Combustion tests show that the reduction of RVC from 18 to 14 allows an increase of 12% of power at 4000 rpm as well as near zero NO_x and PM emissions at 1640 rpm – BMEP 3,7 bar with nevertheless a significant increase of CO and HC emissions. Moreover, whatever the compression ratio, full load performance has been achieved with larger bowl with lower hole number for the nozzle injector and finally a lower swirl level. As opposed to these results, at low load the reduction of emissions promoted by a thin pulverization of fuel and the best homogenization of fuel/air mixing has required an antagonist design.

INTRODUCTION

Passenger cars powered by diesel engine have known a great success over the last decade as they offer a low fuel consumption with high specific power and torque. Nevertheless, research efforts have to be continuously lead to comply with more and more stringent emission standards, particularly regarding NO_x and PM release.

Many upgraded solutions are investigated to face this key issue of NO_x/PM trade off as the reduction of the volumetric compression ratio, defined hereafter VCR. Actually, the use of a reduced VCR is quite efficient to decrease both NO_x and PM as the natural decrease in the in-cylinder temperature leads to less requirement of exhaust gas recirculation for the same NO_x target. In addition, at full load, higher specific power and torque may be obtained : for the same maximum pressure peak, the injection timing may be advanced and a higher boost pressure is allowed.

The design of the combustion chamber, the number of the nozzle holes and the cylinder head swirl number have a great importance on the engine performance at full and part load operation, as they govern the air fuel mixing rate.

The increase in number of nozzle holes while keeping a constant hydraulic state flow of injector involves the decrease in the orifice diameter of the nozzle. Smaller orifices allow to increase the mixing rate as many physical processes are involved :

- first, a reduced orifice diameter leads to a reduction of the fuel spray penetration [1,2,3] following a factor proportional to the square root of diameter, as reported in the equation (1). As the total entrained mass of air into transient jet is linked to the fuel penetration to the 3rd power (see. equation (2)) [4], the decrease in orifice diameter results in decreasing entrained air mass to a power of 3/2 of diameter as expressed in the equation (3). Thus, the reduction of orifice diameter from 0,148 to 0,144 and finally to 0,137 (orifice diameters of considered nozzles in this study) leads to a loss in air entrainment respectively

equal to 4% and 10%. However, the conservation of the hydraulic flow, requires to increase the number of holes from 6 to 7 and finally 8 holes. Hence, the total entrained mass of air is improved of 12% and 20%

$$P(t) \sim K_1 \rho_a^{-1/4} (P_{inj} - P_a)^{1/4} d^{1/2} t^{1/2} \quad (1)$$

$$M_{air}(t) = K_2 \rho_a P^3(t) \quad (2)$$

$$M_{air}(t) = K_3 \rho_a^{1/4} (P_{inj} - P_a)^{3/4} d^{3/2} t^{3/2} \quad (3)$$

- second, studies [5] have shown that the amount of entrained air into fuel spray upstream the flame lift-off is higher when reducing the orifice diameter (leaner fuel distribution and higher local mixing rate).
- lastly, with the decrease in orifice diameter, the reduction of the fuel spray penetration allows to limit the risk of fuel liquid and vapor phase impingement on the walls, especially in case of operating conditions with a long ignition delay. On the other hand, for higher load operation and for specific engine settings, the reduction of the fuel spray penetration can lead to an increase in soot and CO emissions as the air available on the periphery of bowl is not consumed.

Swirl motion into combustion chamber usually leads to a decrease in fuel spray penetration while increasing the radial fuel vapor spreading. The exchange surface between air and fuel is therefore improved and a higher amount of air is mixed with fuel [6]. Nevertheless, drawbacks may occur with an excessive increase in swirl, such as higher heat losses that directly affect the fuel conversion efficiency, and higher risks of fuel vapor overlapping that conduct to rich fuel regions. The increase in swirl motion into the piston bowl is made possible in two ways. First, by increasing the cylinder head swirl itself. Nevertheless, this method involves a serious disadvantage on air filling efficiency, especially at full load. Second, the increase of swirl can be obtained by decreasing the width-to-depth ratio of piston bowl.

