Fuel 199 (2017) 324–331

Contents lists available at ScienceDirect

Fuel

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

Full Length Article

Extended modeling of decelerating turbulent jets for diesel spray's penetration after end-of-injection

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HIGHLIGHTS

- Decrease of momentum flux is caused by injected fluid mass flow rate reducing.
- Injected fluid mass flow rate over the tip cross-section was formulated.
- Analytical expression of turbulent jet penetration was derived.
- A new analytical model of diesel spray penetration was developed theoretically.

ARTICLE INFO

Article history:
Received 25 October 2016
Received in revised form 4 January 2017
Accepted 28 February 2017
Available online 9 March 2017

Keywords:
Turbulent jets
Decelerating state
Analytical formulation of jet penetration
Diesel spray model
After the end-of-injection

ABSTRACT

In order to meet the growing demand for reduction of pollutant emissions and improvement in thermal efficiency, advanced combustion modes, such as low-temperature combustion (LTC), premixed charge compression ignition (PCCI), have been adopted in modern diesel engines. Compared to conventional diesel combustion, the ignition delay becomes longer in advanced combustion modes. The longer ignition delay usually results in the fuel injection ending earlier than ignition. Therefore, the spray propagation after end-of-injection (EOI) plays a significant role in diesel combustion process, and it is needed to estimate the spray evolution and mixture formation after EOI. For this purpose, a simple and analytical diesel spray model including the spray evolution after the EOI is developed in this study. To develop the model, the theoretical analysis on the decelerating process of the turbulent jet tip is performed based on the integral momentum flux and mass flow rate of the injected fluid over the cross-sectional area at the jet tip. It is observed that the decrease of mass flow rate of the injected fluid over the tip cross-sectional area causes reduction of the momentum flux over the tip cross-sectional area and the turbulent jet tip decelerating. Then the mass flow rate of injected fluid over the tip cross-sectional area is formulated, furthermore the analytical equation of the turbulent jet tip penetration during decelerating state is derived, and the correctness of the developed analytical equation has been proved theoretically. Finally, the calculation is extended to diesel spray penetration, and the calculated results are validated against the one-dimensional discrete model and the experimental data from Engine Combustion Network (ECN) respectively.

1. Introduction

In the recent two decades, increasingly stringent pollutants emission regulation has become the primary challenge for diesel engines. To reduce fuel consumption, NOx and PM emissions simultaneously, advanced combustion modes have been proposed, such as premixed charge compression ignition (PCCI), low temperature combustion (LTC). The main aim of the advanced combustion modes is to obtain more homogeneous and leaner mixture and suppress locally high combustion temperature. Normally, the early and late injection strategies combining with exhaust gas recirculation (EGR) are adopted to realize the advanced combustion modes [1–3]. In these ways, the injected fuel is mixed with low-temperature in-cylinder gas, thereby the ignition delay is increased leading to more homogeneous fuel-air mixture before ignition compared to conventional diesel combustion. Especially the advanced combustion modes are usually adopted to reduce the NOx emissions, so that aftertreatment can works efficiently under

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http://dx.doi.org/10.1016/j.fuel.2017.02.105
0016-2361/

low load operating conditions. The smaller injection amount and high-pressure fuel injection is responsible that the fuel injection ends much earlier than ignition. Meanwhile, with the high-pressure common-rail fuel injection system widely used on diesel engines, the multiple-injection has played a significant role in the latest diesel engines to extend the advanced combustion modes to high load operating conditions [4–6]. Injections of a small amount of fuel prior and after main injection are called the pilot and post injection respectively. Using pilot and post injection strategies, the short injection is carried out during compression process (pilot injection) or expansion process (post injection). The short injection duration and low in-cylinder temperature cause the end-of-injection (EOI) to be completed prior to the ignition, consequently this part of fuel is able to fulfill PCCI strategy. Thus, spray propagation after EOI contributes greatly to the mixture formation and subsequently combustion process in modern diesel engines. To control the combustion process of modern diesel engines, the diesel spray propagation and mixing characteristics after EOI must be well understood and clarified [7–9]. Currently, Wakuri's model [10], Siebers's model [11], Hiroyasu's model [12] and Desantes's model [13] are frequently used for parametric analysis of spray behavior. However, they only predict the spray propagation during injection in quasi-steady state and do not involve any information after EOI.

