

Cfd Diagnostic Methodology for the Assessment of Mixture Formation Quality in GDI Engines

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09/11/2011Federico Millo, Marco Badami, Andrea Bianco and Enrico Delogu
Politecnico di Torino

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ABSTRACT

The fuel injection plays a crucial role in determining the mixture formation process in Gasoline Direct Injection (GDI) engines.

Pollutant emissions, and soot emissions in particular, as well as phenomena affecting engine reliability, such as oil dilution and injector coking, are deeply influenced by the injection system features, such as injector geometric characteristics (such as injector type, injector position and targeting within the combustion chamber) and operating characteristics (such as injection pressure, injection phasing, etc.).

In this paper, a new CFD methodology is presented, allowing a preliminary assessment of the mixture formation quality in terms of expected soot emissions, oil dilution and injector coking risks for different injection systems (such as for instance multihole or swirl injectors) and different injection strategies, from the early stages of a new engine design.

The proposed methodology, after a number of validation steps with experimental data gathered from different injection systems, has then been applied to guide the choice of the injector type, injection targeting and injection pressure and phasing for a GDI engine.

INTRODUCTION

Gasoline Direct Injection (GDI) has been gaining an increasing interest in the last years, especially in combination with downsizing and turbocharging, due to its noteworthy potential for fuel consumption and CO₂ emissions reduction.

Numerical simulation currently plays a fundamental role in the optimization of the mixture formation process for

Gasoline Direct Injection (GDI), since the analysis of fuel spray atomization and penetration is a key factor for the optimization of small-bore high-speed automotive engines and for the development of advanced combustion strategies, such as, for instance, the Stratified Charge Combustion. The penetration of the fuel spray into the cylinder along with the distribution of the fuel in the combustion chamber are as a matter of fact not only crucial parameters in order to reduce pollutant emissions and soot emissions in particular, but also extremely important in determining several phenomena affecting engine reliability, such as lubricant oil dilution and injector coking.

In this paper, a new CFD methodology is therefore presented, allowing a preliminary assessment of the mixture formation quality in terms of expected soot emissions, oil dilution and injector coking risks for different injection systems (such as for instance multihole or swirl injectors) and different injector geometric (e.g. injector position and targeting within the combustion chamber) and operating characteristics (e.g. injection pressure, injection phasing, etc.).

The proposed methodology is aimed to get a preliminary assessment, from the early stages of a new engine design, of the mixture formation quality and does not require the complete engine cycle simulation, including the combustion phase, but only experimental data for the characterization of the fuel injection system which can be usually obtained from optical investigations in pressurized vessels. Thanks to the identification of a reduced number of mixture quality indexes that can be evaluated at spark timing, significant reductions of CPU time can be achieved and the need of sophisticated combustion models can be avoided, although a considerable level of information can be gathered.

Finally, an example of the application of the proposed methodology to guide the choice of the injection system characteristics for a GDI engine is presented.

MODELING METHODOLOGY

Since the analysis of fuel spray atomization and penetration is of crucial importance for the assessment of the mixture formation quality, reliable fuel spray models are necessary. Two different spray models have therefore been developed and set-up, the first for swirl injectors, the second for multihole injectors. Although the development and the validation of these models has been already described in detail in previous works, a brief summary has been reported in the following sections for reader's convenience.

SPRAY MODELING FOR SWIRL INJECTORS

Although numerous models are available in literature to describe the break-up of fuel sprays, and many of them are currently available in the most widely used software codes for internal combustion engines CFD simulations [1,2], for swirl injectors a modeling methodology already developed in a previous work was preferred [3].

This methodology is characterized by a reduced amount of input data, since its main aim is to provide a reliable spray model suitable to be used even in the first phases of an engine development, thus when only the main features of the injector may be known.

According to the most diffuse approach for the injection modeling in internal combustion engines, the spray is simulated using the Eulerian-Lagrangian approach, in which the gaseous phase is modeled with the Eulerian conservation equations, while the Lagrangian equations are applied to the liquid droplets. Intrinsic in this methodology is the assumption that the liquid core at the exit of the injector can be properly modeled using droplets of a suitable dimension [4,5].

