MODELLING OF PIEZO-ELECTRIC INJECTION IN A HIGH PRESSURE CELL AND VALIDATION FOR NON-EVAPORATING AND EVAPORATING CONDITIONS

P. Béard

IFP, Philippe.BEARD@ifp.fr

ABSTRACT The drive for substantial CO₂ reductions in gasoline engines in the light of the Kyoto Protocol and higher fuel efficiencies has increased research into Gasoline Direct Injection (GDI) engines. Moreover, using recently developed piezo-electric injection, injection strategies and stratified combustion can be improved. This makes possible combustion with low global fuel/air equivalence ratio which allows a fuel saving. The challenge for this type of DJ engines is to control the fuel/air equivalence ratio in the area of the spark plug in order to get a stable combustion. Thus, it requires an accurate description of injection conditions to simulate properly the spray behaviour, the air/fuel mixing and the subsequent combustion. A model was developed to simulate piezo-electric injection. It accounts for the variation of the orifice area during the transient opening and closing phases. Comparisons between numerical results and experimental measurements of spray characteristics in a high pressure cell are shown for both non-evaporating and evaporating conditions at different ambient pressures. In most of the studied configurations, simulations are in good agreement with the experimental visualizations. The impact of injection conditions and grid size on the spray penetration is demonstrated.

Keywords: Piezo-electric injection, modelling, transient conditions

1. INTRODUCTION

The challenge for Gasoline Direct Injection (GDI) engines is to control the fuel/air equivalence ratio in the area of the spark plug in order to get a stable combustion, to improve fuel efficiency and to reduce pollutant emissions. This can be achieved in particular through the use of the spray-guided concept and the recently developed piezo-electric injection which allows improved injection strategies. Computational Fluid Dynamics (CFD) has been used at IFP in the development phase to help in the design and optimization of downsized [1] and/or DI gasoline engines. This is particularly challenging because many complex physical phenomena are involved. First of all, injection conditions of the liquid fuel and the physics of spray formation are poorly known. Yet, as the spray and combustion models are improved, it becomes more and more crucial to know accurately the initial characteristics (diameter, velocity...) of the injected droplets. A model of transient injection conditions (TIC) was developed to reproduce the variations of the main spray global characteristics for a common-rail Diesel injection system [2]. The aim of this paper is to demonstrate the impact of transient injection conditions on the behavior of piezo-electric gasoline sprays.

The IFP-C3D code is briefly described in the first part of this paper. In a second part, comparisons between numerical results and experimental measurements of spray characteristics in a high pressure cell are presented for non-evaporating conditions. Then, the influence of injection conditions and grid size on the spray penetration is studied. Finally, results are shown for evaporating sprays in thermodynamic conditions typical of those encountered in a GDI engine.

2. CODE DESCRIPTION

2.1 Spray modeling

Three-dimensional simulations are performed using the multidimensional IFP-C3D code [3][4] that has been embedded in the AMESim platform [5]. This code solves the unsteady equations of a chemically reactive mixture of gases, coupled with the equations for a multi-component vaporizing fuel spray. The Navier-Stokes equations are solved using a finite volume method improved with the ALE (Arbitrary Lagrangian Eulerian) method. The code uses the well known time splitting decomposition. The temporal integration scheme is largely implicit.

Concerning the liquid phase, the physical phenomena of evaporation, breakup-up using the Wave-FIPA model [6] and spray/wall interaction [7] are included. Collision and coalescence are neglected as the collision model is strongly mesh dependent [8]. In the following simulations, the injected liquid spray is discretized using 50,000 computational parcels.