In addition to the effect on swirl motion, width-to-depth ratio of piston bowl has a great influence on the combustion process and on the soot formation. Indeed, a decreased width-to-depth ratio increases the probability of fuel jet-wall interaction. Various investigations are under study to point out the influence of a jet-wall interaction on the mixing rate and soot formation processes. Under Diesel engine conditions with an injection pressure of 150 MPa, Bruneaux [7] showed that no further additional mixing is obtained with a plane wall- jet interaction compared to a free jet. On the other hand, at 200 MPa, an increased mixing has been observed. Pickett [8] recently studied the effect of a plane wall-jet interaction on soot formation processes as well as simplified confined wall-jet interaction so as to reproduce engine wall-jet interaction and subsequent jet-jet interaction effects. Results showed that a plane wall-

jet interaction leads to lower soot emissions and that the plane wall position relative to flame lift-off length has significant effects on the soot formation. A confined wall-jet interaction causes higher soot emissions as combustion gases are redirected towards incoming fuel jet. This causes a shorter flame lift-off length and makes the air fuel mixture richer in the lift-off area.

As a result of all these aspects, the study described in this paper focuses on the consequences of a reduced VCR on the overall combustion system design. A specific attention has been paid to the width-to-depth ratio of piston bowl, the number of nozzle holes and finally the cylinder head swirl number. Further investigations have also been lead to quantify precisely benefits and drawbacks of the reduction of compression ratio from 18:1 to 14:1.

EXPERIMENTAL SET UP

TEST ENGINE

Tests were performed on a single cylinder engine with a displacement of 0,5 Liter (Bore x Stroke = 85 x 88 mm) and equipped with a 4 valve cylinder head based on the PSA DW10 production engine. The compression ratio was defined between 18 : 1 and 14 : 1 by changing the piston bowl volume while keeping a fixed distance between the upper part of the piston and the cylinder head at top dead center. The engine was equipped with a Bosch CRS2 common rail fuel injection system able to deliver a fuel pressure up to 160 MPa. The hydraulic flow of the injector nozzle was chosen at 340 cm³/ 10 MPa / 30 s.

The intake air flow was controlled via a sonic flow nozzle. Air was dehumidified, and a high air capacity was implemented to damp air pulses. An air heater was also used to control the inlet air temperature and produce experimental conditions similar to those encountered with multicylinder engines. Exhaust gas were collected by a production exhaust manifold. A damping capacity was set up downstream the exhaust line ; exhaust pressure was adjusted by a throttle. The engine was managed by an electric generator able to absorb a maximal power of 50 kW. In addition, the engine test bench was equipped with a system controlling the pressures and temperatures of fuel, lubricant and water feeding the engine.

MEASURING EQUIPMENT

The cylinder was equipped with a water cooled pressure transducer Kistler 6043A60. Data acquisitions were performed by an intern IFP product KlepatV3 in which a precise method to set up the top dead center is implemented. Fuel consumption was given by an AVL 733 s. Exhaust emissions were analysed by a six gases analysis device. Fuel air ratio was measured by an oxygen sensor in the exhaust line and it was also calculated from the gas analysis. The EGR rate was computed from intake and exhaust CO₂ rates measures.

COMBUSTION SYSTEM

To quantify the consequences of a reduced volumetric compression ratio (VCR) on the overall combustion system design, different combustion systems were studied. Accordingly, for two values of VCR from 18:1 to 14:1, the influence of width-to-depth ratio of the piston bowl (W/H), number of nozzle holes (with the same global permeability) and cylinder head swirl number was investigated. The studied combinations are summarized in the Table 1.

