THERMAL EFFECTS ON THE DIESEL INJECTOR PERFORMANCE THROUGH ADIABATIC 1D MODELLING. PART I: MODEL DESCRIPTION AND ASSESSMENT OF THE ADIABATIC FLOW HYPOTHESIS

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ABSTRACT

The fuel flow along common-rail injectors is usually treated as isothermal, although the expansions across the injector orifices lead to variations in the fuel temperature that in turn modify the fuel properties influencing injector dynamics. This investigation introduces the hypothesis of adiabatic flow to account for local temperature variations in the computational model of a solenoid injector previously introduced by the authors in its isothermal variant. The main contribution of the study consists on the assessment of the validity of this hypothesis by qualitatively estimating the relative importance of the heat transfer processes during the injection event and in the time lapse among injections. Results of this tentative assessment for engine-like conditions imply that heat transfer is usually still occurring by the time of a new injection, meaning any initial temperature difference among the fuel and the injector wall is not expected to be completely mitigated before each injection event. The magnitude of reduction of this temperature difference depends on the injection frequency through engine speed and load. Anyway, the assumption of adiabatic flow seems to hold once the steady conditions of the injection are reached, meaning that any temperature change predictions considered with the adiabatic hypothesis may be valid as long as a certain temperature change is accounted for at the injector inlet. In a second part of the paper, the capabilities of this new model are validated against experimental data, allowing the use of the model to explore the influence of the thermal effects on the injection event.

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KEYWORDS

diesel, injection, computational, 1D modelling, fuel temperature, adiabatic flow

LIST OF NOTATION

A_o	outlet area
Ad	adiabatic number
Bs	isentropic bulk modulus
C_a	area coefficient
C_d	discharge coefficient
C_i	coefficients for the fuel properties correlations
C_v	velocity coefficient
c_p	fluid heat capacity at constant pressure
D	orifice diameter
e	specific internal energy
F_i	coefficients for the fuel dynamic viscosity expression
Fo	Fourier number
f	fuel properties correlation function
h	specific enthalpy
h_t	stagnation specific enthalpy
J_0	Bessel function of first kind of order zero
K_{Ad}	constant for the transient radial heat conduction in an injector duct
k_f	fuel thermal conductivity
L	orifice length
m_i	total mass injected per cycle
$\dot{m}_f$	fuel mass flow
$\dot{m}_{th}$	theoretical mass flow
Nu	Nusselt number
n_e	engine speed
p	pressure
p_0	reference pressure

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p_b injector backpressure

p_{dw} downstream pressure

p_i injection pressure

p_{up} upstream pressure

Pr Prandtl number

r radial coordinate

r_0 cylinder diameter

Re Reynolds number

St Stanton number

T temperature

T_0 reference temperature

T_{dw} downstream temperature

T_i fuel temperature at the injector inlet

T_{up} upstream temperature

T_w wall temperature

t time

t_k characteristic time of heat transfer due to conduction

t_{res} fuel residence time in the injector

u flow velocity

u_{th} theoretical flow velocity

v_f fuel specific volume

GREEK SYMBOLS

α fuel thermal diffusivity

β volumetric thermal expansion coefficient

$\varepsilon_{\Delta T}$ percentual deviation among the theoretical temperature change across an orifice and the experimental one

θ temperature difference among the fluid and the wall

θ_i initial temperature difference among the fluid and the wall

λ_n eigenvalues for the problem of transient radial heating by conduction through an infinite cylinder

μ_f fuel absolute viscosity

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$\mu_{f,0}$ fuel absolute viscosity at atmospheric pressure

ρ_f fuel density

τ_{Ad} ratio among residence time and conduction characteristic time

ABBREVIATIONS

BMEP Brake Mean Effective Pressure

DI Direct Injection

ET Energizing Time

OA control volume outlet orifice

OZ control volume inlet orifice

ROI Rate of Injection

1. INTRODUCTION

The mixing process is key to the performance of direct-injection engines, due to its direct influence on the combustion phenomenon and therefore on the engine performance and the formation of emissions [1]. Some researchers have then focused on the fuel injection system, analysing the role of the injector topology [2,3], the nozzle geometrical features [4–8] and the operating conditions [9–11] on the injector internal flow, spray formation and fuel-air mixing processes. In the common-rail diesel engines community the gathered knowledge led, for instance, to a generalized progressive increase in the fuel injection pressure.

Most studies in the literature focus on the nozzle internal flow and spray development, whereas the injector upstream flow has not been paid so much attention. Given that the injection event is a transient process and considering the high flow velocities through the injector and the small dimensions of its ducts, experimental observations in this regard present several difficulties. Nevertheless, the internal flow through the injector rules its own dynamics, especially in non-direct acting injectors [12]. This justifies the need for computational models as an approach to carry out research in the topic.

In this regard, 1D modelling has been widely employed, allowing the study of the whole injection process at a low computational cost. Most investigations take the assumption that the flow along the injector is isothermal [13–15], including past investigations carried out by the authors [16–18]. However, the ultra-high pressures reached by current diesel injection systems [19] (up to 300 MPa) imply that the thermal effects taking place within the injector become relevant. Thus, there may be important fuel temperature variations along the injector due to friction heating or due to the important fuel depressurization that takes place at the injector flow restrictions (i.e. control volume orifices or most

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importantly the nozzle [20–22]). The local changes in fuel temperature and pressure influence the fuel thermophysical properties that are a function of these thermodynamic variables, such as the speed of sound, density or viscosity [23]. These properties, in turn, may impact the injected mass flow rate and spray development [24,25].

Due to the aforementioned reasons, several modellers started accounting for local temperature variations in their models, considering the injector internal flow to be adiabatic [26,27], even though they covered a short range of injector operating conditions. CFD modellers also explored the thermal effects in diesel injectors. Theodorakakos et al. [20] focused on the nozzle flow, reporting that the flow could be either heated or subcooled at the nozzle outlet depending on the nozzle discharge coefficient and the flow regime induced by the injector operating conditions. On the other hand, Yu et al. [28] extended the modelling of thermal effects to diesel sprays injected through convergent-divergent nozzles even in cavitating conditions.

Despite these efforts, few attempts have been made to contrast the calculated local temperature variations along the injector against results obtained experimentally in order to assess the hypothesis of adiabatic flow. Even though this hypothesis is undoubtedly more accurate than the isothermal assumption, it may not hold if the fuel is slowly injected into the combustion chamber letting heat transfer to take place with the surrounding atmosphere across the injector walls. The authors tried to qualitatively assess the validity of the adiabatic flow assumption through experimental measurements on diesel injector calibrated orifices [29]. These orifices were submitted to controlled pressure drops and the measured temperature variations were compared to the theoretical ones given by isenthalpic expansions. It was found that the flow through a diesel injector orifice could be regarded to as adiabatic for the operating conditions that led to low fuel residence times within it (most conditions found during the steady-state stage of the injection were found to fall in this category). Later, the authors implemented the adiabatic flow hypothesis in a CFD code and modelled these same calibrated orifices [22], reporting good agreement among the temperature variations predicted by the model and the experimental results. Then, the model was applied to the flow through diesel injection nozzles. Nevertheless, these investigations aimed at the study of a single injection event, without taking into account the time lapse among injections. Depending on the injection frequency induced by the operating conditions, heat transfer may take place during this period modifying the inlet fuel temperature prior to each injection.

This study deals with the implementation of the adiabatic flow assumption in a diesel injector 1D model, so that the model accounts for local temperature variations along the injector. The present paper describes the theoretical foundation for this implementation on an existing 1D model of a Bosch CRI 2.20 diesel injector previously presented by the authors in its isothermal variant [18]. Most importantly, a discussion on the validity of the adiabatic hypothesis for the injector

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internal flow is presented. This consists on estimating, from a qualitative point of view, which operating conditions (both for a real engine and for the injection rate and visualization measurements typically performed for the study of the injection process in laboratory environments) could lead to accepting or refusing the adiabatic flow hypothesis with more confidence.

A second part of the paper [30] shows a thorough practical validation of the adiabatic model covering the predictions of mass flow rate and temperature change through the orifices, the total injected mass per cycle and the pressure wave transmission along the injector. It then analyses the results purely provided by the validated model. This includes the fuel temperature variations along the injector ducts and orifices and the extent of their influence on the injection phenomenon, including the analysis of the impact of the fuel temperature on the effectiveness to split injections when using multiple injection strategies.

2. DESCRIPTION OF THE INJECTOR 1D MODEL

The computational investigation has been carried out based on a Bosch CRI 2.20 injector. This common-rail injector is driven by a solenoid valve and is ballistic (which means that the mechanical needle lift limit is not reached during the usual operation of the injector). As it has been said, the assumption of adiabatic flow has been implemented in a 1D model of this injector that was already introduced by the authors in previous investigations [18]. In the previous version of the model, the fuel flow was assumed to be isothermal. Therefore, the fuel temperature was considered to be constant along the injector ducts during the complete injection event. This meant that all the significant fuel properties (i.e. speed of sound, density and viscosity) remained unaltered both spatially and temporally. In the present investigation, local variations of the fuel temperature and its associated properties are considered during the injection by means of the adiabatic flow assumption introduced in the model. This model was implemented in AMESim, a commercial platform for multi-domain simulation [31].