In a standard injection model (for example in KIVA), the injection velocity of the liquid particles is computed from the injection rate and an estimated constant value of the orifice outlet section:

\[ V_{in}(t) = \int [m(t),S_{in}] \] (1)

Due to the increase of injection pressure for piezo-electric injectors compared to swirl injectors, cavitation can occur inside the nozzle for GDI operating conditions [9] This can lead to high initial velocity of the spray (up to 200 m/s) that was observed in the experimental visualizations. To improve the prediction of the injection conditions, a strategy similar to that developed for the modeling of transient injection conditions of high pressure Diesel sprays [2] was adopted. In the case of a
piezo-electric injector (Fig. 1), the model accounts for the geometrical variations of the orifice area during the transient opening and closing phases of the orifice to compute the injection velocity:

\[ V_{\text{inj}}(t) = f[m(t), S_{\text{inj}}(t)] \]  

(2)

2.2 Parallelization

The OpenMP paradigm [10] was chosen for parallelization, as it is standardized, portable, scalable, adapted to modern super-scalar SMP machines and has an attractive performance to development cost ratio (easy to implement using only compiler directives). Profiling of the sequential version allowed to concentrate paralleling efforts on the most CPU time consuming routines: pressure solver, diffusion, matrix inversion (conjugate gradient method with SOR algorithm) and gradient terms (for pressure and Reynolds stress tensor). Overall, about 50 % of the code is parallelized providing a very good and scalable speed-up (around 3 for 4 processors). Details and extensive validations of the code can be found in [4].

3. NON-EVAPORATING CASES

3.1 High pressure cell

Simulations of the piezo-electric injection are conducted in a constant volume cell which enables to study the spray behavior under thermodynamic conditions similar to those of a GDI engine. Calculations are performed in a non uniform cartesian mesh (194400 cells: fig. 2). In the spray region close to the injector (located on the upper face), the cell volume is 1 mm³ as for typical engine grid spacings. Then it increases up to \( \Delta x = \Delta y = \Delta z = 3 \) mm in the bottom corners of the high pressure cell.

The injected fuel is iso-octane. Droplets are injected with an initial radius following a Rosin-Rammler distribution with a Sauter Mean Diameter SMR = 7 µm.

The injection pressure is 19 MPa and the injection duration is 0.6 ms. From the measured temporal evolution of the valve lift, the durations of the opening and closing phases of the outward opening injector are 0.2 ms and 0.1 ms respectively, i.e. 50 % of the total duration of the injection. These durations can also be estimated from the injection rate (Fig. 3).

3.2 Influence of ambient pressure

First, injection at room conditions (\( T_{\text{ch}} = 295 \) K, \( p_{\text{ch}} = 0.12 \) MPa) is considered.

Figure 4 shows comparisons between experimental visualizations by shadowgraphy and the spray computed with the standard injection model based on the maximum injection section. It appears that the spray penetration velocity is underestimated in this simulation. Accounting for the variations of the injection section during the opening and closing of the orifice leads to an increase of the injection velocity during these transient phases (Fig. 5). At the beginning of the injection, the injection rate is low but the injection velocity is already high as the injection section is very small. Then, the injection velocity decreases as the nozzle valve opens before increasing due to the increase of the injection rate. Computing the injection velocity with the maximum injection area leads to an underestimation of the spray momentum during both the opening and closing phases.

Comparisons between the spray computed with the new model and the experimental visualizations are presented in fig. 7. At the end of injection, both the spray shape and penetration are in good agreement with the experiment. This shows the impact of the injection conditions on the spray behavior. Nevertheless, the simulation seems to underpredict the presence of small recirculating droplets at the periphery of the spray. This leads to a smaller spreading of the droplets than in the experiment at \( t = 1.5 \) ms. Yet the temporal evolution of the drop SMR on the spray axis is reproduced quite well (Fig. 10).

At a higher ambient pressure, the spray shape is badly predicted with the standard injection model (Fig. 6). If the transient injection phases are accounted for, the computed liquid spray is in good agreement with the experimental visualizations (Fig. 8 and 9).
Figure 3 : Injection rate.

Figure 4 : Comparison between experimental visualization and liquid spray computed with the standard injection model ($T_{ch} = 295$ K, $p_{ch} = 0.12$ MPa).

Figure 5 : Influence of the transient opening and closing phases on the injection velocity normalized by the Bernoulli velocity $V_B$.

