NUMERICAL INVESTIGATION OF THE EFFECT OF FUEL INJECTION MODE ON SPRAY/WALL INTERACTION AND EMISSION FORMATION IN A DIRECT INJECTION DIESEL ENGINE AT FULL LOAD STATE

by

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The effect of three injection rate modes on fuel spray-wall impingement and combustion process in a direct injection diesel engine is investigated in present work. A three-dimensional computational fluid dynamics model for flow field, spray, spray-wall interactions, combustion, and emission formation processes have been used to carry out the computations. The optimized omega combustion chamber geometry was used in the diesel engine model instead of baseline cylindrical geometry and the results were verified for this improved combustion chamber geometry. Results for different injection modes indicate that using the ramp injection rate curve, the spray-wall impinging is increased due to higher injection rate at the end of injection duration. Also the increased in-cylinder temperature, piston surface temperature and higher turbulence intensity leads to enhanced wall-film evaporation. Soot mass fraction also decreases due to improved air-fuel mixing and evaporation of wall-film by reduction of the fuel rich zones especially in the impingement regions. Results for different injection modes indicate that using the ramp injection rate shape slightly retards the combustion process and improves combustion characteristics while maintaining lower NO_x *and considerably lower soot emissions compared to the boot and rectangle injection modes. Also, in this injection mode, because of the high pressure injection, higher spray droplet velocities (higher Weber number) and increased wall spray height than the other modes, it could be said that the dominant impingement regime may be the splashing regime. The results of model for baseline diesel engine are compared with the corresponding experimental data and show good levels of agreement.*

Key words: *injection mode, diesel engine, computational fluid dynamics, spray-wall impingement, combustion, spray, emission*

Introduction

Internal combustion engines have proved very practical and useful due to the high efficiency and considerable power to weight ratio and volume and are widely used in many fields such as transportation, automotive industry, *etc*. Nowadays, the crisis of depletion of energy re-

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sources and environmental dilemmas, a lot of researches are being done to improve thermal efficiency and to control and reduce emission trade-off on such engines. There has been a lot of works done in the field of engine research using both thermodynamic methods and computer aided modeling and simulations. With the progressive development of computational fluid dynamics (CFD) and flow field models, it is possible to study and predict droplet distribution, temperature, pressure, and other parameters at every desired point and time within the combustion chamber [1].

Flow field models, such as KIVA II code in 1990 [2, 3] and numerical Ricardo code [4] in 1992, are premiere examples. Several works have been reported in the literature corresponding the injection strategies, injection mode and shaping, and also spray-wall interactions. The 3-D KIVA 3 was used by Patterson *et al.* to study the effects of injection pressure and split injections on diesel engine performance and soot and NO_x emissions. The results show the use of the updated version of KIVA gives good agreement between measured and predicted engine cylinder pressures and heat release data for single injection cases [5]. Experiments were conducted by Henein *et al*. [6] to investigate the characteristics of a common rail fuel injection system using a flow rate test rig and a single cylinder research diesel engine. Different injection modes were investigated including main injection, main-post injection and pilot-main injection. The analysis indicated that the common rail fuel pressure affects all the injection parameters including the start of fuel delivery, its duration and amount under all modes of injection. Naoki *et al*. [7] proposed the application of dual mode diesel engine by premixed diesel combustion at low loads which was achieved by early injection and conventional diesel combustion at high loads. The results indicated that NO_x and smoke could be decreased by this method. The effect of injection rate shaping has been investigated by Rottmann *et al*. [8] on small bore direct ignition (DI) diesel engines. The results show that the ramp mode retards the combustion and reduces NO_x mass fractions which are also verified by the present work for the studied heavy-duty DI diesel engine. The effect of fuel injection rate has also been experimentally investigated in early works by Nishimura [9] in 1998 and it was found that injection pattern represents an important factor in emission reduction and combustion control. As for the spray-wall interactions, detailed investigations of wall-spray development are reported in Mohammadi *et al.* [10] and Allocca *et al.* [11]. Also, in the previous work by the authors [12], the effect of spray impingement on combustion process and emission formation was investigated at various speeds by the same model. It was shown that an increase in engine speed leads to increased spray impinging and average wall temperature in cylinder. Also at higher engine speeds, combustion was delayed and fuel was consumed in a shorter time period by the enhanced air and fuel mixing. The shorter combustion duration provided less available time for soot and NO_x formation.

At the present work, the 3-D CFD code AVL FIRE 8.31 has been used to study the effect of injection modes on wall-film formation, combustion process and emission in the OM-355 diesel engine with the optimized combustion chamber geometry. This paper also demonstrates the usefulness of multidimensional models to gain insight into the spray-wall, spray-flow filed, and turbulence-mixing interactions and to provide direction for exploring new engine concepts about combustion process and emission.