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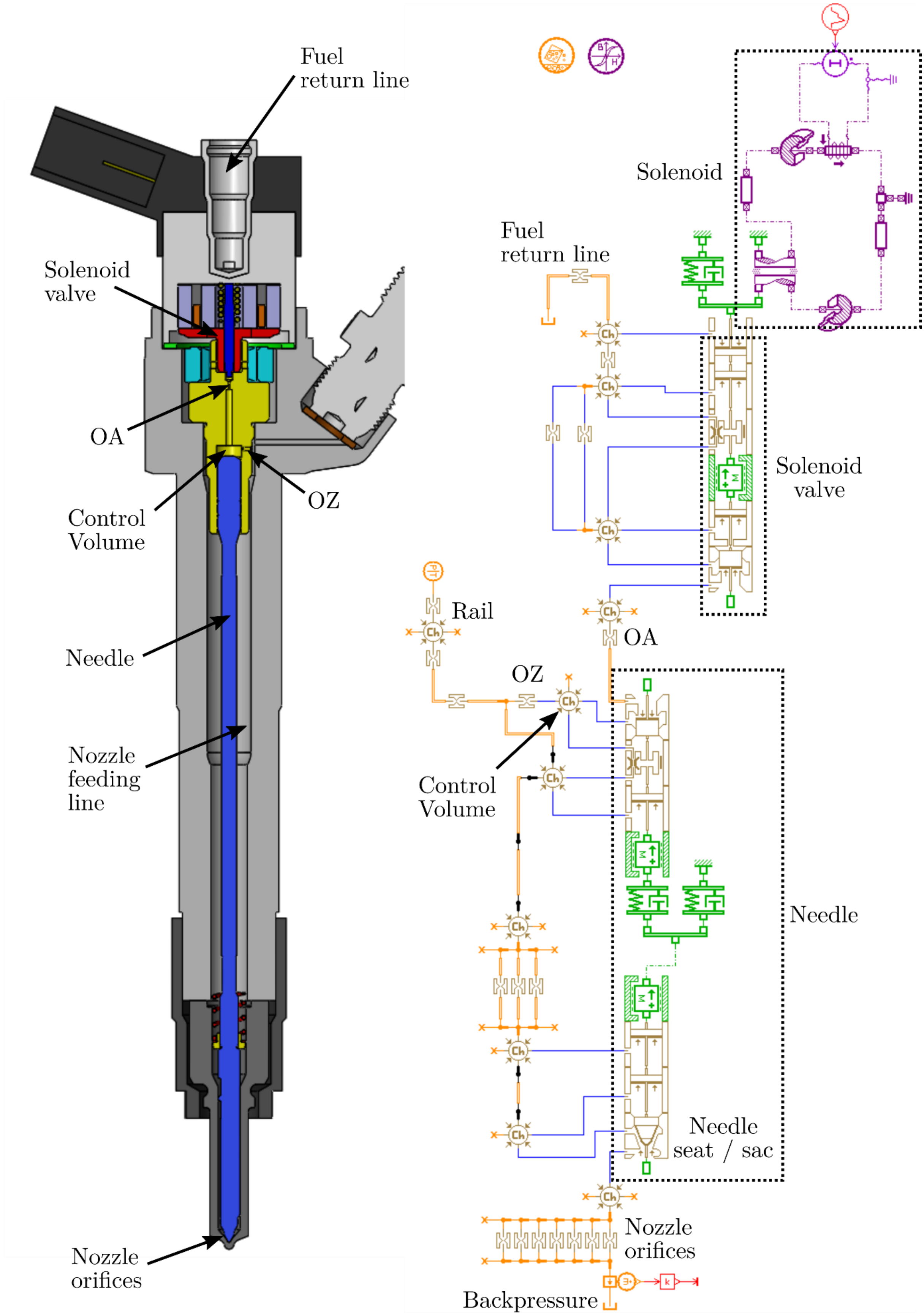

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169 Figure 1. Bosch CRI 2.20 injector diagram and AMESim model sketch.

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A complete sketch of the internal parts of the injector is shown in **Figure 1**, together with their equivalence in the adiabatic AMESim model. When the injector is energized, the fuel flowing along the injector is submitted to several hydraulic restrictions on its path from the injector inlet to any possible outlet:

- Part of the flow from the injector inlet flows directly towards the nozzle orifices through the *nozzle feeding line*.
- Part of the flow from the injector inlet enters the control volume through the OZ orifice (also known as *control volume inlet orifice*). This fuel flow can in turn follow 2 different paths: on the one hand, part of the fuel leaks through the gap among the needle and the control volume piece, lubricating the needle and feeding the nozzle orifices; on the other hand, part of the fuel flows through the OA orifice (also known as *control volume outlet orifice*) and leaves the injector through the *fuel return line*.

The internal dimensions and relevant hydraulic parameters were introduced to the model after a thorough geometrical and hydraulic characterization. The geometrical characterization included a complete metrology of the internal parts of the injector carried out according to the techniques described in [32]. On the other hand, the hydraulic characterization consisted on experimentally determining the evolution of the discharge coefficient with *Re* for the most relevant flow restrictions. This included the characterization of the cavitation phenomenon, found for the control volume outlet orifice (OA). For more details on this characterization and the model parameters, please refer to the already mentioned previous work by the authors [18]. The main features of the model are summarized as follows:

- The elastic deformations of the mechanical movable pieces are modelled, as a key factor to properly predict the injection delay (once the injector is energized, the elastic deformation of the needle must be recovered prior to the new injection event).
- Pressure wave transmission is modelled through the injector internal lines. To do so, the deformation of the pieces shaping the injector ducts is considered by means of the material modulus of compressibility. These factors, together with an accurate description of the fuel speed of sound evolution with the pressure and temperature, are mandatory to properly predict the time among injections when using split injection strategies.
- Friction among mechanical elements is accounted for, together with the friction among the fuel and the mechanical elements. Fuel leakage among elements is also considered.
- The discharge coefficient of the orifices is computed at each time step as a function of the local *Re* established [33]. The effect of cavitation in the discharge coefficient behaviour is also considered. Possible reductions in area in the orifices due to the lift of mechanical valves are taken into account as well.
- As it has been stated, the relevant fuel properties depend on the local temperature (thanks to the assumption of

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adiabatic flow) and pressure and are also varied temporally.

It is important to remark that the model accounts for the hydraulic, mechanical and electromagnetic phenomena relevant in the actuation of this solenoid diesel injector, but the local temperature changes are only included in the hydraulic components in what constitutes the assumption of adiabatic fuel flow here assessed. In Part II, these local changes in fuel temperature are shown to present an important influence on the hydraulics dominating the dynamics of the injector needle [30]. The assumption of adiabatic flow, on the other hand, implies that the temperature of the mechanical and electromagnetic metallic components is assumed to remain constant. This could have relevant implications in the case of the needle. As far as the solenoid valve is concerned, it only lifts to a maximum of around 20 µm [18], which is achieved almost instantaneously once the solenoid is energized (and almost instantaneously closed back towards its seat once de-energized). The evolution of this valve in different families of solenoid injectors shows that its influence on the injection rate curve from one generation to another is limited [34], making it safe to assume that any variations of the electromagnetic properties of a given valve related to its possible metal temperature changes are small compared to the influence of the hydraulics in the control volume piloting the needle.

3. IMPLEMENTATION OF THE ADIABATIC FLOW ASSUMPTION

As stated previously, the implemented model accounts for local variations in fuel temperature and pressure during the injection event. All the fuel temperature changes along the injector are computed assuming adiabatic flow, with no interaction with the surroundings in terms of heat transfer.

In absence of external work, the first law of thermodynamics establishes that the total enthalpy or stagnation enthalpy (h_t) of a fluid is conserved during an adiabatic process, as shown in Eq. (1):

$$\Delta h_t = \Delta\left(h + \frac{u^2}{2}\right) = 0 \tag{1}$$

where h is the fluid specific enthalpy and u its velocity.

Along any hydraulic component (such as an orifice submitted to a pressure drop or a hydraulic line with an associated loss due to friction) within the injector, Eq. (1) may be rewritten as:

$$h(T_{dw}, p_{dw}) = h(T_{up}, p_{up}) - \frac{1}{2}\Delta u^2 \tag{2}$$

Used by the model, Eq. (2) implies that the temperature downstream (i.e. at the outlet) of any hydraulic component may be computed if the upstream temperature (i.e. at the inlet), the pressure drop and the velocity change across the component are known, as long as the fuel specific enthalpy has been expressed as a function of the pressure and temperature (the

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particular dependency used in this investigation is introduced in Section 3.1). This includes the flow through the small calibrated orifices (OZ, OA and nozzle orifices) and the flow through long ducts along which pressure losses are expected (for instance, the nozzle feeding line).

It is interesting to analyse Eq. (2) in the view of the important restrictions to the flow along the injector, since this will allow understanding in Section 3.1 which locations and under which circumstances the fuel may heat up or cool down:

- In the case of the particular components where the velocity change is not relevant (as it happens along the nozzle feeding line), Eq. (2) leads to the conservation of specific enthalpy in order to calculate the temperature change through them:

$$h(T_{dw}, p_{dw}) \approx h(T_{up}, p_{up}) \tag{3}$$

- In the case of other hydraulic components, the established pressure drops are very high (such as the nozzle orifices and the OA orifice). Thus, the velocity upstream of them is small compared to the downstream velocity. Therefore, Eq. (2) can be approximated as follows:

$$h(T_{dw}, p_{dw}) \approx h(T_{up}, p_{up}) - \frac{1}{2}u_{dw}^2 \tag{4}$$

It is then possible to introduce a velocity coefficient [35] similar to the one defined by Siebers [36]:

$$C_v = \frac{u}{u_{th}} \tag{5}$$

where u_{th} is the theoretical velocity according to Bernoulli:

$$u_{th} = \sqrt{\frac{2(p_{up} - p_{dw})}{\rho_f}} \tag{6}$$

According to these definitions, Eq. (4) can be rewritten to yield the downstream temperature in terms of the pressure drop and the velocity coefficient:

$$h(T_{dw}, p_{dw}) \approx h(T_{up}, p_{up}) - C_v^2 \frac{(p_{up} - p_{dw})}{\rho_f} \tag{7}$$

Furthermore, the velocity coefficient can be expressed in terms of the discharge coefficient and the area coefficient [35]:

$$C_d = C_v \cdot C_a \tag{8}$$

where the area coefficient is defined in an analogous manner to the velocity coefficient so that it takes into account the area contraction in an orifice (relevant when cavitation appears), whereas the discharge coefficient is defined as the ratio among the actual mass flow rate through an orifice and the theoretical one, as shown by

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Eq. (9):

$$C_d = \frac{\dot{m}_f}{A_0\sqrt{2\,\rho_f\,(p_{up} - p_{dw})}} \tag{9}$$

Therefore, Eq. (7) can also be expressed in terms of the discharge coefficient and the area coefficient:

$$h(T_{dw}, p_{dw}) \approx h(T_{up}, p_{up}) - \left(\frac{C_d}{C_a}\right)^2 \frac{(p_{up} - p_{dw})}{\rho_f} \tag{10}$$

In this investigation, the fuel density has been taken at a mean pressure and mean temperature among the ones found at the inlet and the outlet of the corresponding orifice. It is important to note that Eq. (1), upon which the adiabatic assumption has been implemented, applies with no work exchange among the hydraulic restriction and other components of the system. This statement is acceptable for most cases, but it may not be true in the case of the nozzle orifices for the transient stages. For the low values of needle lift of these stages, the needle might exchange mechanical work with the nozzle. Nevertheless, the interaction among needle and nozzle flow is only deemed to be relevant for values of needle lift lower than 75 µm [37], which are usually restricted to a time in the order of 0.1 ms for both the opening and the closing stages [18]. Thus, this phenomenon is restricted during the injection event and the assumption is not expected to importantly influence the results. In any case, the results reported in Part II [30] are restricted to the steady-state stage of the injection.

