Energy Conversion and Management 132 (2017) 221–230

Contents lists available at ScienceDirect

Energy Conversion and Management

journal homepage: www.elsevier.com/locate/enconman

Assessment of compressibility effects on internal nozzle flow in diesel injectors at very high injection pressures

F.J. Salvador [a,*], J. De la Morena [a], J. Martínez-López [b], D. Jaramillo [a]

[a] CMT-Motores Térmicos, Universitat Politècnica de València, Spain
[b] Williams Grand Prix Engineering Limited, United Kingdom

ARTICLE INFO

Article history:
Received 30 September 2016
Received in revised form 14 November 2016
Accepted 15 November 2016

Keywords:
Nozzle
Modelling
Diesel
Dynamic
Compressibility
CFD

ABSTRACT

Diesel fuel injection systems are being used at higher injection pressure conditions over time because of more stringent emissions requirements. Thus, the importance to properly take into account the fluid compressibility on injection CFD simulations is also increasing. In this paper, an investigation of the compressibility effects in nozzle flow simulations has been carried out for injection pressures up to 250 MPa. To do so, the fluid properties (including density, viscosity and speed of sound) have been measured in a wide range of boundary conditions. These measurements have allowed to obtain correlations for the fluid properties as a function of pressure and temperature. Then, these equations have been incorporated to a CFD solver to take into account the variation of the fluid properties with the pressure changes along the computational domain. The results from these simulations have been compared to experimental mass flow rate and momentum flux results, showing a significant increase in accuracy with respect to an incompressible flow solution.

1. Introduction

In the last decades, diesel engine researchers have focused on minimizing the exhaust emissions maintaining the thermal efficiency advantage compared to gasoline engines. In particular, efforts have been made to achieve a combined reduction of nitrogen oxides and soot particles, which are characteristic of the lean diffusive combustion process existing in such engines [1,2].

Two main paths have been followed to reduce exhaust emissions in diesel engines. On the one hand, several aftertreatment components, such as Diesel Particulate Filter (DPF), Diesel Oxidation Catalyst (DOC), Selective Catalyst Reduction (SCR) or Lean-NOx Trap (LNT) have been placed at the engine outlet to collect and/or convert the exhaust emissions before reaching the atmosphere [3,4]. On the other hand, new combustion modes with high levels of Exhaust Gas Recirculation (EGR) and higher rates of premixed combustion have been implemented to reduce the emissions at engine-out [5–8]. The performance of the fuel injection system has been proven as critical for such strategies, since it controls the atomization and fuel-air mixing processes [9–11].

Many authors have tried to study the characteristics of the flow inside the fuel injector, and in particular inside the nozzle orifices. Several studies have made use of transparent geometries for this purpose, but many of them explored simplified geometries [12–15] or were significantly limited in the maximum achievable injection pressure [16–18]. Thus, Computational Fluid Dynamics (CFD) tools have been developed on the last decades as a tool to get further insight in the relationship between the nozzle geometry, the internal flow characteristics and the hydraulic conditions at the nozzle exit [19–22], which are a necessary input for spray combustion models [23–26].

The fuel physical properties (mainly density and viscosity) have a significant impact on the internal nozzle flow characteristics. Battistoni and Grimaldi [23] compared the internal flow and near-nozzle spray details for a standard diesel fuel and a soybean methyl ester (SME), showing that the different viscosity among them severely impacts both the outlet mass flow rate and the spray features. Similar conclusions about the effect of the fuel properties have already been seen both experimentally and numerically for other kinds of biodiesel [27–31] and for winter fuel formulations [32–34]. Recently, a few authors [35–39] have showed that it is important to consider not only the changes in the fuel properties related to the fuel composition, but also those related to the different temperature and pressure conditions along the nozzle geometry, which are traditionally neglected.

In the current paper, an effort to understand the impact of compressibility effects on internal nozzle flow simulations at very high

* Corresponding author at: CMT-Motores Térmicos, Universitat Politècnica de València, Camino de Vera s/n, E-46022, Spain.
E-mail address: fsalvado@mot.upv.es (F.J. Salvador).

http://dx.doi.org/10.1016/j.enconman.2016.11.032
0196-8904/

Nomenclature

a_f	fuel speed of sound
A_o	geometrical nozzle outlet area
C_d	discharge coefficient, $C_d = \frac{\dot{m}}{\rho_f \cdot A_o \cdot u_b}$
D_i	geometrical nozzle inlet diameter
D_o	geometrical nozzle outlet diameter
k-factor	nozzle conicity, $k-factor = \frac{D_i[\mu m] - D_o[\mu m]}{10}$
$\dot{m}$	mass flow
$\dot{M}$	momentum flux
P	fluid pressure
P_b	discharge pressure
P_{inj}	injection pressure
T	fluid temperature
u_{eff}	effective outlet nozzle orifice velocity
u_b	theoretical outlet orifice velocity, $u_b = \sqrt{\frac{2\cdot(P_{inj}-P_b)}{\rho_f}}$