The thin liquid film, just downstream to the nozzle hole exit, disintegrates into droplets and fluid ligaments, because of the exponential growth of instable surface waves in the processes known as primary atomization. The lack of an adequate physical understanding and the long computational time that is necessary in a complete cycle simulation, makes it difficult to carry out a direct simulation of this phenomenon; the fluid is therefore usually considered in the form of droplets out of the nozzle hole exit. This approach does not intend to accurately simulate the spray characteristics in the near injector region but instead intends to deal with an equivalent spray, in the computational domain, which acts like the real spray once the droplets are sufficiently far from the injector exit.

The procedure, which calculates the characteristics to be assigned to the fuel droplets, has to receive the following parameters as inputs:

- Fuel Mass Flow Rate
- Initial Droplet Velocity
- Spray Cone Angle
- Droplet Size Distribution

The first parameter is usually known from the injector characteristics, while the others are evaluated in the proposed model through the phenomenological approach described here below.

The potential theory [6] is used to derive the efflux velocity from the imposed mass flow rate and the geometrical characteristic of the injector, and the axial velocity v_{axial} of the droplet is therefore determined as:

$$v_{axial} = \frac{C_D}{(1-k^2)} \sqrt{2 \frac{\Delta p}{\rho}} \quad (1)$$

where Δp is the relative injection pressure, ρ is the fuel density, and k is the cavity factor, i.e. the ratio between the internal swirl chamber diameter d_c and the external chamber diameter d_e . C_D is the discharge coefficient, that takes into account:

- the difference between the theoretical and the actual velocity at the nozzle exit due to the friction losses;
- the fact that part of the pressure energy is not converted into axial velocity;
- the difference between the nozzle hole section and the actual efflux area (which is a circular crown) due to the hollow cone spray shape.

Its value is a function of the cavity factor k and can be calculated using the potential theory, by which the following equation is derived:

$$C_D(k) = F \left[\sqrt{1-k^2} - k^2 \ln \left(\sqrt{\frac{1}{k^2} - 1} + \frac{1}{k} \right) \right] \quad (2)$$

where F is a correction factor (for which a value of 0.7 was found in [3] to provide the best fit of the experimental data for GDI injectors) which is required since the potential theory neglects the friction losses and the cavitation phenomena.

The spray cone angle too is a function of the cavity factor k and then of the geometrical parameters only:

$$\alpha = 2 \arctan \sqrt{\frac{k^2}{1-k^2}} \quad (3)$$

The Sauter Mean Diameter (SMD), according to Wang and Lefebvre [6], is then calculated by means of the following relationship, which can be deduced by the “dimensional analysis” approach:

$$SMD = 4.52 \cdot \left(\frac{\sigma^{0.5} \mu^2}{\rho_a \Delta p^2} \right)^{0.25} h_s^{0.25} + 0.39 \cdot \left(\frac{\sigma \rho}{\rho_a \Delta p} \right)^{0.25} h_s^{0.75} \quad (4)$$

where σ is the fuel surface tension, μ is the fuel viscosity, ρ_a is the air density and h_s is the thickness of the conical liquid sheet, which can be determined as a function of the cavity factor and of the spray cone angle as shown in [3].

The previous relationships have been implemented in a Fortran subroutine, which allows to inject the fuel parcel with the axial velocity obtained from (1), and with a diameter extracted randomly from a Rosin-Rammler size distribution, the SMD of which is given by Eq. 4 [6,7,8]. The Rosin-Rammler distribution characteristics are then determined as follows:

$$N = 1 - e^{-\frac{(D^q p)}{s^q}} \quad (5)$$

where N is the fraction of droplets whose diameter is smaller than diameter D , and p , q , s are the distribution parameters. Numerical processing [9] of a large set of experimental results concerning different injectors with various fuel pressure levels, showed that the best fit of experimental data could be obtained by setting parameter p at 2.5 and parameter q at 3 respectively. The s parameter is then directly related to the Sauter Mean Diameter calculated in (4) as shown in Eq. 6:

$$s = SMD \cdot p^{\frac{1}{q}} \frac{\Gamma\left(\frac{q+2}{q}\right)}{\Gamma\left(\frac{q+3}{q}\right)} \quad (6)$$

where Γ is the gamma function.

