Fuel 188 (2017) 442–451

Contents lists available at ScienceDirect

Fuel

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

Full Length Article

Experimental assessment of the fuel heating and the validity of the assumption of adiabatic flow through the internal orifices of a diesel injector

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A R T I C L E I N F O

Article history:
Received 11 February 2016
Received in revised form 15 September 2016
Accepted 11 October 2016
Available online 15 October 2016

Keywords:
Diesel
Experimental
Fuel heating
Fuel temperature
Adiabatic flow

A B S T R A C T

In this paper an experimental investigation on the heating experienced by the fuel when it expands through the calibrated orifices of a diesel injector is carried out. Five different geometries corresponding to the control orifices of two different commercial common-rail solenoid injectors were tested. An experimental facility was used to impose a continuous flow through the orifices by controlling the pressures both upstream and downstream of the restriction. Fuel temperature was controlled prior to the orifice inlet and measured after the outlet at a location where the flow is already slowed down. Results were compared to the theoretical temperature increase under the assumption of adiabatic flow (i.e. isenthalpic process). The comparison points out that this assumption allows to predict the fuel temperature change in a reasonable way for four of the five geometries as long as the pressure difference across the orifice is high enough. The deviations for low imposed pressure differences and the remaining orifice are explained due to the low Reynolds numbers (i.e. flow velocities) induced in these cases, which significantly increase the residence time of a fuel particle in the duct, thus enabling heat transfer with the surrounding atmosphere. A dimensionless parameter to quantify the proneness of the flow through an orifice to exchange heat with the surroundings has been theoretically derived and calculated for the different geometries tested, allowing to establish a boundary that defines beforehand the conditions from which heat losses to the ambient can be neglected when dealing with the internal flow along a diesel injector.

1. Introduction

Based on the attention it has been given by researchers in the diesel engine community, the direct injection system is one of the key elements on the engine's outcome. It is directly related to the air-fuel mixture quality [1–3], which results in a strong influence on the combustion phenomenon, thus affecting the fuel consumption and emissions [4–7]. Advances in the direct injection systems features have been used as a vehicle to reduce both soot and NO_x emissions in order to meet the restrictive requirements of the Euro 6 legislation and those to come. As a result, for instance, the injection pressure realizable by the injection systems is growing at a fast rate, already reaching 250 or 300 MPa. This results in a higher complexity of the injection system, which highlights the need for advanced tools that allow to predict its behaviour at the wide range of engine operating conditions [8].

It is therefore important to develop computational tools to simulate the internal flow through injector systems. Approaches have been made through one-dimensional modelling of the complete injector flow [9–14] or three-dimensional modelling of the nozzle flow [4,15–18]. Some of these works assume that the flow is isothermal [11–14]. Nevertheless, the raising injection pressures may also induce relevant fuel temperature changes due to friction heating or due to important fuel depressurization across the injector control orifices or the nozzle [15,16]. These fuel temperature changes, in turn, affect the fuel properties, which are strongly dependant on temperature and pressure [19–23]. For this reason, some modellers prefer to assume the flow along the injector as adiabatic [9,10]. However, attention has never been given to the validation of this hypothesis, which may not be true under some real circumstances. Even though the temperature difference among the fuel and the injector walls may be high, heat transfer to the surroundings is not expected to be relevant if the flow velocity is high enough to lead to low residence times of a fuel particle within the injector. However, the adiabatic assumption may not hold if

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http://dx.doi.org/10.1016/j.fuel.2016.10.061
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Nomenclature

A_o	outlet area
A_P	area submitted to heat transfer
Ad	adiabatic number
C_d	discharge coefficient
c_p	fluid heat capacity at constant pressure
D_i	inlet diameter
D_o	outlet diameter
e	specific internal energy
h	specific enthalpy
k	thermal conductivity
L	length
$\dot{m}$	mass flow
$\dot{m}_{th}$	theoretical mass flow
Nu	Nusselt number
p	pressure
p_0	reference pressure
p_{dw}	downstream pressure
p_{up}	upstream pressure
p_v	vapour pressure
P	perimeter
Pr	Prandtl number
$\dot{q}$	heat flux
Re	Reynolds number
Re_{crit}	critical Reynolds number
St	Stanton number
T	temperature
T_0	reference temperature
T_{dw}	downstream temperature
T_f	fluid temperature
T_{up}	upstream temperature
T_w	wall temperature
u	flow velocity
u_m	mean flow velocity
u_{th}	theoretical flow velocity