Model description

The numerical model for heavy duty OM-355 diesel engine with the specifications on tab. 1 is carried out using AVL Fire code. The Calculations are carried out on the closed system from intake valve closure at –118 °CA to exhaust valve open at 120 °CA. Initial pressure is set to 86 kPa and initial temperature is calculated by using of experimental air-fuel ratio data in the engine. Considering the symmetry of the model, problem was only solved for a quarter section of the whole geometry. An analysis was carried out on the effect of combustion chamber geometry prior to main calculations and results indicated that using the omega piston cusp improved the combustion parameters compared to the baseline cylindrical piston head. Therefore the results for the rest of calculations are presented for the omega combustion chamber type in this paper. Figure 1 shows the numerical grid. The turbulent flow within the combustion chamber is simulated using the RNG κ-ε turbulence model, modified for variable-density engine flows. Combustion process is modeled by eddy breakup model. This model assumes that in premixed turbulent flames, the reactants (fuel and oxygen) are contained in the same eddies and are separated from eddies containing hot combustion products. The rate of dissipation of these eddies determines the rate of combustion according to eq. (1):

Table 1. Engine specifications (OM-355 Mercedes Benz)

Engine type	Heavy duty DI diesel engine
Injector	Centric/4 holes
Injection period	From –18 °CA up to 0 °CA
Engine speed at max torque	1400 rpm
Engine speed at max power	2200 rpm
Piston diameter* stroke	150 128 mm
Cylinder volume	11.58 L
Peak of injection pressure	500 bar
Max output power	240 hp
Max outlet torque	824 Nm
Number of cylinders	6, vertical type
Compression ratio	16.1:1
Initial injection pressure	195 bar
Swirl/tumble	Quiescent

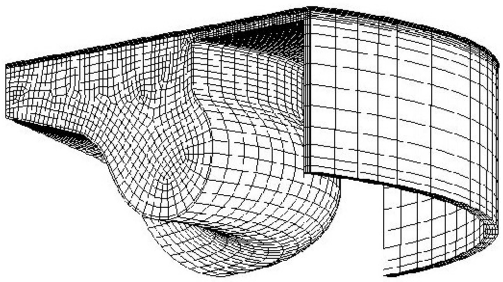


Figure 1. Outline of the computational mesh

$$\overline{\rho \dot{r}_{\mathrm{fu}}} \quad \frac{C_{\mathrm{fu}}}{\tau_{\mathrm{R}}} \bar{\rho} \min. \quad \bar{y}_{\mathrm{fu}}, \frac{\bar{y}_{\mathrm{ox}}}{S}, \frac{C_{\mathrm{pr}} \bar{y}_{\mathrm{pr}}}{1 \quad S} \tag{1}$$

The first two terms of the "minimum value of" operator determine whether fuel or oxygen is present in limiting quantity, and the third term is a reaction probability which ensures that the flame is not spread in the absence of hot products. Above equation includes three constant coefficients (C_{fu}, τ_{R}, C_{pr}) and C_{fu} varies from 3 to 25 in diesel engines. An optimum value was selected according to experimental data [13]. Spray breakup and droplet distribution is modeled by advanced Wave Standard [14]. In this model the growth of an initial perturbation on a liquid surface is linked to its wave length and to other physical and dynamic parameters of the injected fuel and the domain fluid.

Droplet parcels are injected with characteristic size equal to the nozzle exit diameter (blob injection). The Dukowicz model was applied for treating the heat-up and evaporation of the droplets. This model assumes a uniform droplet temperature. In addition, the rate of droplet temperature change is determined by the heat balance, which states that the heat convection from the gas to the droplet either heats up the droplet or supplies heat for vaporization. NO_x formation is modeled by the Zeldovich mechanism while, soot formation is modeled by Kennedy, Hiroyasu, and Magnussen mechanism [15].

Wall-film model

Due to imperfect atomization and evaporation, a portion of the injected spray droplets impact on the walls of the combustion chamber and in special conditions, lead to formation of a wallfilm. This influences the combustion process and consequently the production of emissions, as an incomplete combustion in the vicinity of the wall will result in high HC emissions and soot particles. The spray-wall interaction model used in the simulations was based on the spray-wall impingement model described in [14-16]. At this model, behavior of the impinging droplet is influenced by major factors like wall temperature and Weber number which identifies a number of impingement regimes. In principle, higher surface temperatures prevent the formation of wall film due to rapid boiling or due to droplet re-bounding because of the Leiden frost phenomenon. To gain insight into this fact, a code was written and integrated with the AVL Fire solver to calculate the mean cylinder surface temperature at desired crank angle. There are four spray-wall interaction regimes depending on the inlet droplet velocity. At very low inlet velocities the droplet sticks to the wall or to the wall film. When the inlet velocity increases a vapor or gas boundary layer is trapped underneath the droplet and causes the liquid to rebound. During the rebound, parts of the kinetic energy are dissipated and the outgoing normal velocity is usually lower than the incoming normal velocity. A further increase of the velocity leads either to the spread or the splash regime. In the spread regime the complete liquid spreads along the wall with hardly any normal velocity. In the splash regime a part of the liquid remains near the surface and the rest of it is reflected and broken up into secondary droplets. Figure 2 presents the reflection of a parcel containing droplets at the wall. The wall film module takes care of the splashing regime of impinging droplets.