3.1. Fuel model

The equations derived in Section 3 highlight the need to express the fuel specific enthalpy as a function of temperature and pressure in order to compute the temperature variations across an injector hydraulic component with the adiabatic flow assumption. The fuel specific enthalpy may be related to its specific internal energy according to Eq. (11):

$$h = e + \frac{p}{\rho_f} \tag{11}$$

Let us now evaluate the derivative of the fuel specific enthalpy as a function of temperature and pressure:

$$\mathrm{d}h = \left(\frac{\partial h}{\partial T}\right)_p \mathrm{d}T + \left(\frac{\partial h}{\partial p}\right)_T \mathrm{d}p = c_p\,\mathrm{d}T + \frac{1}{\rho_f}\left[1 + \frac{T}{\rho_f}\left(\frac{\partial \rho_f}{\partial T}\right)_p\right]\mathrm{d}p \tag{12}$$

Introducing the volumetric thermal expansion coefficient:

$$\beta = \frac{1}{v_f}\left(\frac{\partial v_f}{\partial T}\right)_p = -\frac{1}{\rho_f}\left(\frac{\partial \rho_f}{\partial T}\right)_p \tag{13}$$

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Eq. (12) can then be written in terms of β:

$$\mathrm{d}h = c_p\,\mathrm{d}T + \frac{(1-\beta\,T)}{\rho_f}\,\mathrm{d}p \tag{14}$$

Please note that the second term of the right hand side of Eq. (14) would be zero for an ideal gas (for which $\beta \cdot T = 1$), but cannot be neglected for a liquid. Eq. (14) can be integrated if the fuel properties evolution with the pressure and temperature is known.

A winter diesel fuel has been considered for the computational study, replicating the fuel utilized in the experiments used for the model validation later explained in Section 5 [38]. Some of its properties (namely speed of sound, bulk modulus and density) were experimentally characterized by the authors for the range of pressures and temperatures of interest for common-rail systems [23]. As far as the heat capacity at constant pressure (c_p) is concerned, its evolution with the pressure was assimilated to the ISO 4113 test fluid one [39], whereas its dependency with the temperature was estimated from pure alkanes [40,41]. This assumption is not deemed to introduce a noticeable error, given the low variation of c_p for diesel fuels on the relevant range of temperatures [42,43]. It is possible to use these data in Eq. (14): specific enthalpy variations (Δh) can be determined among conditions by taking small variations of temperature (ΔT) and pressure (Δp) instead of differentials. This allows to build the *specific enthalpy map* of the fuel, as shown in **Figure 2**.

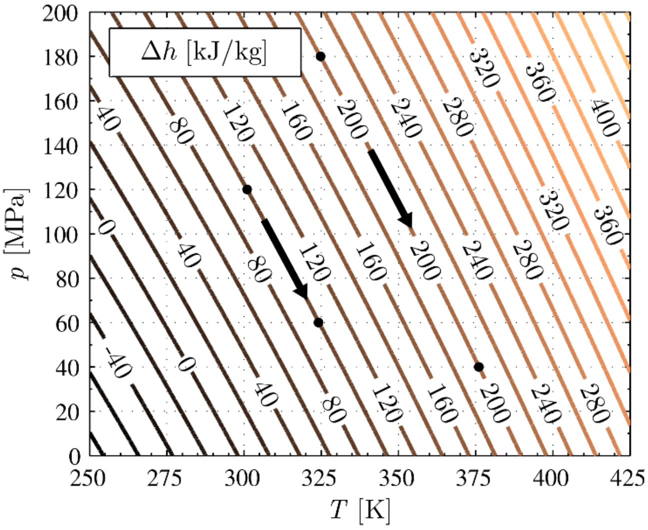

Figure 2. Lines of iso-specific enthalpy as a function of temperature and pressure for the winter diesel fuel. Reference conditions for h = 0 J/kg: T_0 = 298 K; p_0 = 0.1 MPa.

As it has been commented, in the case of the hydraulic components of the injector through which the flow velocity change is not relevant, Eq. (3) practically leads to the conservation of specific enthalpy ($\Delta h \approx 0$). Thus, across such a component (for instance, along the nozzle feeding line) the flow could be regarded to as isenthalpic and the temperature change due

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to a pressure loss would be directly given by **Figure 2**. In these cases, **Figure 2** illustrates that the fuel would warm upon expansion, as opposed to what happens to an ideal gas, due to the existence of the second term of right hand side of Eq. (14). Hence, for example, an isenthalpic expansion from 120 MPa to 60 MPa with an upstream temperature of 300 K would lead to a downstream temperature of 325 K (ΔT = 25 K), or an isenthalpic expansion from 180 MPa to 40 MPa with an upstream temperature of 325 K would lead to a downstream temperature of 375 K (ΔT = 50 K). This makes sense in the view of Eq. (11): under the assumption of adiabatic flow, the heating induced by viscous dissipation or a fuel depressurization is assumed to remain within the fluid, contributing to increase its internal energy. As a consequence, in the case of an isenthalpic expansion, the fluid temperature rises as it expands. In addition, the higher the fuel temperature is, the more accused the slope of the iso-enthalpy lines in **Figure 2**. This means that, for isenthalpic expansions with a given pressure drop, higher temperature rises will be achieved with a low upstream temperature.

In the general case for which the stagnation enthalpy is preserved, the temperature change also depends on the flow velocity change along the process. As given by Eq. (2), an increase in fluid velocity due to a depressurization (for instance across the nozzle orifices) would lead to a reduction in specific enthalpy downstream of the orifice. According to **Figure 2**, this reduction in specific enthalpy would imply a reduction in fluid temperature. Therefore, the net change in fluid temperature given by Eq. (2) could be positive or negative depending on the magnitude of the velocity change across the restriction. In the case of the nozzle orifices and the OA orifice, as developed in Eq. (10), this dependency of the fuel temperature on the velocity increase can also be expressed in terms of the pressure drop and the discharge coefficient. This reasoning is consistent with 3D CFD calculations by the authors in a previous work [22] concerning the flow along the orifices of a nozzle with a high discharge coefficient. In this work, a temperature increase was observed at the orifice walls (shear, low-velocity zones) with a temperature decrease at the centre of the orifice. Overall, the fuel bulk temperature decreased along the orifice, as the 1D computation would suggest through Eq. (9) for an orifice with a high discharge coefficient submitted to an important pressure drop.

The evolution of each of the fuel properties with the temperature and pressure was provided to AMESim, so that their local variations are considered. Specifically, the fuel density, speed of sound and bulk modulus dependency with temperature and pressure were introduced in the form of the polynomial expressions given by Eq. (15), whose coefficients were fitted on the basis of experimental data, as shown in a previous work by the authors [23]:

$$f(T,p) = C_0 + C_1(T - T_0) + C_2(p - p_0) + C_3(T - T_0)(p - p_0) + C_4(T - T_0)^2 + C_5(p - p_0)^2 \qquad (15)$$

where f denotes either the fuel density, speed of sound or isentropic bulk modulus.

As far as the fuel viscosity is concerned, experimental measurements were only available at atmospheric pressure [22].

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Thus, dynamic viscosity was introduced as a function of the temperature according to the correlation shown in Eq. (16):

$$\mu_{f,0}(T) = F_1 e^{F_2(T-T_0)} \tag{16}$$

The evolution of the dynamic viscosity with the pressure was estimated according to the Kouzel expression [44], reproduced in Eq. (17):

$$\mu_f(p,T) = \frac{\mu_{f,0}}{1000} \cdot 10^{\left[\left(\frac{p-0.10133}{1000}\right)\left(-1.48+5.86\mu_{f,0}^{0.181}\right)\right]} \tag{17}$$

The coefficients C_i or F_i used for these correlations are summarized for each property in **Table 1**, together with the R^2 values reported.

Property	C_0 [-]	C_1 [-]	C_2 [-]	C_3 [-]	C_4 [-]	C_5 [-]	F_1 [-]	F_2 [-]	R^2 [-]
a_f [m/s]	1350.59	-3.1485	4.4928	0.0074	0	-0.0070	-	-	0.9996
ρ_f [kg/m3]	826.54	-1.021	0.603	0.0014	0.00125	-0.00083	-	-	0.9995
B_s [MPa]	$1.489\cdot10^9$	$-8.766\cdot10^6$	$1.199\cdot10^7$	0	0	-7461.3	-	-	0.9998
$\mu_{f,0}$ [Pa·s]	-	-	-	-	-	-	3.2158	-0.0263	0.9906

Table 1. Coefficients for the fuel properties correlations of Eq. (15). Each fuel property is given in the referred units introducing T in K and p in MPa. Reference conditions: T_0 = 298 K; p_0 = 0.1 MPa.

4. DISCUSSION: ON THE VALIDITY OF THE ASSUMPTION OF ADIABATIC FLOW

Before applying the adiabatic assumption to the flow along the injector, it is key to assess the limits of the extension of this hypothesis. In the frame of the present work, this assessment can ensure the reliability of the simulations performed by the computational model, given that local changes in temperature and pressure are taken into account. More generally, this assessment can be a contribution of interest to both experimentalists and modellers in the field that usually require the injector outlet temperature as an input or boundary condition to their studies. It is thus interesting to evaluate the validity of this hypothesis considering the injector flow in both the typical engine-like operating conditions and the usual operating conditions reproduced in laboratory environments to conduct measurements such as the rate of injection curve or spray visualization by means of experimental test rigs. These kinds of measurements have become usual to evaluate the injector performance [45–48] and the spray development and mixing characteristics [49]. The rate of injection measurements have indeed been used to validate the computational model of this investigation in the second part of the paper [30]. In any case, it must be pointed out that the assessment here presented has been carried out in qualitative terms, comparing the relative importance among phenomena and providing with orders of magnitude for their relevance.