W/H	VCR 18:1 Number of nozzle holes			VCR 14:1 Number of nozzle holes		
	6	7	8	6	7	8
2,5				x		
3	x	x		x	x	
3,75	x	x	x	x	x	x
4,5		x	x			

Table 1 – Combination of different studied combustion systems

Piston bowl designs have omega shape with bowl central tip, lip and re-entrant. Figure 1 presents a schematic view of a bowl. The main defining terms currently used in this study to characterize a bowl are reported. Re-entrant is defined as following :

$$\text{Re-entrant} = \frac{W - W_{\text{lip}}}{W_{\text{lip}}}$$

Bowl designs in CR 14:1 were obtained from VCR 18:1 ones taking into account the following rules :

1. width-to-depth ratio conservation (W/H) ;
2. conservation of re-entrant ratio;
3. the homothetic number applied to bowl diameter to reduce the CR is equal whatever the W/H ratio ;
4. tip dimensions are identical ;

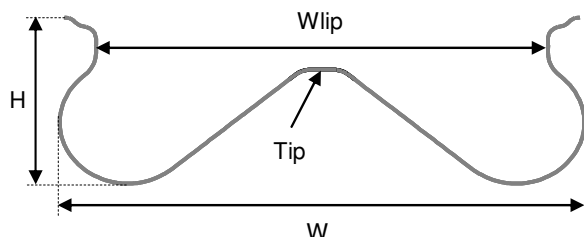


Figure 1 – Schematic view of piston bowl with re-entrant

Table 2 gives details on piston bowl dimensions.

		VCR 18 : 1			VCR 14 : 1		
		W/H = 3	W/H = 3,75	W/H = 4,5	W/H = 2,5	W/H = 3	W/H = 3,75
W	(10 ⁻³ m)	48.0	52.0	55.5	50.5	54.2	58.7
H	(10 ⁻³ m)	16.00	13.86	12.33	20.2	18.1	15.7
Réentrant	(%)	15	15	15	15	15	15

Table 2 – Main dimensions of piston bowl

EXPERIMENTAL CONDITIONS

The general experimental conditions are presented in Table 3 .

		Part load		Full load
		1640	2500	4000
Engine speed	rpm	1640	2500	4000
IMEP	MPa	0,44	0,903	maximum*
Inlet pressure (P2)	MPa	0,105	0,15	0,23 to 0,29
Exhaust pressure (P3)	MPa	0,11	0,16	P2 + 0,08
Inlet temperature	C	60	60	50
Rail Pressure	MPa	variable	variable	160

* with constraints in smoke and exhaust temperature

Table 3 – Experimental conditions

For the full load operating point, tests consisted in increasing the fuel air equivalence ratio (by progressively increasing the amount of injected fuel quantity) at a constant engine speed. SOI was adjusted to keep a constant maximal in cylinder pressure equal to 16 MPa.

For part load operating points, EGR rate variations were conducted while keeping at constant values the other settings (injection timing, rail pressure, EGR temperature ...). Further details on parameter settings are given in the following subsections as they differ with the VCR.

RESULTS AND DISCUSSION

RESULTS AT 1640 RPM - 0,44 MPA IMEP WITH VCR 18:1

Engine settings : $\text{SOI}_1 = 23^\circ$; $\text{SOI}_2 = -6^\circ$; $\text{Ti}_1 = 280 \mu\text{s}$; Rail pressure = 100 MPa; $T^{\circ}\text{EGR} = 110^\circ\text{C}$; EGR = 13 % to 30 %

The key issue regarding the operating point of 1640 rpm - 0,44 MPa IMEP is related to pollutant emissions including NOx, soot, CO and HC. Usually, CO are created during the premixed burning of fuel rich regions [9] while its oxidation rate is governed by the time elapsed above its freezing temperature (1700 K). Consequently, CO - resulting from a partial oxidation of unburnt fuel - occurs when air/fuel mixing is not intense enough. Generally speaking "under mixed conditions" that are also source of soot emissions. As opposed to that, some operating conditions lead to an excessive mixing between fuel and air that causes very lean local equivalence ratios. Under these "over mixed conditions", since not enough high local temperature is achieved to complete combustion, a high amount of CO is also released [10]. These last conditions often occur for very low load operating conditions such as 1640 rpm - 0,44 MPa IMEP.