This procedure has been extensively tested with different type of injectors and different injection pressures and backpressures during its development [3], and further tests

were performed for the injector used in the present work. Some examples of the comparisons between the simulation results and the experimental data are shown in Figures 1,2: a good agreement can be observed between numerical and experimental sprays, both for the early spray development and for the fully developed spray. Finally, Figure 3 shows a satisfactory agreement also for the radial distribution of the spray axial velocity.

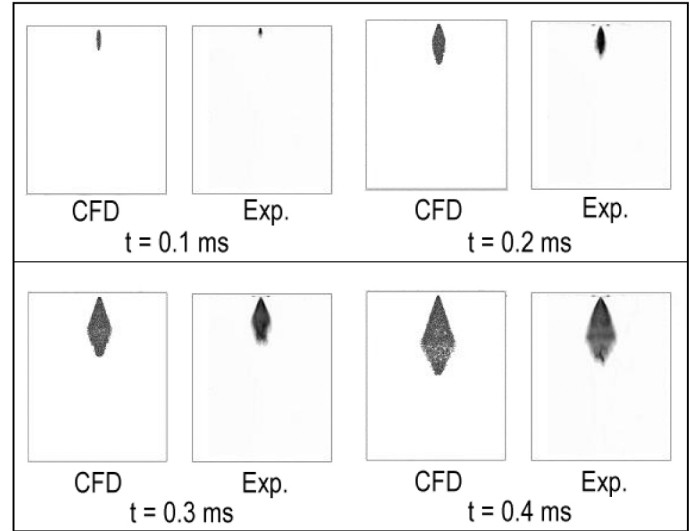


Figure 1. Comparison of the early spray evolution between numerical simulation and experimental data (swirl injector, $p_{inj} = 50$ bar, constant volume vessel with 1 bar backpressure [9]).

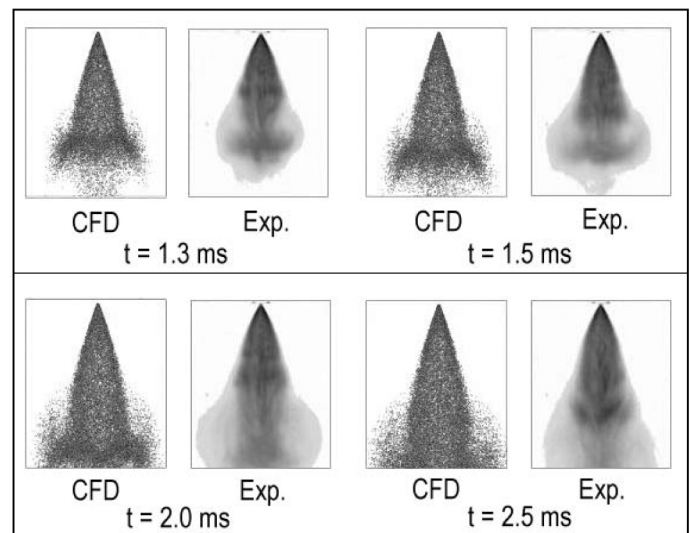


Figure 2. Comparison of the fully developed spray between numerical simulation and experimental data (swirl injector, $p_{inj} = 50$ bar, constant volume vessel with 1 bar backpressure [9]).

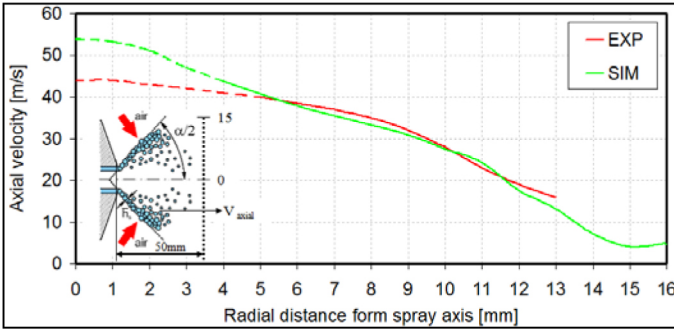


Figure 3. Numerical and experimental axial velocity measured 50 mm from the tip of a swirl injector, at different radial distances from the spray axis (2, 1 - 4, 5 ms from SOI, [9]).