The qualitative evaluation of the validity of this assumption has been divided into the analysis during the injection event itself (when the fuel is flowing, while the injector is energized) and the analysis of relevance of heat transfer to or from

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the fuel during the time among successive injections (where the fuel is at rest).

4.1. During the injection

The validity of the hypothesis while the fuel is effectively flowing through the most important injector internal orifices was already qualitatively assessed experimentally by the authors [29], considering that the most important thermal effects will take place in these restrictions given the pressure drops induced by them. The most relevant features of that work are next summarized for continuity in the discussion presented in the present paper. Specifically, continuous flow was established along the control volume orifices of the Bosch CRI 2.20 injector and a Denso G4S injector. In the facility, a given single orifice could be isolated, with its inlet connected to an accumulating rail by a high-pressure hydraulic line and its outlet connected to a discharge volume through another high-pressure line with identical diameter. The flow could then be submitted to different controlled pressure drops across the orifices. Then, the temperature was measured at two different locations (one in the hydraulic line 16 cm upstream of the orifice and the other in the hydraulic line 16 cm downstream of the orifice) where the controlled flow was assumed to be developed and attained a similar velocity. Under these conditions, according to Eq. (2), the flow is forced to be virtually isenthalpic (Eq. (3)). This implies that the theoretical fuel temperature change across the orifice is given by an isenthalpic evolution as the ones depicted in **Figure 2** (always leading to heating among these 2 locations). Therefore, the temperature changes measured for different operating conditions could be compared to the theoretical ones predicted by Eq. (2). The deviations could then be attributed to heat exchange with the surroundings. The analysis of these deviations gave a hint on the conditions that led to heat exchange becoming more or less relevant to the problem.

In this regard, the dimensionless parameter defined in Eq. (18) was theoretically derived to account for all the variables that influenced the proneness of the flow through an orifice to exchange heat with the surroundings:

$$Ad = \frac{1}{4}\frac{D}{L}St^{-1} = \frac{1}{4}\frac{D}{L}\frac{Pr\,Re}{Nu} \tag{18}$$

Please note that *Ad* is a corrected *St* number with a fourth of the *D*/*L* ratio of dimensions. For details on this derivation and the significance of the *D*/*L* factor, the reader may refer to the complete previous study [29].

The different diameter of the internal orifices and the different conditions (i.e. pressure drops) tested were associated to several values of *Ad*. **Figure 3** shows how the percentual deviation ($\varepsilon_{\Delta T}$) among the theoretical temperature change and the experimental one were found to evolve with the *Ad* parameter for the non-cavitating conditions tested (the flow through the orifices of a Denso G4S injector was also studied in order to gather further data). It was observed that most points corresponding to the same *Ad* value collapse to similar values of $\varepsilon_{\Delta T}$ regardless the combination of orifice and

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condition tested. Thus, *Ad* can be considered as a good indicator of the proneness of the flow through an orifice to exchange heat with the surroundings.

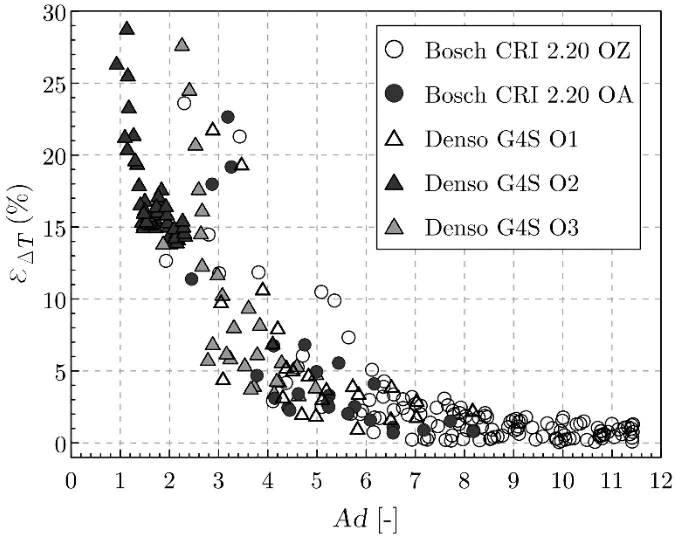

Figure 3. Percentual deviation $\varepsilon_{\Delta T}$ among the theoretical temperature change and the experimentally determined one for the *Ad* values induced by each non-cavitating condition tested for each orifice (data for the Bosch CRI 2.20 and the Denso G4S injector orifices). Data from [29].

The main finding of the investigation was that low values of *Ad* lead to a high departure to the temperature change predicted by the theoretical adiabatic expansion, whereas these deviations are bounded for high *Ad* values: the deviations are generally lower than 10% for *Ad* > 4 and lower than 5% for *Ad* > 6. It may then be stated that:

- For a given orifice, the tendency of the flow across it to exchange heat with the surroundings depends on the established value of *Re*/*Nu* and the flow regime. Empirical correlations in the literature relate *Nu* to *Re* (among other factors). These correlations depend on the flow regime, but generally establish $Nu = a \cdot Re^b$, where $b < 1$ [50,51]. Hence, *Ad* is higher the greater *Re*. The significance of this fact may be analysed as follows. Low values of *Re* yielding low values of *Ad* are associated to low flow velocities, implying high residence times for the fuel in the ducts. Hence, heat transfer is allowed to take place. Consequently, low pressure drops through an orifice induce a higher interaction with the ambient than high pressure drops. On the contrary, high flow velocities are induced for high pressure drops leading to high values of *Re* and *Ad*. In these conditions, it is possible to neglect the heat transfer and to assume that all the heat generated due to the fluid expansion remains within the fluid, being invested in raising its temperature.
- For a given operating condition (associated to a particular pressure drop across the orifices), the orifice diameter also influences the proneness of the flow across it to exchange heat, not only due to its impact through *Re* and *Nu*. The term *D/L* is relevant in *Ad* since, for a particular channel length, a lower diameter implies greater wall

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effects. On the one hand, this is explained by a higher proportion of the flow being in touch with the orifice surface, interacting with the surroundings. On the other hand, the boundary layer is also proportionally thicker, meaning that its viscous effects will reduce the flow velocity close to the orifice walls, increasing the fuel residence times and favouring heat transfer.

In consequence, the flow established through the injector orifices with operating conditions leading to high average values of *Ad* behaves in a nearly adiabatic manner. In this regard, it was found that the diameter of the Bosch CRI 2.20 control volume orifices was large enough to obstruct the heat transfer process during a given expansion, unless the pressure drop was too low. It was checked that pressure drops in the order of the ones imposed during the steady stage of the injection lead to large enough values of *Ad* [1]. During the transient stages of opening and closing this assumption could not be true, although the characteristic times of these stages are way lower than the time lapse among injections dealt with in the next subsection.

4.2. During the time lapse among injections

4.2.1. Theoretical approach

Both in real engine-like operating conditions and controlled experimental environments, an important temperature difference may be established among the fuel (stored in a tank far from the injector) and the injector walls (located close to the engine head, significantly warmer during the engine operation). Injection in real engine conditions, with fuel being injected into the cylinder in a pulsed manner and remaining at rest during the lapse among injections, is a transient phenomenon. This transiency, together with the amount of varied operating conditions that may be achieved during an engine run, hinders the analysis. The approach taken in this study to qualitatively assess whether the heat transfer is relevant is to compare the fuel residence time in the injector (t_{res}) to a characteristic time of heat transfer due to conduction from the injector walls to the fuel (t_k). Hence, a dimensionless parameter τ_{Ad} has been defined as the ratio among times, as given by Eq. (19):

$$\tau_{Ad} = \frac{t_{res}}{t_k} \tag{19}$$

The parameter τ_{Ad} allows evaluating the importance of heating among injections to establish the relative importance of the thermal effects during the injection, so that:

[1]As an example, Part II [30] shows that for a fairly low rail pressure of 40 MPa (a limiting case in terms of absolute pressure drop along the orifices, since the injector does not open properly for rail pressures below 30 MPa) the pressure drop along the Bosch CRI 2.20 OZ orifice is around 12 MPa. The associated *Ad* established along the OZ orifice is about 6.2.

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- If τ_{Ad} << 1, the fuel residence time in the injector is way lower than the characteristic time of heat transfer and the fuel does not significantly modify its temperature during the time lapse among injections. Once an injection starts, the fuel will modify its temperature due to its internal evolution along the injector (i.e. expansions across the orifices) or will exchange heat with the ambient depending on the related *Ad* value, as analysed earlier. Anyway, low values of *Ad* are not expected if $t_{res} << t_k$, since low fuel residence times imply either high injection frequencies (limited to 4000 rpm in diesel engines) or large pressure gradients that would lead to high *Ad* numbers by definition. Therefore, the assumption of adiabatic flow does not seem to be compromised at any stage for low values of τ_{Ad}.
- If $\tau_{Ad} \approx 1$, the fuel residence times are similar to the characteristic time of heat transfer and the fuel importantly changes its temperature during the lapse among injections. Hence, the temperature of the fuel inside the injector has enough time to get closer to the injection wall temperature prior to each injection event. At the same time, this means that the temperature difference between the wall and the fluid is minimized by the time an injection starts. For this reason, the ability of the fuel to modify its temperature upon its internal expansions (retaining the heat within itself in what would constitute an adiabatic process) or to transfer this heat to the ambient can be assumed to depend on the related values of the *Ad* number.
- If τ_{Ad} >> 1, the fuel residence times are much higher than the characteristic times of heat transfer. Thus, by the time of a new injection, the heat transfer process will be completed and the fuel temperature will already coincide with the injector wall temperature at all locations. During the injection itself, the assumption of adiabatic flow still depends on the *Ad* number: please note that the situation of $t_{res} >> t_k$ could be generated either by very low injection frequencies (not usual in diesel engines) or by small pressure gradients piloting the injector. On the one hand, high *Ad* values could still be reached during the injection (for instance, for high values of rail pressure) regardless how low the injection frequency was. This fact, together with the non-existent temperature difference among the fuel and the wall by the time of a new injection would still support the assumption of adiabatic flow during the injection. On the other hand, as seen in Section 4.1, the case of low values of *Ad* could lead to the fuel not being able to retain the temperature changes produced by its internal evolution through the hydraulic restrictions even though the temperature difference between the wall and the fluid upstream of the restriction was negligible. In this latter case, the assumption of adiabatic flow during the injection would no longer be valid.

