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Numerical modelling of diesel spray using the Eulerian multiphase approach

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ABSTRACT

This research investigates high pressure diesel fuel injection into the combustion chamber by performing computational simulations using the Euler–Eulerian multiphase approach. Six diesel-like conditions were simulated for which the liquid fuel jet was injected into a pressurised inert environment (100% N_2) through a 205 μm nozzle hole. The analysis was focused on the liquid jet and vapour penetration, describing spatial and temporal spray evolution. For this purpose, an Eulerian multiphase model was implemented, variations of the sub-model coefficients were performed, and their impact on the spray formation was investigated. The final set of sub-model coefficients was applied to all operating points. Several simulations of high pressure diesel injections (50, 80, and 120 MPa) combined with different chamber pressures (5.4 and 7.2 MPa) were carried out and results were compared to the experimental data. The predicted results share a similar spray cloud shape for all conditions with the different vapour and liquid penetration length. The liquid penetration is shortened with the increase in chamber pressure, whilst the vapour penetration is more pronounced by elevating the injection pressure. Finally, the results showed good agreement when compared to the measured data, and yielded the correct trends for both the liquid and vapour penetrations under different operating conditions.

1. Introduction

Diesel engines produce pollutant emissions that cause environmental problems and can be harmful to human health. Allowable pollutant emissions from diesel engines have been regulated over the last few decades and new, more stringent regulations are expected within the next years. These regulations include the European emission standards arising from one of the governmental policies as an option for accomplishing cleaner production [1]. Due to the promotion of biofuels by the European Union [2], diesel engines must be subject to further development and meet higher efficiency standards [3] in order to remain the most used transportation vehicle powering system on the market. A significant amount of CO_2 is released into the atmosphere by combusting fossil fuels [4], and a rapid emission reduction (up to 85%) has to be achieved by 2050 [5]. It is reasonable to use the diesel engines as the internal combustion engine power source due to their more efficient energy conversion and higher safety factor when compared to the spark ignition engines [6]. In internal combustion diesel engines there is mostly diffusion combustion present, meaning that the spray characteristics have the direct influence on the fuel energy conversion and the formation of harmful substances [7–11]. There are challenges associated with having a very short amount of time available for the fuel spray to atomise and form an adequate mixture for quality combustion. Therefore, suitable fuel injectors are needed to provide sufficient control on the spray process and to meet the basic requirements for the atomisation and mixing process. High pressure injectors are one of the most commonly used injectors in commercial applications today [12]. They are designed to improve the atomisation process and to increase the turbulence levels within the combustion chamber for better mixing between the air and fuel. Numerous studies about spray processes have helped engineers to establish the criteria needed for the designing and developing more efficient combustion devices, whilst minimising the pollutant emissions [10,13,14]. The understanding of the complex nature of the fuel spray formed by high pressure injectors in experimental investigations is limited and this understanding can be significantly improved by numerical simulations. It can be stated, that the uncertainties arising from the experiments can be figured out by performing numerical

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Nomenclature

Roman	*Description (Unit)*
∇p	pressure gradient (Pa/m)
B	coefficient for break-up model (–)
$C^*_{\varepsilon 1}$	turbulence model coefficient (–)
$C^*_{\varepsilon 2}$	turbulence model coefficient (–)
C_1	turbulence model coefficient (–)
C_2	turbulence model coefficient (–)
c_D	drag coefficient (–)
c_T	turbulent dispersion force coefficient (–)
C_μ	turbulence model coefficient (–)
d	diameter of colliding droplet (m)
D	droplet diameter (m)
D_l	size of a break-up product (m)
D_n	size class diameter (m)
S_{ci}	Schmidt number
f	elliptic relaxation function (s^{-1})
$\mathbf{f}$	body force vector (N/m^3)
h	specific enthalpy (J/kg)
H_{kl}	enthalpy exchange term between phase k and l (W/m^3)
k	turbulence kinetic energy (m^2/s^2)
L	length scale (m)
L_A	atomization length scale (m)
L_T	turbulent length scale (M)
$\dot{m}_{E,k1}$	evaporated mass exchange of a single droplet (kg/s)
$\boldsymbol{M}_{D,k1}$	drag forces (N/m^3)
$\mathbf{M}_{kl}$	momentum exchange term between phase k and l (N/m^3)
$\boldsymbol{MT}_{,k1}$	turbulent dispersion forces (N/m^3)
N_{collis}	modelled number of interfacial droplet collisions (-)
N_k	droplet number density (m^{-3})
N_n	number of blobs (–)
P_k	turbulence kinetic energy production (m^2/s^3)
$\dot{Q}_{E,k1}$	heat flow rate into a single droplet (W)
$\boldsymbol{q}$	heat flux (W/m^2)
R_n	diameter (m)
R_t	target diameter (m)
S_{Y_i}	mass source term for the species i ($kg/(m^3s)$)
T	time scale (s)
t	time (s)
$\mathbf{v}$	velocity (m/s)
Δt	calculation time step (s)

