Introduction

Currently, one of the major drawbacks of Diesel engines compared to classic gasoline engines is their noise. Legislation on maximum noise levels is becoming more stringent all over the world, and the demand for quiet engine designs is growing day by day. For these reasons, many efforts are being undertaken in order to reduce this disadvantage, especially as acoustical comfort now belongs to those features which play an essential role in the customer’s purchasing decision.

The knock and resonance due to the oscillation of burned gases in the combustion chamber seems to be the main excitation source of the engine block during the combustion process and is usually perceived as uncomfortable to the human ear. This is why several studies have dealt with combustion noise [1][2][3][4].

The combustion process in the engine can be analysed by measuring the pressure at a suitable point inside the cylinder. However, this methodology cannot be used for the optimum design of a combustion chamber because the manufacturing of various engine bowls is an expensive procedure. Hence, some kind of pre-design must take place before a definitive geometry is adopted and tested in an experimental bench. This is usually done with modelling, as is described for instance in [5] The modal theory is also an alternative [6], but this method is limited because it is neither possible to simulate the influence of the type and intensity of the excitation nor model a real bowl geometry.

In this paper the CFD program Fluent is used to study the resonance caused by the auto-ignition process in real engines with the aim of providing support to ongoing experimental work. Three aspects are analysed: first, three-dimensional calculations have been performed for three different combustion chamber geometries under the same operating conditions in order to determine the influence of the bowl geometry on the response in frequency and on the energy of resonance, two factors that allow qualifying the acoustic behaviour of an engine. Secondly, two of the geometries have been calculated taking into account the motion of the piston and a more realistic approach of the combustion process. Finally, this approach is used to analyse the influence of the pilot injection by comparing the results of the calculations made with only the main injection and with the pilot plus main injections.

Numerical methodology

The finite volume commercial program Fluent [7] with the standard k-ε turbulence model has been used to calculate all the cases presented here. The turbulence model was chosen because it is the most efficient and often used in reciprocating internal combustion engines calculations [8][9][10][11]. The 2nd order upwind differencing scheme (UD) is used for the momentum, energy and turbulence equations. Other computation parameters were the usual ones for this type of calculations[12][13].
For each case, a mesh independence study was performed to ensure the numerical accuracy of the solution. The temporal discretization is explicit, with a variable time step depending on the flow conditions and on the cell size, in order to keep a Courant number of approximately 1.

The initialization values for pressure and temperature in the combustion chamber were obtained from experimental tests[14][6], with both variables considered as homogeneous in the whole domain. The perfect gas assumption was made for the working fluid. The temperature on the walls was set using a correlation defined by Salavert [14], which depends on the mean piston speed of the engine, following a model proposed by Woschni [15].

In order to simulate the auto-ignition occurring in the combustion chamber, a small zone within the bowl perimeter has been defined as a patch where a pressure and temperature gradients are imposed at the start of the calculations. These correspond to the gradients caused by the start of combustion and they are determined experimentally with a combustion diagnosis programme [16]. The duration of the pressure pulse imposed corresponds to one time-step and its intensity is defined according to the quantity of energy liberated during the premixing combustion phase, calculated from the combustion diagnosis programme.

By placing various monitors in the engine bowl, it is possible to obtain the pressure and temperature temporal evolution in different points of the combustion chamber and study the effects of the oscillating pressure waves within the human hearing frequency range (20 Hz to 20 kHz).

**Specifics for steady state calculations**

For the bowl geometry study, three-dimensional calculations have been performed for the four geometries considered. In this case, the ignition area was reduced to a small zone, as indicated in Figure 1, corresponding to a single excitation source. Due to the symmetry with respect to a longitudinal plane that would go through the combustion chamber axis and through the middle of the excitation zone, the computational domain could be reduced to half the geometry, thus reducing computational cost. The computations were stopped when there was a clear indication that the pressure waves had reached a periodic steady state. Depending on the geometry mesh, the time step used ranged from 1.34 e-7 to 2.29 e-7 s, in any case amply sufficient to solve the high frequency range considered here.

**Specifics for moving mesh calculations**

The moving dynamic mesh method (MDM) of Fluent has been used to calculate the compression and expansion strokes of the engine, from the closing of the admission valve (-155.7 crank angle) to just before the opening of the exhaust valve (+ 130.8 crank angle).

For these calculations the time step was about 2.0 e-7 s, sufficient for the high frequency range considered in this study.

To simulate the combustion phase without introducing the combustion reactive equations a simplified approach is used here, which consists in imposing a source term in the energy equation equal to the heat released (HR) during the combustion process, calculated with the diagnosis programme.

As these calculations are computationally expensive, it has been decided in this first approach to reduce the problem to an axi-symmetric geometry. This would be the equivalent of a multi-hole injection simplified case giving rise to an annular ignition area. This way of simulating the auto-ignition process imposed by the axi-symmetric hypothesis is not entirely realistic, since it is known from experiments that ignition does not occur at the same instant for all sprays. It also means that the symmetric mode of resonance is ignored. However, there is evidence [17] that the shape and position of the ignition area have no significant influence on the frequency position of the combustion chamber resonance peaks of the asymmetric modes, though they affect the levels of energy of resonance. Since the main idea here is to verify the method, results will be assessed qualitatively, rather than quantitatively.
**Influence of the bowl geometry**

Three engine bowl geometries (see Figure 1) have been calculated under the same operating conditions and considering a single excitation source. The cylindrical bowl A is used as reference. Bowls B and C are real engine combustion chambers of the same engine, but have very different shapes.

The pressure monitors were distributed evenly in all cases, along longitudinal planes every 45º and at various depths, so as to cover the whole chamber (see Figure 1).