HC emissions originate in regions where the flames quench on the walls or where an excessive dilution with air prevents the combustion process from either starting or going to completion. In the latter case, CO emissions under "over mixed" process are also released.

Whatever the way used to increase the air fuel mixing (e.g. increase in the number of nozzle holes, increase in swirl number, or decrease in W/H ratio of piston bowl), the impact on the combustion process is similar, as illustrated by the rate of heat release on Figures 2a, 2b, 2c. These figures reveal that an increase in mixing rate leads to :

- first, a lower magnitude of the maximal peak of the first combustion. This proves that a lower fraction of fuel introduced during the first injection is burnt. A larger fraction of fuel is therefore scattered in the combustion chamber and has more time to be mixed with air ;
- second, even though the main auto ignition delay is not affected, the main initial burning rate is smoother under the combined effects of :
 1. a less pronounced first burning rate. Chemical air fuel mixture preparation is then less mature ;
 2. an increased air/fuel mixing that leads to too much lean local equivalence ratio.

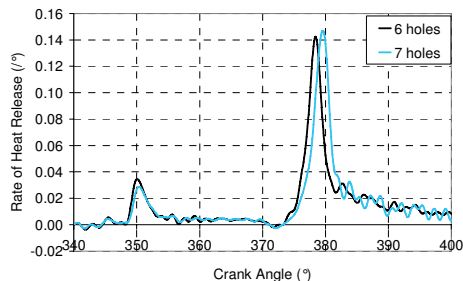


Figure 2a – Influence of hole number on RoHR

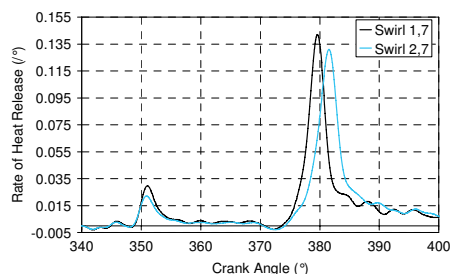


Figure 2b – Influence of cylinder head swirl on RoHR

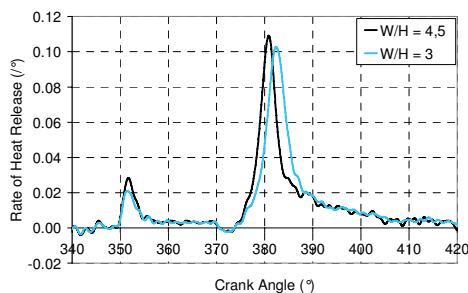


Figure 2c – Influence of W/H ratio on RoHR

Figure 16 in Appendix illustrates the influence of number of nozzle holes on emissions for each studied bowl along an EGR rate variation. In the same way, Figures 17 and 18 display respectively the results when varying the W/H ratio and the swirl number.

These figures reveal that an increase in mixing rate results in a decrease in soot emissions. This is particularly true for highest EGR rates because of the low

quantity of oxydants. On the other hand, no tendency is clearly identified for HC and CO emissions. All the physical phenomena involved in HC and CO formation and oxidation processes (such as mixing processes or fuel impingement on the walls [9, 10]) are implied.

Influence of the number of nozzle holes

With the narrowest bowl (W/H = 3), the increase in number of nozzle holes leads to a slight reduction in HC and CO emissions. With the intermediate bowl (W/H = 3,75), an increase in the number of holes from 6 to 7 results in higher CO and HC emissions while a further increase to 8 holes allows to achieve the lowest CO and HC emissions (the differences between number of nozzle holes are still low). Lastly, with the widest bowl (W/H = 4,5), the increase in number of holes from 7 to 8 causes higher CO and HC emissions.

