International Journal of Heat and Fluid Flow 68 (2017) 79–86

Contents lists available at ScienceDirect

International Journal of Heat and Fluid Flow

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

# A new approach to compute temperature in a liquid-gas mixture. Application to study the effect of wall nozzle temperature on a Diesel injector

Raúl Payri, Jaime Gimeno, Pedro Martí-Aldaraví*, Mary Alarcón

*CMT - Motores Térmicos, Universitat Politècnica de València, Edificio 6D, Camino de Vera s/n, Valencia, 46022, Spain*

ARTICLE INFO

*Article history:*
Received 1 June 2016
Revised 2 November 2016
Accepted 29 December 2016

*Keywords:*
Diesel
Nozzle
Temperature
Boundary condition
Compressible fluid
Engine Combustion Network

ABSTRACT

Thermal effects on internal combustion engines have been always a hot topic. Its effects also on the injection system are still under research. In this work, a homogeneous model called Eulerian Spray Atomization (ESA) model is used to simulate Engine Combustion Network (ECN) Spray A conditions. A new approach is used for the thermodynamic model in Diesel spray simulations. Experimental values of the liquid enthalpy are directly used to obtain the temperature, without any need of knowing its heat capacity. This allows to accurately study heating and cooling phenomena inside the Diesel nozzle. The results show that two different boundary conditions can be applied at nozzle walls, either fixed temperature or adiabatic walls, and obtain good prediction of injection parameters. Also, the effect of fuel temperature has been investigated. For Spray A conditions, nozzle and spray parameters are the same regardless the fuel temperature, but fluid properties inside the nozzle (density, velocity...) significantly change.



## 1. Introduction

Reduction of fuel consumption is one of the goals in research of internal combustion engines (ICE). Since few years ago, one of the strategies under investigation for ICE is the increase of injection pressure (Wang et al., 2010; Chan et al., 2014). Temperature on fuel systems has been always a topic under investigation (Catania et al., 2004; Salvador et al., 2016), but under such high pressure conditions, thermal effects become a major issue (Strotos et al., 2015).

In order to study the thermal effects on the injection systems, fuel properties need to be known in advance as function of pressure and temperature (Park et al., 2009; Payri et al., 2011). Many documentation is available on pure components, e.g. n-Dodecane (Khasanshin et al., 2003), and blends of differente single-component fuels (Pastor et al., 2015). But that is not the case of real fuels, which have many components plus impurities. Thereby, a single component representative of Diesel fuel such as n-Dodecane is selected for this study. Moreover, n-Dodecane is the standard fuel used by Engine Combustion Network collaborators to perform both experiments and simulations.

Under these circumstances, the main goal of this work is to computationally assess the influence of temperature on the flow pattern inside the nozzle and the near-field behavior of the spray for non-evaporative conditions. Additionally, differences between assuming adiabatic nozzle walls and constant temperature walls (heated/cooled) are analyzed. To do so, a model able to solve the coupling between nozzle flow and the spray would be desirable. New homogeneous Eulerian modeling (Vallet et al., 2001; Blokkeel et al., 2003; García-Oliver et al., 2013) seems to be the best approach. From those, the only one that has demonstrated to deal seamlessly with nozzle flow and spray is the Eulerian Spray Atomization (ESA) model (Salvador et al., 2014; Desantes et al., 2014; Payri et al., 2015b).

In order to validate the simulations, it is important to have extensive and high quality experimental data. The Engine Combustion Network (ECN) (Bardi, 2012) is an international collaboration between research institutes and universities that addresses the study of fuel injection with reliable standards. That is why their condition called “Spray A” has been taken as a reference. It consists of a single-hole nozzle aligned with the injector axis employed for research purposes. Also the injection conditions (given in next sections) are representative of actual engine conditions in normal car operation.

This paper is divided in four sections. Additional to this brief Introduction section, in Section 2 the basis of the thermal and computational modeling are explained, together with the configuration of the simulations. The following Section 3 shows the results obtained and its analysis. And the final Section 4 summarizes the main findings of this investigation.

* Corresponding author.
*E-mail address:* pedmar15@mot.upv.es (P. Martí-Aldaraví).

http://dx.doi.org/10.1016/j.ijheatfluidflow.2016.12.008
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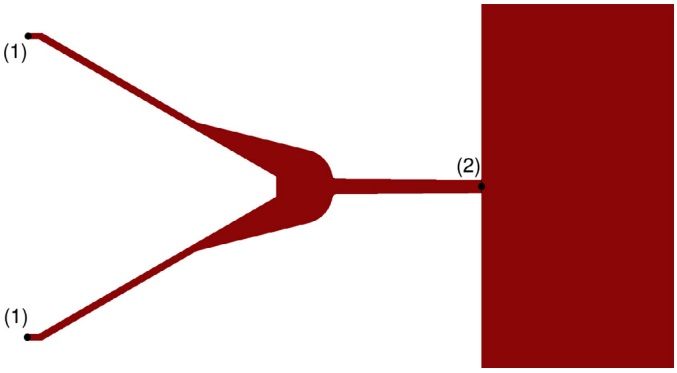


**Fig. 1.** Sketch of a single-hole Diesel nozzle.

## 2. Methodology

### 2.1. Theoretical background

Pressure gradient inside Diesel nozzles accelerates the fuel converting its flow work into kinetic energy. This expansion comes with certain fluid cooling. In the process, there are losses. One can assume that those losses are converted into heat (turbulent kinetic energy is neglected) which increases fuel temperature. Therefore, it is possible to estimate the temperature change as function of the losses, in this case measured with the velocity coefficient, $C_v$ (Payri et al., 2004).