Musculus et al. [14] conducted the experimental research on non-combusting diesel jets in a single-cylinder heavy-duty optical diesel engine and a constant-volume combustion chamber with common-rail injection system. Compared to the entrainment of diesel spray during injection, the entrainment was observed to be enhanced in upstream region of the diesel spray after EOI. Moon et al. [15] also discovered that the strong perturbation induced entrained motion was generated near the nozzle tip just after EOI based on constant-vessel experiments, and the entrained motion was even stronger than that of quasi-steady state (during injection), which enhanced the evaporation and mixing. To analyze the phenomena of entrainment enhancement after EOI, Musculus and Kattke [16] developed a one-dimensional discrete model to describe the fuel mass flow rate and momentum flux along the axis of diesel jets. The results revealed that the deceleration state of diesel jet travels from the nozzle to the jet downstream with increased entrainment rate after the EOI, this phenomenon is called the entrainment wave. After the front of entrainment wave arrives at the diesel jet's tip, the whole diesel jet goes into deceleration state, and the jet penetration gradually coincides with the relation of fourth-root dependence on time [16]. In fact, the fundamental studies on turbulent jets had indicated that the turbulent jet behaviors during and after deceleration state are quite different from steady state. For example, based on the experimental research on the unsteady turbulent water jets, Sangras et al. [17] found that the jet penetration gradually transits from a square-root dependence on time to a fourth-root dependence on time after EOI. Although the one-dimensional discrete spray model developed by Musculus and Kattke [16] is able to predict the spray penetration after the arrival of the entrainment wave at the diesel jet's tip, the model cannot intuitively reveal the diesel spray penetration with the variation of time, injection parameters and ambient gas parameters. Additionally, because it is a one-dimensional discrete spray model, it is computationally very expensive to analyze and understand the spray penetration with varying parameters. Musculus [18] summarized the previous researches on the decelerating turbulent jets, and derived an one-dimensional analytical equations based on the momentum flux along the jet axis, which describes very well the penetration and entrainment rate of turbulent jets after EOI. This model can be modified to simulate the diesel spray penetration. However, after the entrainment wave front arriving at the head of jet, the analytical solution of the jet penetration was not obtained, numerical integration is needed to calculate the jet penetration. In our previous study [19], the volumetric flow rate of fuel during injection was assumed to be shared by fuel and air for the period after the entrainment wave arrives at the spray tip. In that way, the spray penetration during decelerating state can be calculated. Nevertheless, this calculation is based on the discrete method proposed by Musculus and Kattke [16], thereby the analytical derivation and theoretical understanding can be found elsewhere and will not be presented. Thus, to intuitively analyze and understand diesel spray penetration, it is needed to develop a simple analytical equation of the diesel spray penetration after the arrival of the entrainment wave at the diesel jet's tip.

In this study, a new formulation for the turbulent jets penetration after the arrival of entrainment wave at the tip of jet based on the momentum flux and mass flow rate over the tip cross-sectional area is derived. It can be used for general understanding of the turbulent jet propagation after EOI. The most important work focuses on the calculation of the momentum flux and injected fluid mass flow rate over the tip cross-sectional area. Then the analytical equation of turbulent jets penetration is extended to diesel spray penetration calculation. Finally, the validation is done by comparing the calculated results of the new diesel spray model with that of one-dimensional discrete diesel spray model and experimental data from Engine Combustion Network (ECN).

2. Derivation of turbulent jets propagation

2.1. Methodology

A simple and theoretical method to describe the turbulent jet spreading has been proposed using the momentum flux and mass flow rate over cross-sectional area along the jet axis [13,16,18]. In this study, the calculation is based at Musculus [18] work, but the calculations of momentum flux and mass flow rate focuses on the jet tip. Thus, the assumptions for the derivation are the same as those of Ref. [18], which can be briefly summarized as:

(1) The injected fluid density is assumed to be equal to that of ambient fluid.
(2) The turbulent jet is assumed as a non-vaporizing and incompressible flow.
(3) Turbulent viscous forces in the jets are neglected.
(4) The effects of molecular and turbulent diffusion are ignored.
(5) The net force due to any axial pressure gradient is assumed negligible.
(6) The turbulent jet has a conical shape with a constant spreading angle.
(7) The radial profile of mean axial velocity remains unchanged during and after injection.

The turbulent jet can be treated as a conical jet issuing from a round nozzle. The momentum flux ($\dot{M}_{\rm tip}$) and mass flow rate of injected fluid ($\dot{m}_{\rm j,tip}$) integrated over the tip cross-sectional area are derived as

$$\dot{M}_{\rm tip} = \rho\beta A_{\rm tip}\bar{u}_{\rm tip}^2, \tag{1}$$

$$\dot{m}_{\rm j,tip} = \rho\beta\bar{X}_{\rm j,tip}\bar{u}_{\rm tip}A_{\rm tip} \tag{2}$$

where ρ is the mass density of fluid both injected fluid and ambient fluid according to assumption (1), β is a factor used to consider the radial profile of axial velocity and volume fraction of injected fluid. In addition, $A_{\rm tip}$ is the area of tip cross-sectional area, $\bar{X}_{\rm j,tip}$ and $\bar{u}_{\rm tip}$ is the average volume fraction of injected fluid and average velocity over the tip cross-sectional area respectively.