These results reveal that the number of nozzle holes that allows to minimize HC and CO emissions depends on the piston bowl width. This is probably a consequence of the competition between two physical effects involved in HC and CO formation :

- first, an excessive mixing between fuel and air ;
- second, the interaction between the fuel jets and the walls.

For the narrowest bowl, as the bowl diameter is low, the second effect is the prevailing phenomenon on HC and CO emissions. As a result, an increase in number of nozzle holes - that implies a reduction in fuel sprays penetration - allows to reduce HC and CO emissions. When increasing the bowl diameter, the fraction of fuel impinging on the walls naturally decreases. In this case, an increase in the mixing rate is then suitable to reduce HC and CO emissions as far as over mixing is not exceeded.

Influence of W/H ratio

The decrease in W/H ratio of piston bowl from 4,5 to 3,75 leads to a drastic reduction of soot emissions particularly at high EGR rate ($\Phi > 0,65$). This result clearly illustrates the direct effect of an increased swirl value on the mixing rate. However, a further reduction of W/H to 3 does not lead to an additional decrease in soot emissions. This result may be explained by the fact that the fuel vapor jet-jet interaction is increased. Concerning HC and CO emissions, no monotonous tendency is identified when reducing W/H ratio. Once again, it is supposed that different physical processes are in competition. That is why the following explanations are given considering only the results done with the highest EGR rate.

The intermediate bowl (W/H = 3,75) delivers the lowest CO emissions and the same HC emissions as the narrowest bowl (W/H = 3). In addition, a zoom on PM emissions shows that it causes the lowest soot emissions. These results indicate that this bowl would provide the best mixing rate as it would offer the best trade off between the swirl intensity and the jet-jet proximity.

Influence of swirl number

The increase in cylinder head swirl number results in a strong decrease in soot emissions over the wide range of EGR rate while a decrease in CO emissions is only delivered with the highest EGR rate. It is assumed that the reduction of soot and CO emissions is coupled with the improvement in air fuel mixing. In the same way, HC emissions are not affected by the swirl (except for the highest EGR rate where an increased swirl results in higher HC emissions).

RESULTS AT 2500 RPM X 0,9 MPA IMEP WITH VCR 18:1

Injection settings : $SOI_1 = 48^\circ$; $SOI_2 = 11^\circ$; $Ti_1 = 230 \mu s$;
Rail pressure = 96 MPa ; $T^{EGR} = 130^\circ C$; EGR = 8 % to 20 %

Key issue regarding the operating point 2500 rpm - 0,9 MPa IMEP is different that considered at 1640 rpm - 0,44 MPa IMEP . Here, a particular attention is paid to NOx and soot emissions under respect of noise and ISFC limits because this operating point is operating on the extra urban driving cycle EUDC of the European driving cycle (NEDC). On the other hand, no formal attention is brought to CO and HC emissions because after treatment with catalytic system is already warmed up.

Generally at 2500 rpm - 0,9 MPa IMEP, the combustion process is governed by a premixed combustion phase generating combustion noise and a mixing-controlled combustion phase affecting the exhaust soot (the faster the burning rate is, the better the soot oxidation rate) and the fuel conversion efficiency (the faster the burning rate is, the better the combustion phasing in the cycle).