SPRAY MODELING FOR MULTIHOLE INJECTORS

Assuming that for multihole injectors the main spray characteristics can usually be gathered from experiments in optical accessible pressurized vessels and can be available even in the first phases of a new engine development, the proposed methodology does not attempt to set-up a model able to capture the physical phenomena governing the fuel jet primary break-up, but rather intends to obtain an equivalent spray in the computational domain, which acts like the real spray once the droplets are sufficiently far from the injector exit. The main characteristics of this equivalent spray are determined as summarized here below, aiming to achieve the best fit with the experimentally measured characteristics of the real spray.

Initial droplet velocity, size distribution and spray angle

The fuel enters the cylinder with the following three main calculated characteristics:

- initial droplet velocity
- initial droplet size distribution
- initial droplet direction (spray angle)

The initial droplet velocity v is calculated by means of the relative injection pressure across the nozzle:

$$v = C_d \sqrt{\frac{2\Delta p}{\rho}} \quad (7)$$

where the nozzle discharge coefficient C_d is set to 0.7, considering this value representative for commercial multihole injectors [9,10,11].

The droplets injected into the cylinder are then characterized by a Rosin-Rammler size distribution, whose parameters have

the same values introduced for the swirl injector (i.e. $p = 2.5$, $q = 3$ and s determined on the basis of Eq. 6 in order to obtain a certain SMD value). The SMD value is then determined by the experimental data fitting procedure described hereafter.

As far as the initial spray angle is concerned, the initial direction of a fuel droplet is calculated by means of a maximum spray angle α_{max} and a random spatial distribution. The maximum spray angle is then determined, along with the SMD, by means of the experimental data fitting procedure described hereafter.

Experimental data fitting procedure

The SMD and spray maximum angle are determined aiming to achieve the best fit with the experimentally measured characteristics of the real spray.

First, a correlation coefficient R between numerical and experimental data is calculated:

$$R = 1 - \frac{\sum (Exp - Sim)^2}{\sum Exp^2 - \frac{(\sum Exp)^2}{n}} \quad (8)$$

where Exp represents an experimentally measured value (e.g. the spray tip penetration at a certain instant after the start of injection), Sim represents the corresponding calculated CFD value and n is the total number of compared data. The correlation coefficient R can be evaluated considering the spray tip penetration (R_{PEN}), the SMD (R_{SMD}) and the droplet velocity (R_{VEL}) during the whole injection process. After obtaining the correlation coefficients for all the three abovementioned main spray characteristics, a global correlation coefficient F_{out} can be found as a weighted average of the single correlation coefficients:

$$F_{out} = w_{PEN} \cdot R_{PEN} + w_{SMD} \cdot R_{SMD} + w_{VEL} \cdot R_{VEL} \quad (9)$$

Hence, once the global correlation coefficient F_{out} has been defined, the couple of values of SMD and maximum spray angle for which F_{out} is maximized is found through a direct optimization search, in order to obtain the best numerical fit for the available experimental data.

An example of the results obtained for the multihole injector used in the present work is reported hereafter.

The main test conditions for the available experiments in a pressurized vessel are summarized in Table 1.

Table 1. Multihole injector: available experimental data sets

Case	p_{inj} [MPa]	p_a [MPa]	T_a [K]	Injected mass [mg/hole]
1	5	0.1	293	6.76
2	8	0.1	293	8.33
3	10	0.1	293	9.16

The best fitting with experimental data was obtained by setting $SMD = 28 \mu m$, $\alpha_{max} = 17^\circ$. A good agreement between the experimental data and simulation results could be achieved for all the three different injection pressures. As an example, the comparisons between experimental and simulation results for the test case #3 are shown in Figure 4,5,6 as far as spray tip penetration, SMD and axial velocity are concerned, while a qualitative comparison between real and “virtual” spray images is shown in Figure 7.

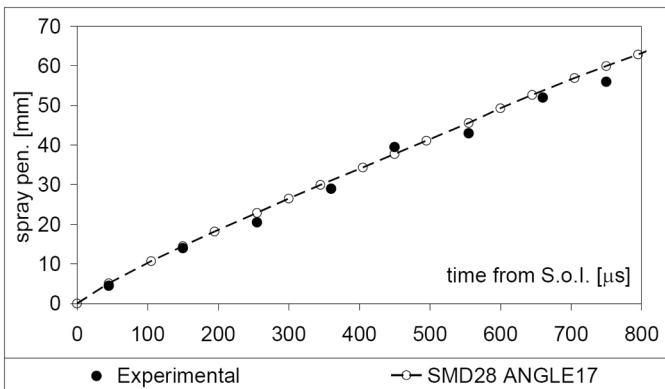


Figure 4. Spray penetration comparison for test case #3 ($p_{inj} = 10 \text{ MPa}$)