For the present study, results are presented in terms of the energy of resonance and the response in frequency obtained with a standard Fourier analysis. The energy of resonance (ER) is calculated from:

$$ER = \frac{1}{t} \int_{t=0}^{t} |p(t)|^2 dt$$  \hspace{1cm} (1)

where $p$ is the pressure of resonance obtained by filtering the total frequency response.

With the information from the monitors it is possible to appreciate the global influence of the bowl geometry on the resonance. In Figure 2 the frequency response obtained at a point located 3 mm below the excitation source is represented for all three combustion chambers. It has been verified [17] that the location of the excitation source has practically no influence on the position of the resonance frequency peaks, though it does affect their amplitude. Consistently with the modal theory, the comparison clearly shows that the frequency of the resonance peaks decreases when the bowl characteristic dimension is larger, which corresponds to the maximum diameter in the case of the re-entrant chambers considered here.

A representation of the pressure level in the whole domain is given by the energy of resonance (ER) at each monitor located at position $(R, \theta, l)$, as is illustrated by Figures 3 and 4 for plane $\theta = 0$º, a plane located at 180º from the excitation source. In all geometries the value of ER has been non-dimensionalized by the ER of the excitation pulse, so that the levels in each chamber can be directly compared. In each case, results are presented for two different positions of the excitation source (indicated in Figure 1), along four horizontal lines located at depths ranging from close to the piston head (line 1) to near the bottom of the combustion chamber (line 4). All lengths (diameters, depths, ...) have also been non-dimensionalized by the cylindrical bowl A diameter.

As may be seen in Figures 3 and 4, the ER increases with the radial distance of the monitor location due to the pressure waves bouncing off the walls, and the minimum values correspond to the monitors located in the central part of the bowl. The maximum levels of ER are reached for this 0º plane [17]. This is due to the fact that in stationary regime the pressure waves propagate in the radial direction only.

In all cases, the excitation source located deeper in the bowl gives raise to higher levels of energy of resonance. It is also worth noting that the monitors located nearest to either the cylinder head or the bowl bottom measure higher ER than monitors located more centrally in the bowl, up to 30% more on the periphery of the combustion chamber. Since in the experiments the pressure sensors are placed near the cylinder head, this means that sensors located near the periphery of the bowl will register the maximum energy of resonance, while those located near the bowl axis will register the minimum energy.

The comparison of the three combustion chambers allows to conclude that the central bump of the combustion chamber has a dampening effect on the pressure oscillations. Indeed, independently of where the excitation source is located, the cylindrical bowl has the highest ER levels. In addition, when the excitation source is located above the central ‘bump’ or at approximately the same level, bowls B and C have very similar ER levels, except at the periphery, with bowl B consistently slightly below. However, when the auto-ignition point is located deep in
the bowl, below the ‘bump’, bowl B clearly has the lowest levels of ER. It can be concluded that the marked ω shape of the bowl B, which is neatly divided in two volumes, dampens better the pressure oscillations.

**Influence of the pilot injection**

As indicated above, axi-symmetric calculations of the compression and expansion strokes of the engine, from the closing of the admission valve (-155.7 crank angle) to just before the opening of the exhaust valve (+ 130.8 crank angle) have been performed for bowls B and C. The experimental information required for the initialization and for the boundary conditions of the calculations has been taken from a real engine case running at 1500 rpm, with 76 Nm load and 30% EGR. A swirl coefficient of 2.15 was assumed, obtained from the experimental characterization of the engine on the flow test bench. As mentioned above, the auto-ignition process is simulated by imposing pressure and temperature gradients in the pre-defined excitation zone during one time-step, and then introducing the heat release (HR) function corresponding to the real pilot plus main injection case shown in Figure 5 as a source term in the energy equation.

Figure 6 shows the comparison of the frequency responses of both combustion chambers considered. It confirms that the frequency of the resonance peaks decreases when the maximum bowl diameter is larger (bowl C). In this case, the first resonance frequency is practically above the human hearing range, but this is due to the fact that the symmetric modes disappear because of the axy-symmetric assumption. Full 3D calculations will have to be performed to really assess quantitatively the frequencies of the resonance modes.

To study the influence of the pilot injection on the resonance modes engine C was also calculated with the HR function corresponding to a main injection only, but with the same quantity of fuel injected as in the pilot + main considered in the previous case (see Figure 5). Figure 7 shows that the effect of the pilot injection is to shift the position of the resonance peak towards higher frequency values. This can be explained by looking at the average temperature evolution in the engine (Figure 8). With the pilot injection, the temperature reached during the main combustion is higher, so that the speed of sound is higher and the pressure waves oscillate at a higher speed.

**Conclusions**

The geometry study of the combustion chambers has allowed to detect the damping influence of the central ‘bump’ on the energy of resonance, which is more significant when the bowl is clearly divided in two volumes and when the excitation source is deeper (larger spray penetration into the bowl).

The 2D calculations based on the axi-symmetric assumption may not be realistic, but they show the expected general trends and confirm the potential of this method. Full three-dimensional calculations will be launched as the next step, so that the solution may be compared with the experimental measurements.
Figure 2 Influence of the bowl geometry on the frequency response with a single excitation source

Figure 3 Evolution of ER in plane 0° for different bowl geometries with excitation position closest to combustion chamber bottom
Figure 4 Energy of resonance for different bowl geometries with excitation position closest to cylinder head

Figure 5 Heat release functions versus crank angle (pilot+main injections and main injection only)
Figure 6 Comparison of frequency responses of engines B and C for pilot+main injections

Figure 7 Comparison of frequency responses of engine C with pilot+main injections and main injection only.

Figure 8 Temperature evolution in engine C with pilot+main injections and main injection only.
References