Fig. 1 represents a Diesel nozzle, where the liquid fuel flows from the inlet (1) to the outlet (2). Inlet conditions (pressure, temperature and velocity –which is approximately zero–) are known, whilst at the outlet just the pressure is known (back pressure).

As it is explained later in this section, the enthalpy of the liquid fuel can be expressed as function of two thermodynamic variables such as pressure ($p$) and entropy ($S$), $h_l(p, S)$. In the same way, the entropy is function of pressure ($p$) and temperature ($T$). These relationships can be easily found in the literature (Khasanshin et al., 2003). Therefore, enthalpy and entropy at inlet are known. If the process is assumed to be adiabatic, and also isentropic, the theoretical exit velocity $U_2$ can be calculated using Eq. (1).

$$h_0 = \text{constant} \rightarrow h_{l,2}(p_1, T_1) = h_{l,2}(p_2, T_2) + \frac{U_2^2}{2} \tag{1}$$

For a given value of losses, the effective velocity is calculated from the definition of the velocity coefficient:

$$U_{eff} = C_v \sqrt{\frac{2(p_i - p_b)}{\rho_l}} \tag{2}$$

Eq. (1) can be re-written to obtain the real enthalpy at the outlet section, and from this value, the temperature of the fuel at this section can be obtained.

$$h_{l,2}(p_2, T_2) = h_{l,2}(p_1, T_1) - \frac{U_{eff}^2}{2}$$

The relationship between losses and inlet-outlet temperature difference is shown in Fig. 2 for ECN Spray A injection conditions (see Section 2.4). As previously shown by Theodorakakos et al. (2012), who carried out a similar analysis, despite the fuel depressurization, temperature increases, particularly for the partial needle opening case with low values of $C_v$. In this plot, the value of $C_v = 0$ corresponds to the iso-enthalpy process where the flow work is transformed into thermal energy. And the maximum value of $C_v$ corresponds to the isentropic process, where the flow work is transformed into kinetic energy without losses.

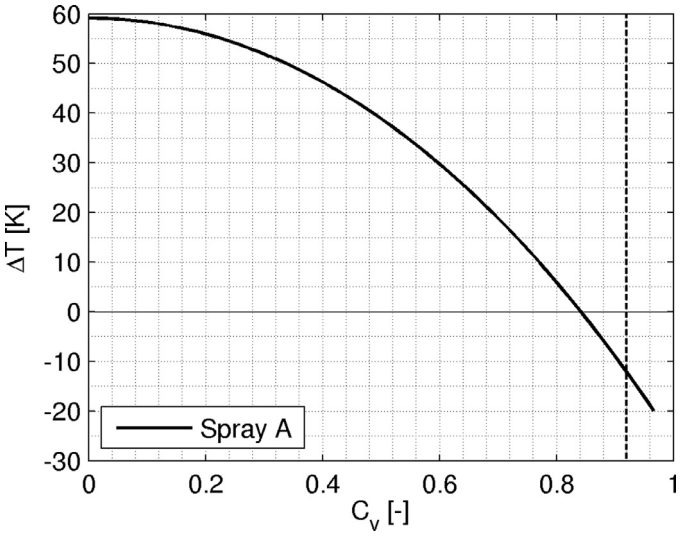


**Fig. 2.** n-Dodecane temperature change as function of the velocity coefficient (losses) for ECN injection conditions, $p_i = 150$MPa and $p_b = 2$MPa.

The experimental steady state value of $C_v = 0.92$ (Kastengren, 2012) is marked in Fig. 2 with a dashed line. For that value, a temperature drop of 12.2 K is expected according to this simplified zero-dimensional model. A difference of 10 K in fuel temperature leads to an approximate difference of 1 mm in liquid length penetration (Siebers, 1999). This may not be negligible and thermodynamic models used in simulations should be able to predict it.

### 2.2. Calculation of mixture enthalpy

Homogeneous Eulerian models are based on the definition of the liquid mass fraction, $Y$ (see Section 2.3). All thermo-physical properties of the mixture can be computed from this quantity. Concretely, the enthalpy of the mixture, $h$, with Eq. (3).

$$dh = Y dh_l + (1 - Y) dh_g \tag{3}$$

The enthalpy is transported following the balance Eq. (4); where $K$ is the mean kinetic energy $K = 1/2\ \boldsymbol{U}^2$; $\kappa_{eff}$ is the effective thermal diffusivity, which takes into account also turbulence; and $\boldsymbol{\tau}\nabla\boldsymbol{U}$ is the viscous dissipation, which may play a significant role inside Diesel nozzles due to large velocity gradients found near orifice walls.

$$\frac{\partial(\rho h)}{\partial t} + \nabla(\rho \boldsymbol{U} h) - \nabla\left(\kappa_{eff} \nabla h\right) = \frac{\partial p}{\partial t} - \frac{\partial K}{\partial t} - \nabla(\rho \boldsymbol{U} K) + \boldsymbol{\tau}\nabla\boldsymbol{U} \tag{4}$$

In order to obtain the temperature from the energy (enthalpy in this case), some homogeneous models (Trask et al., 2012) employ an equivalent specific heat capacity, $C_p = Y C_{p,l} + (1 - Y) C_{p,g}$, where $C_{p,\,l}$ and $C_{p,\,g}$ denote the specific heats at constant pressure of the liquid and gas respectively; which both are assumed to be constant. Then, they use Eq. (5).

$$h = C_p\ T \tag{5}$$

Others (García-Oliver et al., 2013) take into account the variation of the liquid specific heat with the temperature throughout a Rowlinson–Bondi equation (Bondi, 1996). Those approximations work well for simulating sprays, where variations of pressure are small. However, pressure gradients inside Diesel injector nozzles are quite large and change liquid properties, and so the injector performance (Tat and Gerpen, 2003). Therefore, the enthalpy must