Influence of the number of nozzle holes and swirl

Figure 3 and Figure 4 illustrate respectively the influence of number of nozzle holes (6 or 7 holes) and swirl number on the rate of heat release. The increase in mixing rate does not affect the ignition delay and the premixed combustion mode phase. As a result, the combustion noise keeps constant (see Figures 19 and 20). On the other hand, the chemical energy release is faster during the mixing-controlled combustion phase when increasing the mixing rate. Accordingly, this leads to a decrease in soot and CO emissions as well as a slight reduction in fuel consumption (see. Figures 19 and 20). It should be noted, however, that with the intermediate bowl ($W/H = 3,75$), an increase in the number of nozzle holes to 8 does not lead to an extra gain in soot and CO emissions. This result highlights the drawback of an excessive increase in number of holes. As previously discussed in the introduction, an increase in number of nozzle holes causes a shorter fuel spray penetration and a decrease in the spacing between the

fuel jets. As a consequence, a worse utilization of air on the periphery of piston bowl and an increase in fuel vapor sprays overlapping may appear. This effect should be expected to be higher with the increase of fuel quantity and so with an increase in engine load.

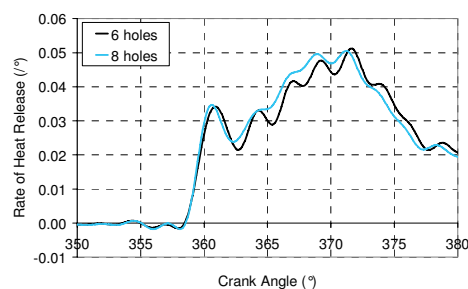


Figure 3 – Effect of number of nozzle holes on Rate of Heat Release

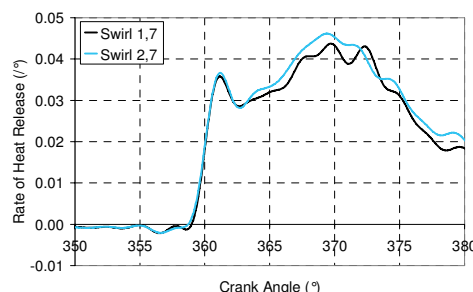


Figure 4 – Effect of swirl on Rate of Heat Release

Influence of W/H ratio

The reduction of piston width to depth ratio (W/H) from 4,5 to 3,75 leads to a strong decrease in PM and CO emissions of nearly 60% and 70 % at $INOx = 2,4 \text{ g/kWh}$ and constant noise and ISFC levels (Figure 21). However, no additional benefit is observed when reducing W/H from 3,75 to 3. The analysis of the profile of the burnt mass fraction given in Figure 22 points out that the exhaust PM and CO emissions are closely related to the rapidity of the combustion end (90% burnt mass crank angle). Consequently, the decrease in W/H ratio seems to be beneficial as far as the increase in swirl motion associated to the reduction of piston diameter does not cause an excessive fuel vapor jet-jet interaction

RESULTS AT 4000 RPM FULL LOAD WITH A CR 18:1

More than ever, the key issue at 4000 rpm full load is related to the ability of combustion system to deliver the highest mixing rate between fuel and air so that efficiency and equivalence ratio are promoted.

However, the optimization of the air utilization is all the more complex as the combustion lasts over around 40 crank angle degrees and a large amount of fuel is injected. In addition, a great mixing rate assuming a rapid burning rate and then a high efficiency does not

systematically match with a high fuel air equivalence ratio capacity.

Usually, the mixing rate is improved with an increase in fuel sprays penetration, an increase in swirl as far as fuel vapor jet-jet interaction does not occur, and an increase in air density. On the other hand, the mixing rate is downgraded with the impingement of fuel vapor on the piston bowl walls because this leads to subsequent jet-jet interaction [8] and a bad split of fuel between piston bowl and squish area [11].

Influence of W/H ratio

The influence of bowl width to depth ratio (W/H) at 4000 rpm, full load, have been studied over a wide range of boost pressure from 0,23 MPa to 0,27 MPa. Results are shown on the Figure 23.

The widest bowl (W/H = 4,5) always delivers the greatest efficiency. The analysis of rate of heat release for a constant fuel air equivalence ratio (Figure 24) shows that the burning rate during the mixing combustion phase is faster with the widest bowl. This may be explained by the fact that the free length of sprays is longer and, moreover, that the fuel vapor jet-jet interactions are reduced because the bowl diameter is wider and the swirl is lower.