Figure 6 : Comparison between experimental visualization and liquid spray computed with the standard injection model ($T_{ch} = 295$ K, $p_{ch} = 0.45$ MPa).
Figure 7: Comparison between experimental visualization and liquid spray computed with the new injection model ($T_{ch} = 295$ K, $p_{ch} = 0.12$ MPa).

Figure 8: Comparison between experimental visualization and liquid spray computed with the new injection model ($T_{ch} = 295$ K, $p_{ch} = 0.3$ MPa).
Figure 9: Comparison between experimental visualization and liquid spray computed with the new injection model (T_{ch} = 295 K, p_{ch} = 0.45 MPa).

Figure 10: Comparisons between computed and measured SMR on the spray axis (T_{ch} = 295 K, p_{ch} = 0.12 MPa).

3.3 Mesh sensitivity
Additional tests are performed in a coarse mesh (Fig. 11) where the grid spacings in the spray region are increased to \( \Delta x = \Delta y = 1.5 \text{ mm} \) and \( \Delta z = 2 \text{ mm} \) (62400 cells).

Figure 11: Coarse computational mesh of the high pressure cell (xz plane).
The usual mesh sensitivity of this kind of simulations is demonstrated on Fig. 12 and Fig. 13 where the penetration velocity of the liquid spray is underestimated even with the new model because of numerical diffusion of the spray momentum. In this coarse mesh, results are worse with a standard injection model. This emphasizes the necessity of using an adequate grid spacing for engine simulation.
4. EVAPORATING CASES

More recently, ambient evaporating conditions similar to those in a GDI engine were considered. The injection pressure is 20 MPa and the injection duration is 0.6 ms.

In this case, the liquid phase is observed experimentally in a vertical plane passing through the injector axis by Laser Induced Exciplex Fluorescence (LIEF). It must be pointed out that the laser plane comes from the left side of the cell. As part of its energy is absorbed during the crossing of the spray, the intensity of the signal coming from the right part is more weak, resulting in asymmetric images (Fig. 14).

For the numerical results, liquid drops are colored by their radius. The same spray parameters as for the previous non-evaporating cases are used.

For $T_{ch} = 455$ K and $p_{ch} = 0.53$ MPa, the spray penetration is fairly well simulated, as well as the recirculation of small drops in the vortex at the tip of the spray (Fig. 14). For different ambient conditions, the shape and the penetration velocity of the liquid spray still agree quite well with the experimental visualizations (Fig. 15).

5. CONCLUSION

This paper presents several results obtained using latest developed injection model. The numerical modeling of piezo-electric injector was exposed in a high pressure cell for both non evaporating and evaporating ambient conditions. It was shown that neglecting the geometrical variations of the injection section during both the opening and closing phases of the orifice leads to an important underestimation of the spray momentum and penetration. For most of all the studied conditions, computed results with the improved injection model are in good agreement with the experimental measurements. The mesh sensitivity and the necessity of using an adequate grid spacing is emphasized.

Using the same spray parameters as those defined in this study, simulations were performed for a DISI engine at part load [11]. The computed evolution of the average in-cylinder pressure was very close to the experimental one. Further investigations will be carried out for a wider range of operating conditions.

6. NOMENCLATURE

\begin{itemize}
  \item $m$ mass flow rate \ [kg/s]
  \item $p$ pressure \ [Pa]
  \item $S$ section \ [m$^2$]
  \item $t$ time \ [s]
  \item $T$ temperature \ [K]
  \item $V$ velocity \ [m/s]
\end{itemize}
7. REFERENCES


ACKNOWLEDGEMENTS

This work was partially funded by GSM (Groupement Scientifique Moteur : IFP, PSA Peugeot-Citroën, Renault). The authors are most grateful to J-F. Le-Coz and B. Thirouard (IFP) for providing the experimental results. They are indebted to L. Hermant (IFP) for his expert technical assistance.

MAILING ADDRESS

Dr. Philippe BEARD
IFP
Powertrain Engineering (R10)
1-4 Av. de Bois Préau,
92500 Rueil Malmaison, France
Tel: +33-1-47526510
Fax: +33-1-47527068
E-mail: Philippe.BEARD@ifp.fr