Greek	*Description (Unit)*
Y_i	fuel mass fraction (–)
Λ_n	instability wavelength (m)
Ω_n	growth rate (1/s)
ε	turbulence dissipation rate (m^2/s^3)
ζ	velocity scale ratio (–)
θ	enthalpy volumetric source (W/kg)
μ	molecular viscosity (Pas)
μ_t	turbulent viscosity (Pas)
σ_k	turbulence model coefficient (–)
σ_ε	turbulence model coefficient (–)
σ_ζ	turbulence model coefficient (–)
$\boldsymbol{\tau}$	shear stress (N/m^2)
τ_A	overall atomization time scale (s)
τ_a	rate of primary atomization (s)
τ_T	turbulent time scale (s)
τ_W	aerodynamic time scale (s)
υ_t	eddy viscosity (m^2/s)
$\Gamma_{c,i}$	mass source of created droplet (kg/s)
$\Gamma_{c,k}$	mass source of the bigger droplet (kg/s)
$\Gamma_{c,l}$	mass source of the smaller droplet (kg/s)
Γ_{kl}	mass exchange term between phase k and l ($kg/(m^3s)$)
α	volume fraction (–)
ρ	density (kg/m^3)

Subscripts	Description
1	gas phase index
avg	average
Br	break-up index
C	collision index
E	evaporation index
k	phase index
n	bulk liquid phase index
P	primary break-up index
S	secondary break-up index

Superscripts	*Description*
t	turbulent index

simulations [15]. Numerical modelling of spray processes is a very challenging task compared to a single phase flow. The challenges arise due to the fluid interfaces between the phases and the property variations across these interfaces. Thus, the spray models demand complicated techniques for coupling the dynamics of the liquid droplets and the gas carrier. A variety of strategies have been formulated over past years in order to address this problem. In general, most of these strategies have fallen into two basic formulation methods that are commonly used for coupling the dynamics of the liquid and the gaseous phase: the Euler–Lagrangian method and the Euler–Eulerian method. The Euler–Lagrangian [16] method has been used by many researchers and various improvements to the basic scheme have been proposed [17–21]. Over recent years the Discrete Droplet Model (DDM) within the Euler–Lagrangian framework has dominated in predicting the behaviour of the spray process. In this method, the spray is represented by finite number of droplet groups, called droplet parcels. It is assumed that all the droplets within one parcel are similar in size and have the same physical properties. The motion and transport of each parcel is tracked through the flow field using the Lagrangian formulation, whilst the gaseous phase is described solving the conservation equations using the Eulerian formulation. The coupling between the liquid and the gaseous phase is taken into account by introducing appropriate source terms for interfacial mass, momentum and energy exchange [22]. Although various researchers and engineers have used the Euler–Lagrangian formulation as a numerical simulation tool for predicting the characteristics of complex multiphase droplet flows to guide their engineering devices designs, the concepts and applications have severe limitations. This formulation is very sensitive to the grid resolution in the near nozzle region [23] and reveals limitations in the descriptions of dense sprays. This assumes that the spray is sufficiently diluted; usually the discrete phase volume fractions should be less than 10%. It also shows statistical convergence problems, as discussed by [24,25]. Thus, the Euler–Lagrangian formulation is most often used to reliably describe sprays produced by low pressure atomisation [26]. Above mentioned difficulties could be overcome by a stronger physical coupling of the gaseous and liquid phases using the Euler–Eulerian formulation. This method treats the liquid phase and the gaseous phase as interpenetrating continua where both phases are treated from the Eulerian point of view. Hence, this method neglects the discrete nature of the dispersed phase and approximates its effects upon the continuous phase. The same discretisation, and similar numerical techniques and conservation equations are used for both phases. This method was first addressed by [27]. The Euler–Eulerian method has been adopted by a number