For the lowest amount of fuel injected, the narrowest bowl (W/H = 3) gives a better efficiency than the intermediate bowl (W/H = 3,75). Nevertheless, with an increase of fuel quantity, the tendency is reversed. Thus, for the high amount of fuel, a linear correlation between the efficiency and the bowl diameter is identified. The explanations given above to account for the supremacy of a wide bowl regarding efficiency is not sufficient here to explain the obtained result for the low fuel quantity. It is thus assumed that even though the free length of fuel sprays is longer with the intermediate bowl than with the narrow bowl, the lack of swirl into the piston bowl leads globally to a less exchange between fuel and air. In addition, with the deepest bowl (W/H = 3), more fresh air is available within the bowl far from the fuel spray paths. This is essentially useful to burn up soot rapidly. So, this last explanation also allows to justify the lowest soot emissions when reducing piston diameter for the lowest amount of fuel.

However, with the increase of fuel quantity, the narrowest bowl implies more rapidly fuel vapor jet-jet interactions. In addition, as the amount of soot formed is higher, this bowl is penalized because first the extraction speeds outward the bowl toward squish area are lower in the bottom of the bowl and second the vertical distance from bottom of bowl to squish area is longer. Consequently, it takes a longer time to soot to reach squish area. This finally explains that for the largest amount of fuel, the larger the bowl diameter is, the lower soot emissions.

So, for the same smoke number objective of 3 fsn, the widest piston bowl delivers the greatest specific power. Furthermore, the gain in power with the widest bowl increases with the boost pressure (+ 0,5 kW/L at P2 = 0,23 MPa - +1,7 kW/L at P2 = 0,27 MPa) because :

- first the gain in efficiency increases with the boost pressure ;
- second, the ability of the widest bowl to accept high fuel air equivalence ratio with low smoke emissions is greater and greater with the increase of fuel air equivalence ratio.

Influence of the number of nozzle holes

The influence of number of nozzle holes at 4000 rpm -full load has been studied over a wide range of boost pressure from 0,23 MPa to 0,27 MPa. Results are shown in Figure 26 (illustration given for the bowl with W/H = 3).

A nozzle with a low number of holes always conducts to the best efficiency because it promotes an highest rate of heat release during the mixing combustion phase (see Figure 5). In addition, as expected according to the study of the influence of piston bowl W/H ratio, the higher the boost pressure is, the higher the efficiency gain. This result leads to conclude that an increase in air density promotes all the more the efficient combustion system designs.

Figure 26 shows that for an amount of injected fuel lower than 67 mm³/stroke, the 6-holes nozzle delivers the lowest soot emissions. On the contrary, for a fuel amount higher than 67 mm³/stroke, the lowest soot emissions are obtained with the 7-holes nozzle. This result may be explained by the fact that for the low quantity of fuel, the longest fuel spray penetration (due to a higher hole permeability) coupled with the largest spacing between the fuel jets of the 6 nozzle holes are more suitable for soot emissions. On the other hand, with the increase of fuel quantity, a larger fraction of fuel vapor jets from the 6-holes nozzle interacts with the piston edge walls and leads to a larger fuel vapor jet-jet interaction. Accordingly, an increase in number of nozzle hole that allows to reduce the fuel vapor spray penetration is then recommended.

Once again, the optimal configuration regarding soot emissions depends on the amount of fuel injected and the way this configuration conducts to fuel vapor jet-jet interaction.

Finally, with the increase in boost pressure, although the 6-holes nozzle always delivers the highest efficiency, the final difference of specific power between the two nozzle for a smoke level of 3 fsn is reduced . The specific power is measured at 1,2 kW/L at 0,23 MPa and 0,6 kW/L at 0,27 MPa. It may assume that for a still higher boost pressure, the 7 nozzle holes would deliver the highest specific power. It may so conclude that the choice of number of holes is strictly coupled with the power request.