The characteristic time of heat transfer due to conduction from the injector walls (t_k) has been estimated considering the simplified problem of transient radial conduction in an infinitely long cylinder as the one shown in **Figure 4**, which will

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then be assimilated to the injector nozzle feeding line (since it is the most relevant line in terms of fuel volume and where the effects of heat transfer both among injections and during the injection itself, if any, would be more noticeable). Even though the problem is not generic and the geometry of the infinitely long cylinder importantly departs from the reality present in the nozzle feeding line (recall Figure **1**) of the Bosch CRI 2.20 diesel injector, this provides an analytical solution to be used as a reference. Additionally, the nozzle feeding line of most commercial solenoid-operated injectors is a cylindrical duct (2nd generation Bosch solenoid injectors [52] prior to the present Bosch CRI 2.20, Denso [53], Delphi [54], etc.). Thus, this analysis is assumed to give a representative order of magnitude of the heat transfer from the injector walls in the time lapse among injections that could be used for qualitative comparison with the fuel residence times in the injector. In any case, the impact of assuming a cylindrical duct instead of an annular one on the calculated values of t_k and τ_{Ad} is analysed in Section 4.2.2 through the results of a transient heat transfer simulation.

Assuming the cylindrical shape, the fluid temperature at the centre of the cylinder initially takes a value T_i, whereas the wall remains at its own temperature T_w. The fluid is considered to be at rest so that convection is negligible. The assumption of infinite length ensures one-dimensional conduction in the radial direction. Once the fluid temperature is let to evolve freely, it will modify its value $T(r)$ along the cylinder radial coordinate r, tending to approach the wall temperature. These hypotheses conform the scenario in which heat transfer takes place in an easier way (any heat flux among the fluid and the walls is only pointed in the radial direction), thus leading to the minimum possible characteristic times for heat transfer, t_k. This results in a conservative estimation of τ_{Ad}.

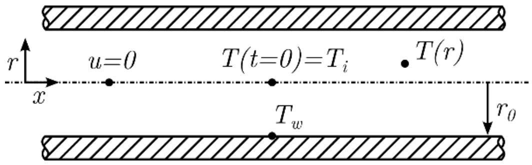

Figure 4. Scheme of an infinitely long cylinder containing fuel at rest.

Before an injection event, the fuel (which entered the injector at a certain temperature) remains at rest and is able to exchange heat with the injector walls (at a different temperature). For any radial position, the temperature difference may be defined in Eq. (20):

$$\theta(r) = |T(r) - T_w| \tag{20}$$

Eq. (21) describes the problem in cylindrical coordinates:

$$\frac{1}{\alpha}\frac{\partial\theta}{\partial t} = \frac{\partial^2\theta}{\partial r^2} + \frac{1}{r}\frac{\partial\theta}{\partial r} \tag{21}$$

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where α is the fuel thermal diffusivity. With the boundary conditions $\theta(r = r_0) = 0$; $\left(\frac{\partial\theta}{\partial r}\right)_{r=0} = 0$; and $\theta(t = 0) = \theta_i$, the general solution for $\theta(r,t)$ is given by Eq. (22):

$$\theta(r,t) = \sum_{n=1}^{\infty} \theta_i J_0(\lambda_n r) e^{-\lambda_n^2 \alpha t} \tag{22}$$

where J_0 is the Bessel function of first kind of order zero and λ_n are the problem eigenvalues, which are the roots of $J_0(\lambda_n r_0) = 0$. For instance, the first roots are $\lambda_1 r_0 = 2.4048$; $\lambda_2 r_0 = 5.5201$; and $\lambda_3 r_0 = 8.6537$. Eq. (22) shows that the fluid temperature at the cylinder axis $T(0,t)$ will exponentially tend to the wall temperature. It is then possible to determine a relaxation time (which can be assimilated to t_k) for which $T(0,t)$ has almost reached this asymptotic condition. The first root ($\lambda_1 r_0 = 2.4048$) leads to the longer relaxation time and must then be the one used in its determination. Focusing at the cylinder axis ($r = 0$) where $J_0 = 1$, the first particular solution from Eq. (22) yields:

$$\frac{\theta(0,t)}{\theta_i} = e^{-\lambda_1^2 \alpha t_k} \tag{23}$$

From where the time may be isolated:

$$t_k = -\ln\left[\frac{\theta(0,t_k)}{\theta_i}\right]\frac{r_0^2}{2.4048^2\alpha} \tag{24}$$

The characteristic time t_k defined in Eq. (24) is the time at which the temperature difference among the fluid at the cylinder axis and the wall, $\theta(0,t_k)$, has been reduced to a certain ratio of the initial difference.

Please note that, from the definition of Eq. (19), the τ_{Ad} ratio defined to assess the problem is proportional to the Fourier number:

$$\tau_{Ad} = \frac{t_{res}}{t_k} = K_{Ad}\frac{\alpha\, t_{res}}{r_0^2} = K_{Ad}\, Fo \tag{25}$$

where K_{Ad} is a constant for the transient radial heat conduction through an injector duct that depends on the problem eigenvalue and the considered ratio among the wall temperature and the fuel temperature at the duct axis. As a remark, the appearance of *Fo* is aligned with the interpretation of τ_{Ad} given earlier, since low values of *Fo* imply that the heat transfer is still at its transient stage, whereas high values of *Fo* imply that steady temperatures are reached and the heat transfer process has been completed.

4.2.2. Estimation for engine-like and laboratory characteristic conditions

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Once the parameter τ_{Ad} has been theoretically defined to qualitatively assess the relevance of heat transfer during the injection process, it is interesting to evaluate it for realistic operating conditions for which experimental data are available:

- Characteristic operating points of a typical passenger car diesel engine are considered. The main characteristics of this 4 cylinder and 1.9 l engine summarized in **Table 2**. Combinations of engine speeds (n_e) from 1250 to 4000 rpm and BMEPs from 3 to 21 bar (representative of the engine load) are taken into account, for a total sweep of 28 engine points. This engine was tested at these operating conditions in different previous works [55–57].
- The conditions reproduced in a laboratory environment to measure the rate of injection (ROI) of the Bosch CRI 2.20 injector (which are part of the data used for the model validation described in the second part of the paper [30]) are evaluated as well. Experimental data of rate of injection are available from a previous work by the authors for a wide range of operating conditions [38]. These include injection pressures (p_i) from 40 to 180 MPa, values of fuel temperature at the injector inlet (T_i) from 253 to 373 K and energizing times (ET) from 0.25 ms to 2 ms. A total of 16 operating conditions were considered for each value of T_i tested.
- Similarly, conditions typically used to gather spray visualization data are also taken into account. This kind of measurements are usually performed in constant-pressure flow or constant-volume preburn vessels and include the determination of liquid-phase and vapor-phase penetration, lift-off length and ignition delay [58]. The same thermodynamic conditions stated for the ROI measurements are considered, with energizing times (ET) ranging from 0.5 to 2 ms in this case, since steady-state conditions are usually sought in the visualization studies.

Engine type	4-cylinder, 4-stroke
Displaced volume	1.9 l
Number of valves/cylinder	4
Stroke	90.4 mm
Bore	82 mm
Geometric compression ratio	17.5:1
Air management	Turbocharged
Maximum power	110 kW @ 4000 rpm
Maximum torque	315 Nm @ 2000 rpm
Injection	Common-rail DI

Table 2. Engine technical data.

As a reference for the fuel residence time (t_{res}), the time needed for the injector to fully evacuate a fuel volume equivalent to the one of its internal lines has been considered. In the case of the Bosch CRI 2.20 injector, the dimensional characterization carried out to build the previous isothermal version of the model provided an approximate value of 6 cm^3 [18] for this volume. For each of the operating conditions tested, the mass injected per stroke (m_i) is known, leading to a certain number of injections being necessary to evacuate the injector volume. If the injection frequency is known, t_{res} can

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be easily estimated. In a four-stroke diesel engine, the injection frequency is half of the engine speed. In the case of the ROI measurements used as a basis for the study, the injection frequency was set to 10 Hz, which is standard in laboratory environments to avoid the influence of one injection on the next one. In the case of the spray visualization measurements, lower injection frequencies are used so that the optical accesses are clear prior to each injection event. An injection frequency of 0.25 Hz has been chosen as representative of this kind of measurement, since it has been used by the authors [59] in the frame of the Engine Combustion Network group (ECN [60]) baseline measurements. For illustrating purposes, **Table 3** shows a sample of the determination of t_{res} for 3 of the characteristic operating conditions that have been assessed for each type of usage given to the injector.

Engine-like conditions						
Condition	n_e [rpm]	BMEP [bar]	m_i [mg/st]	No. injections [-]	t_{res} [s]	
Low speed and load	1250	3.2	11.04	463	44.4	
Maximum torque	2000	20.7	59.69	86	5.2	
Maximum power	4000	15.4	48.22	106	3.2	
Laboratory measurements conditions					**ROI**	**Spray visualization**
Condition	p_i [MPa]	ET [ms]	m_i [mg/st]	No. injections [-]	t_{res} [s]	t_{res} [s]
Low pressure, short	40	0.5	5.25	972	97.2	3888
Medium pressure, long	120	1.0	44.33	116	11.6	464
High pressure, long	180	2.0	116.14	44	4.4	176

Table 3. Sample of t_{res} (time needed for the injector to evacuate a fuel volume equivalent to its own internal volume) for different engine and laboratory characteristic conditions (T_i = 303 K).

As far as the characteristic time of heat transfer due to conduction (t_k) is concerned, it has been seen through the theoretical analysis on the cylindrical duct that it depends on the fuel thermal diffusivity (α) and a characteristic diameter (r_0). α has been evaluated from ρ_f, c_p (see Section 3.1) and k_f (extracted from the literature for a standard diesel fuel [61]) for each operating condition considered in the study. The duct feeding the injector nozzle and surrounding the needle has been taken as a reference for the cylinder r_0 due to its high volume (it accounts for most of the injector volume) and its influence on the internal nozzle flow. The hydraulic diameter of the actual annular duct changes progressively (see **Figure 1**). The section with an average hydraulic diameter (3.84 mm) has been considered representative of the problem for the qualitative evaluation of t_k, leading to $r_0 \approx 1.9$ mm. As shown in Eq. (24), the value of t_k depends on the reduction in the initial difference among the wall temperature and the fluid temperature at the centre of the nozzle feeding line that is considered as a reference.