Greek symbols

α	convective heat transfer coefficient
β	volumetric thermal expansion coefficient
Δp	pressure drop
ΔT	temperature change
ΔT_{exp}	experimental temperature change
ΔT_{th}	theoretical temperature change
μ	absolute viscosity
μ_b	absolute viscosity evaluated at the bulk temperature
μ_w	absolute viscosity evaluated at the wall
ρ	density

the fuel velocity is too low and there is enough time for the fuel to interact with the ambient.

The purpose of this work is to shed light on the previous issues by having a look at the flow through the most important restrictions in diesel injectors from an experimental point of view. Continuous flow through the two control orifices of a Bosch CRI 2.20 injector (described in [23]) and the three control orifices of a Denso G4S (which uses a three-way valve to hydraulically pilot the injector, as described in [24]) was established for different imposed pressure drops, making it possible to measure the generated increase in temperature. Results were compared to the theoretical temperature increase expected if the expansion through the orifice was isenthalpic, which includes the assumption of adiabatic flow. An analysis of the deviations in temperature increase between the experiments and the theoretical isenthalpic process was performed in order to establish the conditions under which the assumption of adiabatic flow through a diesel injector is valid. In this regard, a dimensionless parameter was defined in order to condense the information of the five tested orifices in a single parameter that quantifies the proneness of a certain orifice to exchange heat with the surroundings when working under specific conditions.

2. Theoretical temperature change for an isenthalpic process

As it has been stated, the experimental results of the temperature change across the orifice upon expansion will be compared to the theoretical temperature change predicted under the assumption of adiabatic flow, with no heat transfer to the surroundings. According to the first law and in the absence of external work, this assumption implies that the stagnation enthalpy is conserved along the orifice:

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

Furthermore, if the reference locations upstream and downstream of the orifice at which the pressure is controlled are placed far enough from the orifice (so as to assume that the flow velocity is similar at those locations), the specific enthalpy of the flow is supposed to remain constant along the process carried out in the experiments. Then, under these conditions, the flow may be regarded to as isenthalpic.

The relationship among the specific enthalpy of the flow and its internal energy is given by Eq. (2):

$$h = e + \frac{p}{\rho} \qquad (2)$$

It is important to note that, due to the small diameters involved in the study, the heating induced by viscous dissipation (i.e. friction) along the orifices, or by the fact that the flow is slowed down downstream of the orifices, is deemed to be important [15,16,25–28]. Under the assumption of isenthalpic flow, this heat is supposed to remain within the fluid, contributing to increase its internal energy and therefore its temperature while the fluid expands, according to Eq. (2).

In order to calculate the temperature change in an isenthalpic expansion, let us consider the general formulation for the specific enthalpy as a function of the fluid temperature and pressure:

$$dh = \left(\frac{\partial h}{\partial T}\right)_p dT + \left(\frac{\partial h}{\partial p}\right)_T dp = c_p dT + \frac{1-\beta T}{\rho} dp \qquad (3)$$

where c_p is the fluid heat capacity at constant pressure and β its volumetric thermal expansion coefficient, defined as:

$$\beta = \frac{1}{v}\left(\frac{\partial v}{\partial T}\right)_p = -\frac{1}{\rho}\left(\frac{\partial \rho}{\partial T}\right)_p \qquad (4)$$

Note that the second term of the right hand side of Eq. (3) is null for an ideal gas (for which $\beta T = 1$), but cannot be neglected for a liquid. If the variation of the fluid properties (c_p, β and ρ) with respect to the pressure and temperature is known, Eq. (3) can be integrated taking into account that the final result is independent on the integration path [29]. This implies that it is possible to determine the fluid temperature after the expansion if the temperature upstream of the orifice and the pressure both upstream and downstream of