Greek symbols

ΔP	pressure drop, $\Delta P = P_{inj} - P_b$
ρ_f	fuel density
υ_f	kinematic viscosity
μ_f	dynamic viscosity
μ_0	dynamic viscosity at 0.1 MPa pressure

injection pressure (up to 250 MPa) has been performed. For this purpose, the fuel used for the study has been widely characterized at different levels of temperature and pressure, producing the corresponding correlations for the fuel density, viscosity and speed of sound. Then, the hydraulic behavior of the injector has been determined in terms of injection rate and momentum flux for different levels of injection pressure and backpressure. These results have been finally compared to internal flow CFD simulations carried out with two strategies: constant fuel properties (incompressible) and pressure-dependent fuel properties (compressible). This procedure allows to quantify the differences obtained in the main flow parameters when compressibility effects are considered compared to the more simple incompressible solution generally seen in the literature [15,40,41].

The paper is structured in 5 sections. In Section 2, the main experimental methodologies used along the study are described, together with the correlations obtained for the main fuel physical properties. Section 3 details the setup used for the internal flow CFD study, whose main results are depicted in Section 4. Finally, the main conclusions obtained from the work are drawn in Section 5.

2. Experimental tools

In this section, the main experimental techniques used for the study are briefly described.

2.1. Nozzle geometry determination

For the current study, a solenoid-driven diesel injector with a 7-orifices convergent nozzle has been used. In order to perform the internal nozzle flow simulations, it is necessary to have all its geometrical details. To do so, a previously developed and validated silicone molding technique has been employed. The technique is based on the injection of the silicone on a semi-liquid state into the nozzle, once the needle has been removed. After a few hours, the silicone becomes solid and can be extracted, maintaining the internal geometry of the sac and the orifices. The mold is later inspected using a Scanning Electron Microscope, determining the corresponding nozzle dimensions.

An example of the pictures obtained through this process can be seen in Fig. 1, while more details on the experimental technique are available in [42]. Finally, the final geometrical values of the nozzle used for the study can be seen in Table 1. In this table, R_a and R_b are the rounding radii at the orifice inlet in the upper and lower side of the orifice, respectively; D_i, D_o and D_m are the diameters in the inlet, outlet and middle sections of the orifices; and *k-factor* is a parameter related to the nozzle orifice conicity, defined as:

$$k-factor = \frac{D_i[\mu m] - D_o[\mu m]}{10} \quad (1)$$

Since the nozzle orifices are significantly convergent (as it can be seen from its high value of *k-factor*), low probability of cavitation formation inside the nozzle is expected [43,44]. Nevertheless, some cavitation could appear when very high injection pressures are used. This will be further analyzed in Section 4.

2.2. Fuel properties characterization

As a first step, the main physical properties of the fuel have been measured under a wide range of pressure and temperature conditions. In particular, a standard European winter diesel fuel has been used. Density measurements were performed on a hydrometer, based on the ASTMD1298 procedure, while a standard viscometer was used to characterize the fuel viscosity. Finally, a custom-made facility was constructed to characterize the speed of sound. This facility was based on a standard common-rail system, onto which a long tube has been installed between the rail and the injector. On that line, two high-speed piezoelectric pressure transducers have been installed at two different positions. Once the injector is commanded and the injection event takes place, a pressure wave is generated inside the system. Knowing the distance between these two transducers, it is possible to characterize the speed of sound by measuring the time lapse that the pressure perturbation takes to travel to one sensor to another. More information about the experimental setup can be seen in [45].

Fig. 2 shows the results from the fuel characterization for a range of 0.1–300 MPa in pressure and 300–400 K in temperature, which are representative of the usage of diesel fuel in advanced common rail systems. These data have been correlated as a function of pressure and temperature, finding the following relationships:

$$\begin{aligned}\rho_f = {} & 826.5 - 1.0217(T-298) + 1.251\cdot10^{-3}(T-298)^2 \\ & + 0.6035(P-0.1) - 8.27\cdot10^{-4}(P-0.1)^2 \\ & + 1.44\cdot10^{-3}(P-0.1)(T-298).\end{aligned} \quad (2)$$

$$\mu_f = 10^{-3}\mu_0\cdot10^{\left[\left(-1.48+5.86\mu_0^{0.181}\right)\cdot\left(\frac{P-0.1013}{1000}\right)\right]} \quad (3)$$

$$\mu_0 = 3.2158\cdot\exp[0.0263(T-298)] \quad (4)$$

$$\begin{aligned}a_f = {} & 1350.6 - 3.1485(T-298) + 4.4928(P-0.1) - 6.96 \\ & \cdot10^{-3}(P-0.1)^2 + 7.4\cdot10^{-3}(P-0.1)(T-298)\end{aligned} \quad (5)$$

where ρ_f is the fuel density in kg/m^3, μ_f is the fuel dynamic viscosity in Pa s, μ_0 is the fuel dynamic viscosity at 0.1 MPa of pressure, a_f is the speed of sound of the fuel in m/s, P is the fuel pressure in MPa and T is the fuel temperature in K.