Table 4 shows the evaluation of t_k from Eq. (24) for several values of reduction of the temperature difference between the fuel centreline and the wall (θ) non-dimensionalized with the initial temperature difference (θ_i). As stated earlier, it is important to note that t_k has also been evaluated through a specific simulation of the transient heat transfer on the actual annular geometry of the Bosch CRI 2.20 injector nozzle feeding line. In the simulation, both the inner and the outer walls

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of the duct are set to a constant temperature T_w different from the initial fuel temperature T_i. Results in this case are also shown in **Table 4**, with θ evaluated at the location where the fuel temperature difference is maximum (analogous to the cylindrical case, where the maximum temperature difference occurs at the centreline). The comparison among the studied geometries shows that the characteristic times always differ by a factor lower than 1.5, implying that the geometry chosen to compute τ_{Ad} will not impact the order of magnitude of the estimations and the validity of the associated qualitative analysis. The cylindrical duct case is then considered hereinafter. A ratio $\theta/\theta_i = 0.3678$ (corresponding to the number e^{-1}) has been selected as a reference for t_k since it leads to the logarithm in Eq. (24) being cancelled, which is a standard in the definition of relaxation times. Please note that, in the view of the values shown in Table **4**, this arbitrary choice of t_k = 7.34 s is not expected to modify the order of magnitude of τ_{Ad} and the conclusions stated in this section either.

	t_k [s]	
θ/θ_i (%) [-]	Cylindrical duct (Eq. 24)	Annular duct (simulated)
75	2.24	2.35
50	5.41	4.22
36.78	**7.34**	5.61
20	11.82	8.37
10	16.91	11.5
5	22.0	14.63

Table 4. Estimation of t_k for several values of the ratio among θ and θ_i, both for the cylindrical duct theoretical case and the simulated annular duct case.

According to Eq. (19), τ_{Ad} can now be evaluated from the estimations of t_{res} and t_k. This parameter is depicted in Figure **5(a)** as a function of the engine-like operating conditions (engine speed and load), whereas it is shown for the conditions of the ROI and visualization measurements (injection pressure and energizing time) corresponding to a reference value of T_i = 303 K in Figure **5(b)**.

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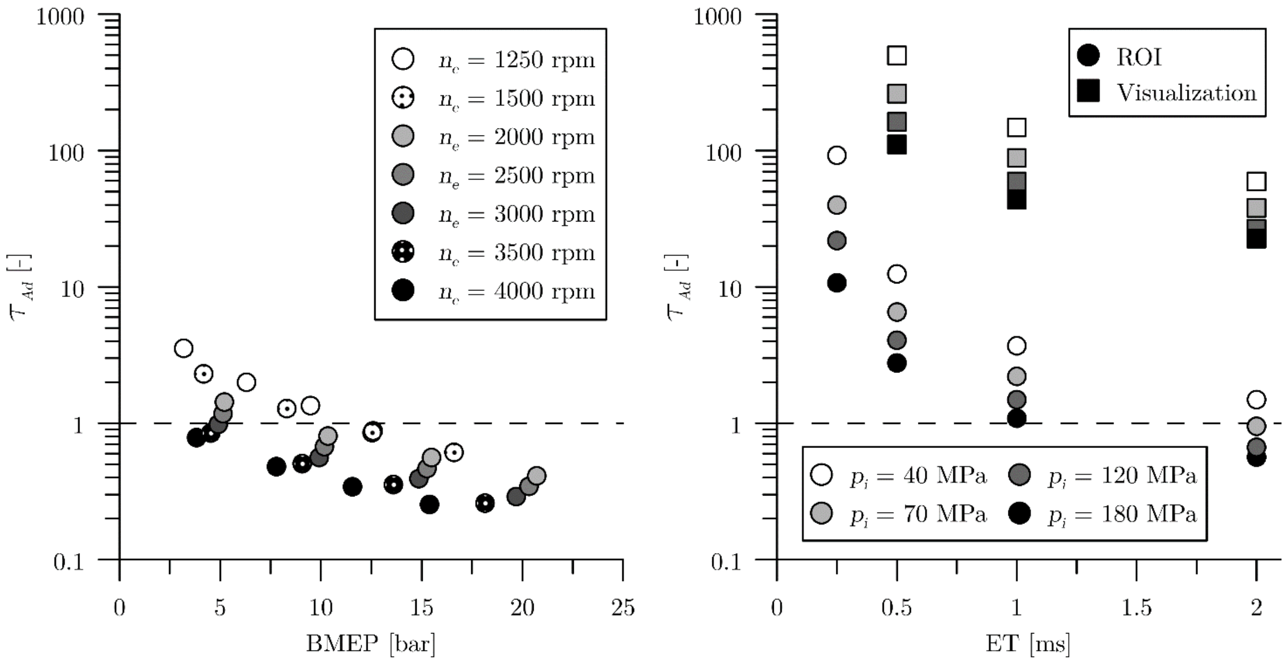

(a) Engine-like conditions. (b) Laboratory measurements conditions.

Figure 5. Estimation of τ_{Ad} for engine-like conditions corresponding to different values of engine speed and load (a) and for typical conditions of laboratory measurements including different values of injection pressure and ET (b).

Results of Figure 5(a) show that τ_{Ad} is higher than 1 for the lowest engine speeds and lowest engine loads, being lower than 1 for all engine speeds in the case of medium to high engine loads. For the low to medium engine loads, τ_{Ad} also gets lower than 1 from a certain intermediate value of engine speed. In any case, τ_{Ad} always remains within the orders of magnitude closer to 1 ($0.1 < \tau_{Ad} < 10$). Therefore, the order of magnitude of the fuel residence time within the injector is similar to the one for the characteristic time of heat transfer with the surroundings by conduction. As stated earlier this implies that, in these conditions, heat transfer seems to be relevant to a certain extent. Thus, the temperature of the fuel inside the injector has enough time to get closer to the injector wall temperature prior to each injection event, departing from the initial fuel temperature right at the injector inlet. In any case, the heat transfer process among injections will still be at its transient stage by the time of a new injection and the fuel temperature is not expected to have reached the wall temperature completely. Prior to each injection, the fuel temperature will be more similar to the wall temperature for the lowest engine speeds and loads ($1 < \tau_{Ad} < 10$), whereas its change from the fuel initial temperature will be less accused (but existent) the higher the engine speed and the engine load ($0.1 < \tau_{Ad} < 1$). During a given injection, however, since the wall temperature and the fuel temperature will already be closer to each other, the considerations previously given about the adiabatic assumption with the *Ad* number are valid: if the *Ad* number (established by both the geometry of each injector restriction and the induced operating conditions) is high (as it has been shown for the Bosch CRI 2.20 injector orifices in steady-state conditions), the already low temperature difference among fluid and wall will prevent the fuel from further

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changing its temperature due to heat exchange. Only its own internal evolution due to the strong pressure drops established across the orifices will be able to modify the fuel temperature. Hence, the assumption of adiabatic flow during an injection may still hold with the particularity that a certain temperature gap should be initially considered for T_i prior to each injection (again, this gap is more accused for the lowest engine speeds and loads). This finding is particularly useful to set up computational simulations or to adjust correlations of certain variables based on experimental measurements. On the contrary, if the *Ad* number is low (as it may happen during the transient operation of the injector where the pressure gradients in the orifice are still low), the temperature changes along the injector cannot be solely explained by the pressure drops across the injector internal orifices and will be partially due to heat exchange with the surroundings. Moreover, for low values of *Ad*, the flow regime across the injector ducts would probably be laminar, with the laminar boundary layer additionally slowing down the flow close to the injector walls and reinforcing the heat exchange relevance.

As far as the laboratory measurements conditions depicted in Figure **5(b)** are concerned, τ_{Ad} decreases with the injection pressure and the injector energizing time, as expected. Focusing on the ROI measurements conditions, similar conclusions to the ones found for the engine-like conditions can be extracted in the view of the results. Nevertheless, some operating conditions lead to a different order of magnitude for τ_{Ad} ($10 < \tau_{Ad} < 100$). In particular, the shortest injections fall in this regime. This means that, under these conditions, the heat transfer process is practically completed by the time of a new injection. In these cases, it is accurate to assume that the fuel temperature already matches the wall temperature prior to each injection. During the injection itself, the low temperature difference among the wall and the fluid could lead again to heat exchange being negligible. The magnitude of the temperature change across the internal restrictions of the injector would depend on the induced *Ad* values, as explained earlier. In any case, it is important to note that the results presented correspond to an injection frequency of 10 Hz. The conclusions could differ if a different injection frequency was selected. In fact, τ_{Ad} is an order of magnitude larger for the laboratory conditions sought in spray visualization measurements, with an injection frequency of 0.25 Hz. Several conditions fall in the regime corresponding to $100 < \tau_{Ad} < 1000$. In these conditions, it is accurate to assume that the initial fuel temperature at the injector inlet completely matches the wall temperature. This result will be of interest to experimentalists that insulate the injector holder for their measurements seeking to control the fuel injection temperature. Again, no relevant heat transfer would take place during the injection as long as the induced values of *Ad* were high, so that the fuel temperature evolution from the injector inlet to the nozzle outlet could be estimated with the adiabatic flow hypothesis as introduced in Section 3. These results are analysed with the implemented computational model in the second part of the paper [30].

Therefore, from the results of Figure **5(b)** it follows that several options are available if the influence of heat transfer

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wanted to be isolated when designing a laboratory experiment. On the one hand, the laboratory user may directly control the relative importance of heat transfer in the injection process through the injection frequency. On the other hand, this issue can also be manipulated by properly selecting the ET of the injections.

It is important to give some remarks about the limitations of this study. The proposed limiting case behaviour (with the implications of t_{Ad} being lower, equal or greater than 1) can only give an idea on the qualitative importance of the different phenomena involved in the process, being reliable only in terms of orders of magnitude. Some relevant phenomena cannot be captured by a single time t_k to account for conduction from injector walls to fuel. For instance, in the real application, important temperature gradients may be present along the injector walls given the higher temperatures at which the nozzle tip is submitted, as opposed to the rest of the injector body. These temperature gradients may differ depending on the duration of the engine run. However, this could only be accounted for through a model with yet more uncertainties that would hinder the analysis of results. In any case, this fact is not deemed to condition the order of magnitude of the results here presented and their qualitative interpretation.