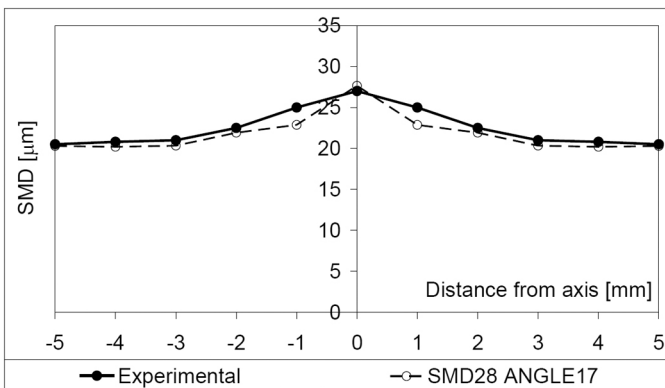


Figure 5. Sauter Mean Diameter (SMD) comparison for test case #3 ($p_{inj} = 10 \text{ MPa}$) - 40 mm distance from the nozzle

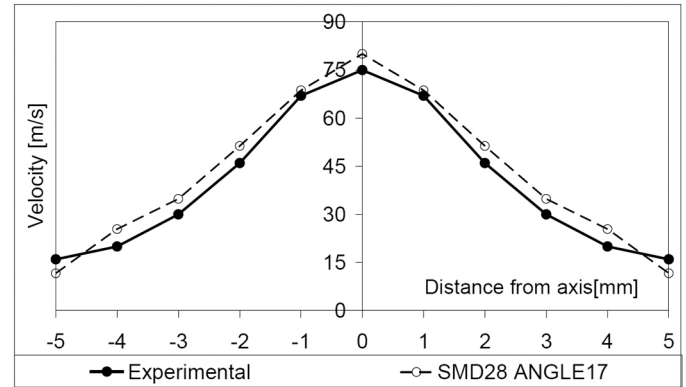


Figure 6. Axial velocity comparison for test case #3 ($p_{inj} = 10 \text{ MPa}$) - 40 mm distance from the nozzle

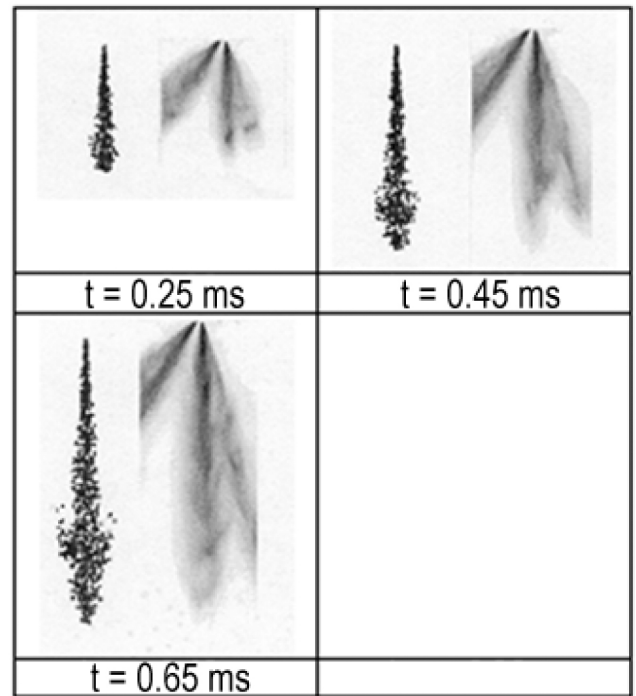


Figure 7. Comparison of the spray evolution between numerical simulation and experimental data at different times after the Start of Injection for test case #3 ($p_{inj} = 10 \text{ MPa}$)

MIXTURE HOMOGENEITY ASSESSMENT AND INJECTOR COKING DIAGNOSTIC

Achieving a “target” (stratified or homogenous, depending on the combustion concept) air-fuel ratio in the combustion chamber is certainly the main goal of the injection system in a GDI engine, and CFD analysis can undoubtedly provide valuable information by obtaining a preliminary assessment,