5. CONCLUSIONS

The introduction of the hypothesis of adiabatic flow to account for local temperature variations in the computational model of a common-rail solenoid diesel injector has been dealt with in this work. The main features of the model have been described, focusing on the implementation of the hypothesis on a previous version of the model (which employed the isothermal flow assumption) and the qualitative assessment of its validity. To do so, two non-dimensional parameters have been analysed to estimate the importance of the heat transfer processes both during the injection and in the time lapse among injections. The main findings of the present work are summarized as follows:

- The adiabatic flow hypothesis implies that the fuel temperature change along an injector hydraulic restriction (such as an orifice) depends on the pressure drop and the velocity change (which in the end is related to the discharge coefficient) through the restriction. If the flow velocity remained constant along an adiabatic process, the conservation of fuel specific enthalpy would imply a heating upon expansion.
- During the injection, the proneness of the flow along the injector to exchange heat with the surroundings depends on a non-dimensional quantity (*Ad*) that is a function of the orifice diameter-to-length ratio, *Re* and *Nu*. Consequently, for the injector of study it was found that heat transfer may be relevant when low values of *Re* are induced. This is the case of the transient stages (opening and closing), for which large pressure drops along the injector orifices are still not established.
- The heat exchange process existing in the time lapse among injections has been analysed by means of a non-

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dimensional parameter (τ_{Ad}) proportional to *Fo* that compares the fuel residence times within the injector to a characteristic time of heat transfer. This characteristic time has been assessed with the simplified case of transient radial conduction in an infinitely long cylinder (although its assessment through a simulation conducted for a more realistic injector duct proved to lead to the same qualitative conclusions). The evaluation of τ_{Ad} for typical engine operating conditions shows that the heat transfer process is generally still at its transient stage prior to each injection. Thus, heat transfer is relevant to a certain extent and, by the time of a new injection, the fuel temperature is not expected to have reached the injector wall temperature completely. In any case, the lower the engine speed and load, the closer these two temperatures will get to each other before each injection. The findings were similar when evaluating τ_{Ad} for the usual operating conditions of the ROI measurements typically performed in a laboratory environment, with the exception of the short injections. In this case, as it also happens with the usual conditions of visualization experiments, the high residence times associated to the injections imply that the heat transfer process will be completed prior to each injection, resulting in the fuel temperature at the injector inlet being equal to the injector wall temperature.

- In the case of laboratory measurements, the injection frequency and the injection energizing time play a key role on the heat transfer processes. This fact may be used to the advantage of the researcher when designing a new experiment, since these conditions may be selected so that heat transfer only occurs either during the injection or in the time lapse among injections and its influence is isolated from the study. On the one hand, using high injection frequencies (or large energizing times) reduces the fuel residence time within the injector, so that the heat transfer in the lapse among injections becomes negligible. In this case, any heat transfer would take place during the injector opening and closing (where low values of *Ad* are present), whose relative importance in the injection event is lower the higher the energizing time. On the other hand, using low injection frequencies (or short energizing times) leads to the heat transfer process among injections being completed so that by the time of a new injection the fuel temperature already matches the wall temperature. In this case, no relevant heat transfer would take place during the injection as long as the induced values of *Ad* were high.
- The previous conclusion implies that the fuel temperature changes along the injector in steady-state conditions could be reasonably predicted through the use of the adiabatic flow hypothesis. Even if heat transfer was relevant during the time lapse among injections, the assumption of adiabatic flow could hold during the injection. This means that a computational model (or an experimental correlation depending on fuel properties) can be reliable if a certain temperature gap is considered for the fuel temperature and its associated properties at the injector

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inlet prior to each injection.

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REFERENCES

[1] Heywood JB. Internal Combustion Engine Fundamentals. vol. 21. 1988.

[2] Payri R, Salvador FJ, Gimeno J, De la Morena J. Influence of injector technology on injection and combustion development, Part 1: Hydraulic characterization. Appl Energy 2011;88:1068–74. doi:10.1016/j.apenergy.2010.10.012.

[3] Payri R, Salvador FJ, Gimeno J, De la Morena J. Influence of injector technology on injection and combustion development, Part 2: Combustion analysis. Appl Energy 2011;88:1130–9. doi:10.1016/j.apenergy.2010.10.012.

[4] Gavaises M. Flow in valve covered orifice nozzles with cylindrical and tapered holes and link to cavitation erosion and engine exhaust emissions. Int J Engine Res 2008;9:435–47.

[5] Som S, Ramírez AI, Longman DE, Aggarwal SK. Effect of nozzle orifice geometry on spray, combustion, and emission characteristics under diesel engine conditions. Fuel 2011;90:1267–76. doi:10.1016/j.fuel.2010.10.048.

[6] Salvador FJ, Carreres M, Jaramillo D, Martínez-López J. Analysis of the combined effect of hydrogrinding process and inclination angle on hydraulic performance of diesel injection nozzles. Energy Convers Manag 2015;105:1352–65. doi:10.1016/j.enconman.2015.08.035.

[7] Nguyen D, Duke D, Kastengren A, Matusik K, Swantek A, Powell CF, et al. Spray flow structure from twin-hole diesel injector nozzles. Exp Therm Fluid Sci 2017;86:235–47. doi:10.1016/j.expthermflusci.2017.04.020.

[8] Salvador FJ, de la Morena J, Carreres M, Jaramillo D. Numerical analysis of flow characteristics in diesel injector nozzles with convergent-divergent orifices. Proc Inst Mech Eng Part D J Automob Eng 2017;231:1935–44. doi:10.1177/0954407017692220.

[9] Lee CS, Lee KH, Reitz RD, Park SW. Effect of split injection on the macroscopic development and atomization

doi: 10.1016/j.fuel.2019.116348

characteristics of a diesel spray injected through a common-rail system. At Sprays 2006;16:543–62. doi:10.1615/AtomizSpr.v16.i5.50.

[10] Wang X, Huang Z, Zhang W, Abiola O, Nishida K. Effects of ultra-high injection pressure and micro-hole nozzle on flame structure and soot formation of impinging diesel spray. Appl Energy 2011;88:1620–8. doi:10.1016/j.apenergy.2010.11.035.

[11] Gumus M, Sayin C, Canakci M. The impact of fuel injection pressure on the exhaust emissions of a direct injection diesel engine fueled with biodiesel-diesel fuel blends. Fuel 2012;95:486–94. doi:10.1016/j.fuel.2011.11.020.

[12] Bianchi GM, Falfari S, Pelloni P, Kong S-C, Reitz RD. Numerical Analysis of High-Pressure Fast-Response Common Rail Injector Dynamics. SAE Tech Pap 2002-01-0213 2002. doi:10.4271/2002-01-0213.

[13] Marcer R, Audiffren C, Viel A, Bouvier B, Walbott A, Argueyrolles B. Coupling 1D System AMESim and 3D CFD EOLE models for Diesel Injection Simulation Renault. ILASS - Eur. 2010, 23rd Annu. Conf. Liq. At. Spray Syst., 2010, p. 1–10.

[14] Arpaia A, Catania AE, Ferrari A, Spessa E. Development and Application of an Advanced Numerical Model for CR Piezo Indirect Acting Injection Systems. SAE Tech Pap 2010-01-1503 2010. doi:10.4271/2010-01-1503.

[15] Plamondon E, Seers P. Development of a simplified dynamic model for a piezoelectric injector using multiple injection strategies with biodiesel/diesel-fuel blends. Appl Energy 2014;131:411–24. doi:10.1016/j.apenergy.2014.06.039.

[16] Salvador FJ, Gimeno J, De la Morena J, Carreres M. Using one-dimensional modeling to analyze the influence of the use of biodiesels on the dynamic behavior of solenoid-operated injectors in common rail systems: Results of the simulations and discussion. Energy Convers Manag 2012;54:122–32. doi:10.1016/j.enconman.2011.10.007.

[17] Salvador FJ, Plazas AH, Gimeno J, Carreres M. Complete modelling of a piezo actuator last-generation injector for diesel injection systems. Int J Engine Res 2014;15:3–19. doi:10.1177/1468087412455373.

[18] Payri R, Salvador FJ, Carreres M, De la Morena J. Fuel temperature influence on the performance of a last generation common-rail diesel ballistic injector. Part II: 1D model development, validation and analysis. Energy Convers Manag 2016;114:376–91. doi:10.1016/j.enconman.2016.02.043.

[19] Wang X, Huang Z, Kuti OA, Zhang W, Nishida K. Experimental and analytical study on biodiesel and diesel spray characteristics under ultra-high injection pressure. Int J Heat Fluid Flow 2010;31:659–66.

Salvador, F.J., Gimeno, J., Martín, J., Carreres, M., “Thermal effects on the diesel injector performance through adiabatic 1D modelling. Part I: Model description and assessment of the adiabatic flow hypothesis”, Fuel 260:116348 (author version).

doi: 10.1016/j.fuel.2019.116348

doi:10.1016/j.ijheatfluidflow.2010.03.006.

[20] Theodorakakos A, Strotos G, Mitroglou N, Atkin C, Gavaises M. Friction-induced heating in nozzle hole micro-channels under extreme fuel pressurisation. Fuel 2014;123:143–50. doi:10.1016/j.fuel.2014.01.050.

[21] Strotos G, Koukouvinis P, Theodorakakos A, Gavaises M, Bergeles G. Transient heating effects in high pressure Diesel injector nozzles. Int J Heat Fluid Flow 2015;51:257–67. doi:10.1016/j.ijheatfluidflow.2014.10.010.

[22] Salvador FJ, Carreres M, De la Morena J, Martínez-Miracle E. Computational assessment of temperature variations through calibrated orifices subjected to high pressure drops: Application to diesel injection nozzles. Energy Convers Manag 2018;171:438–51. doi:10.1016/j.enconman.2018.05.102.

[23] Desantes JM, Salvador FJ, Carreres M, Jaramillo D. Experimental Characterization of the Thermodynamic Properties of Diesel Fuels Over a Wide Range of Pressures and Temperatures. SAE Int J Fuels Lubr 2015;8:2015-01–0951. doi:10.4271/2015-01-0951.

[24] Dernotte J, Hespel C, Foucher F, Houillé S, Mounaïm-Rousselle C. Influence of physical fuel properties on the injection rate in a Diesel injector. Fuel 2012;96:153–60. doi:10.1016/j.fuel.2011.11.073.

[25] Park Y, Hwang J, Bae C, Kim K, Lee J, Pyo S. Effects of diesel fuel temperature on fuel flow and spray characteristics. Fuel 2015;162:1–7. doi:10.1016/j.fuel.2015.09.008.

[26] Seykens X, Somers LMT, Baert RSG. Modelling of common rail fuel injection system and influence of fluid properties on process. Proc. VAFSEP, Dublin, Ireland; July 6-9, 2004, p. 6–9.

[27] Catania AE, Ferrari A, Spessa E. Temperature variations in the simulation of high-pressure injection-system transient flows under cavitation. Int J Heat Mass Transf 2008;51:2090–107. doi:10.1016/j.ijheatmasstransfer.2007.11.032.

[28] Yu H, Goldsworthy L, Brandner PA, Li J, Garaniya V. Modelling thermal effects in cavitating high-pressure diesel sprays using an improved compressible multiphase approach. Fuel 2018;222:125–45. doi:10.1016/j.fuel.2018.02.104.

[29] Salvador FJ, Gimeno J, Carreres M, Crialesi-Esposito M. Experimental assessment of the fuel heating and the validity of the assumption of adiabatic flow through the internal orifices of a diesel injector. Fuel 2017;188:442–51. doi:10.1016/j.fuel.2016.10.061.

[30] Payri R, Salvador FJ, Carreres M, Belmar-Gil M. Thermal effects on the diesel injector performance through adiabatic 1D modelling. Part II: Model validation, results of the simulations and discussion. Fuel 2019;In

Salvador, F.J., Gimeno, J., Martín, J., Carreres, M., “Thermal effects on the diesel injector performance through adiabatic 1D modelling. Part I: Model description and assessment of the adiabatic flow hypothesis”, Fuel 260:116348 (author version).

doi: 10.1016/j.fuel.2019.116348

review.

[31] LMS. Imagine.Lab AMESim v.10. User's manual 2010.

[32] Salvador FJ, Gimeno J, De la Morena J, Carreres M. Comparison of Different Techniques for Characterizing the Diesel Injector Internal Dimensions. Exp Tech 2018;42:467–72. doi:10.1007/s40799-018-0246-1.

[33] Desantes JM, Lopez JJ, Carreres M, López-Pintor D. Characterization and prediction of the discharge coefficient of non-cavitating diesel injection nozzles. Fuel 2016;184:371–81. doi:10.1016/j.fuel.2016.07.026.

[34] Leonhard R, Warga J, Pauer T, Rückle M, Schnell M. Solenoid common-rail injector for 1800 bar. MTZ Worldw 2010;71:10–5. doi:10.1007/BF03227003.

[35] Payri R, Garcia-Oliver JM, Salvador FJ, Gimeno J. Using spray momentum flux measurements to understand the influence of diesel nozzle geometry on spray characteristics. Fuel 2005;84:551–61. doi:10.1016/j.fuel.2004.10.009.

[36] Siebers DL. Scaling liquid-phase fuel penetration in diesel sprays based on mixing-limited vaporization. SAE Tech Pap 1999-01-0528 1999. doi:10.4271/1999-01-0528.

[37] Desantes JM, Salvador FJ, Carreres M, Martínez-López J. Large-eddy simulation analysis of the influence of the needle lift on the cavitation in diesel injector nozzles. Proc Inst Mech Eng Part D J Automob Eng 2014;229:407–23. doi:10.1177/0954407014542627.

[38] Salvador FJ, Gimeno J, Carreres M, Crialesi-Esposito M. Fuel temperature influence on the performance of a last generation common-rail diesel ballistic injector. Part I: Experimental mass flow rate measurements and discussion. Energy Convers Manag 2016;114:364–75. doi:10.1016/j.enconman.2016.02.042.

[39] Chorążewski M, Dergal F, Sawaya T, Mokbel I, Grolier JE, Jose J. Thermophysical properties of Normafluid (ISO 4113) over wide pressure and temperature ranges. Fuel 2013;105:440–50. doi:10.1016/j.fuel.2012.05.059.

[40] Huang D, Simon SL, McKenna GB. Chain length dependence of the thermodynamic properties of linear and cyclic alkanes and polymers. J Chem Phys 2005;122:84907. doi:10.1063/1.1852453.

[41] Bell IH, Wronski J, Quoilin S, Lemort V. Pure and pseudo-pure fluid thermophysical property evaluation and the open-source thermophysical property library coolprop. Ind Eng Chem Res 2014;53:2498–508. doi:10.1021/ie4033999.

[42] Růžička Jr V, Domalski ES. Estimation of the heat capacities of organic liquids as a function of temperature using group additivity. I. Hydrocarbon compounds. J Phys Chem Ref Data 1993;22:597–618. doi:10.1063/1.555923.

Salvador, F.J., Gimeno, J., Martín, J., Carreres, M., "Thermal effects on the diesel injector performance through adiabatic 1D modelling. Part I: Model description and assessment of the adiabatic flow hypothesis", Fuel 260:116348 (author version).

doi: 10.1016/j.fuel.2019.116348

[43] Zábranský M, Kolská Z, Růžička V, Domalski ES. Heat capacity of liquids: Critical review and recommended values. Supplement II. J Phys Chem Ref Data 2010;39:3–404. doi:10.1063/1.3182831.

[44] Kouzel B. How pressure affects liquid viscosity. Hydrocarb Process Pet Refin 1965;44 (3):120.

[45] Winklhofer E, Wiesler B, Ahmadi-Befrui B, Cresnoverh G. The influence of injection rate shaping on Diesel fuel sprays - an experimental study. ImechE 1992;206:173–83.

[46] Nishimura T, Satoh K, Takahashi S, Yokota K. Effects of Fuel Injection Rate on Combustion and Emission in a DI Diesel Engine. SAE Tech. Pap. 981929, 1998. doi:10.4271/981929.

[47] Benajes J, Molina S, De Rudder K, Rente T. Influence of Injection Rate Shaping on Combustion and Emissions for a Medium Duty Diesel Engine. J Mech Sci Technol 2006;20:1436–48.

[48] He Z, Xuan T, Xue Y, Wang Q, Zhang L. A numerical study of the effects of injection rate shape on combustion and emission of diesel engines. Therm Sci 2014;18:67–78. doi:10.2298/TSCI130810013H.

[49] Payri F, Payri R, Bardi M, Carreres M. Engine combustion network: Influence of the gas properties on the spray penetration and spreading angle. Exp Therm Fluid Sci 2014;53:236–43. doi:10.1016/j.expthermflusci.2013.12.014.

[50] Sieder EN, Tate GE. Heat Transfer and Pressure Drop of Liquids in Tubes. Ind Eng Chem 1936;28:1429–35. doi:10.1021/ie50324a027.

[51] Nusselt W. Der Wärmeaustausch zwischen Wand und Wasser im Rohr. Forsch Auf Dem Gebiete Des Ingenieurwesens 1931;2:309–13. doi:10.1007/BF02583210.

[52] Salvador FJ, Carreres M, Crialesi-Esposito M, Plazas AH. Determination of critical operating and geometrical parameters in diesel injectors through one dimensional modelling, design of experiments and an analysis of variance. Proc Inst Mech Eng Part D J Automob Eng 2018;232:1762–81. doi:10.1177/0954407017735262.

[53] Matsumoto S, Yamada K, Date K. Concepts and Evolution of Injector for Common Rail System. SAE Tech Pap 2012-01-1753 2012. doi:10.4271/2012-01-1753.

[54] Schöppe D, Zülch S, Hardy M, Geurts D, Jorach RW, Baker N. Delphi Common Rail system with direct acting injector. MTZ Worldw 2008;69:32–8. doi:10.1007/BF03226918.

[55] Benajes J, Olmeda P, Martín J, Blanco-Cavero D, Warey A. Evaluation of swirl effect on the Global Energy Balance of a HSDI Diesel engine. Energy 2017;122:168–81. doi:10.1016/j.energy.2017.01.082.

[56] Broatch A, Olmeda P, García A, Salvador-Iborra J, Warey A. Impact of swirl on in-cylinder heat transfer in a light-duty diesel engine. Energy 2017;119:1010–23. doi:10.1016/j.energy.2016.11.040.

Salvador, F.J., Gimeno, J., Martín, J., Carreres, M., “Thermal effects on the diesel injector performance through adiabatic 1D modelling. Part I: Model description and assessment of the adiabatic flow hypothesis”, Fuel 260:116348 (author version).

doi: 10.1016/j.fuel.2019.116348

[57] Broatch A, Martín J, García A, Blanco-Cavero D, Warey A, Domenech V. Application of a zero-dimensional model to assess the effect of swirl on indicated efficiency. Int J Engine Res 2018;In press. doi:10.1177/1468087418779726.

[58] Bardi M, Payri R, Malbec L-M, Bruneaux G, Pickett LM, Manin J, et al. Engine Combustion Network: Comparison of Spray Development, Vaporization, and Combustion in Different Combustion Vessels. At Sprays 2012;22:807–42. doi:10.1615/AtomizSpr.2013005837.

[59] Gimeno J, Martí-Aldaraví P, Carreres M, Peraza JE. Effect of the nozzle holder on injected fuel temperature for experimental test rigs and its influence on diesel sprays. Int J Engine Res 2018;19:374–89. doi:10.1177/1468087417751531.

[60] ECN. Engine Combustion Network. Https://EcnSandiaGov/Diesel-Spray-Combustion/ 2010. www.sandia.gov/ecn/.

[61] Kolev NI. Thermodynamic and transport properties of diesel fuel. Multiph. Flow Dyn. 3 Turbul. gas Absorpt. release, diesel fuel Prop. 3rd editio, Springer Verlag; 2007, p. 293–327.

Salvador, F.J., Gimeno, J., Martín, J., Carreres, M., “Thermal effects on the diesel injector performance through adiabatic 1D modelling. Part I: Model description and assessment of the adiabatic flow hypothesis”, Fuel 260:116348 (author version).

doi: 10.1016/j.fuel.2019.116348