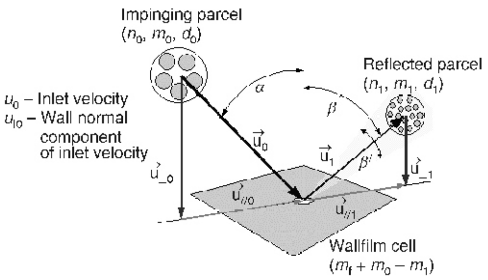


Figure 2. Wall interaction of droplets

According to correlations by Mundo *et al.* [17, 18], the predominant influence of droplet momentum and properties as viscosity and surface tension is taken into consideration by introducing the dimensionless groups Reynolds number and Ohnesorge number for the particular droplet according to eqs. (2) and (3):

$$\mathrm{Re}_{\mathrm{D}} \quad \frac{\rho d_0 u_{\ 0}}{\mu} \tag{2}$$

$$\mathrm{Oh} \quad \frac{\mu}{\sqrt{\rho \sigma d_0}} \tag{3}$$

The Reynolds number compares momentum to viscous forces, whereas the Ohnesorge number relates viscous forces to surface tension. For the Reynolds number, only the wall normal component $u_{\;0}$ of the initial droplet velocity u_0 is used, which accounts for impact angle effects. In addition to the Weber number, a K-value [17] is defined which is modified form of a Weber number is according to eqs. (4) and (5):

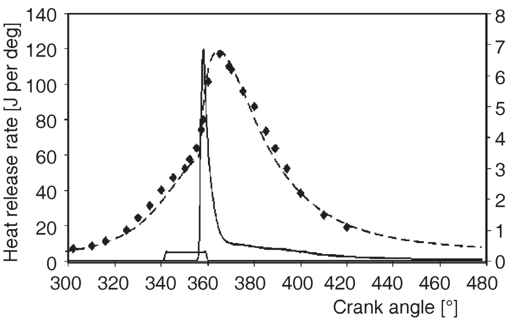
$$\mathrm{We} = (\mathrm{Re_D Oh})^2 \tag{4}$$

$$K \quad \mathrm{Oh\,Re}_\mathrm{D}^{1.25} \tag{5}$$

This K-value is a characteristic quantity which serves to distinguish between different impingement regimes and generally is used as the key parameter for the splashing model.

Results and discussion

Figure 3 show the computed and measured in cylinder pressures and exhaust emission for OM-355 DI baseline diesel engine [13]. The operating condition is 1400 rpm and full load state. As can be seen, there is a good agreement between the measured data and the model prediction. The good agreement between the measured and computed results for this engine operating condition (maximum emission) gives confidence in the model predictions, and suggests that the model may be used to explore new engine concepts about effects of spray-wall interaction on the emission and combustion process at various injection modes.



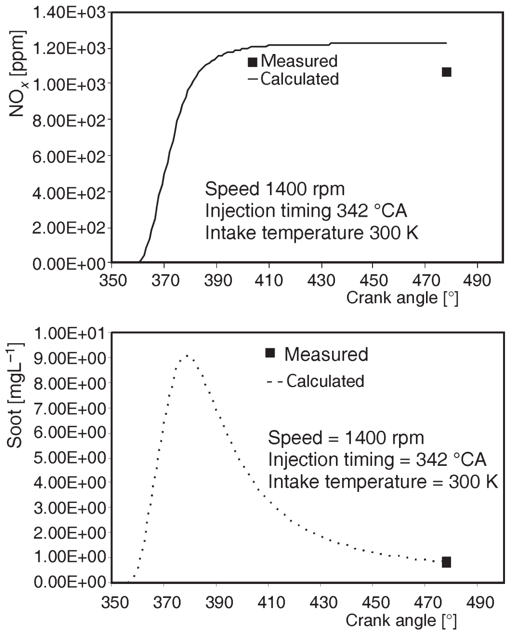


Figure 3. Comparison of calculated and measured [13] in-cylinder pressure and emission data for baseline diesel engine at 1400 rpm

The predicted results are presented in this section for the three injection rate modes considered. Figure 4 shows the boot, ramp, and rectangle injection mode curves that which were applied at the injection period. The amount of injection rates are dimensionless values which Fire code uses to create injection rate profiles. These values are modified such that the

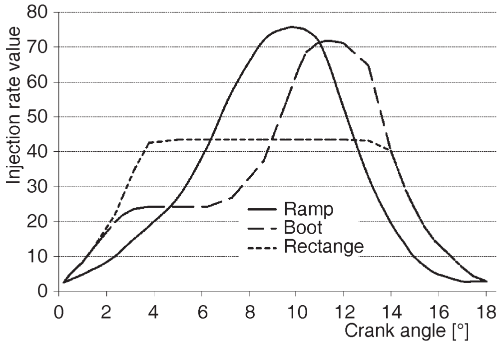


Figure 4. Injection mode curves

area under the curve for all cases is equal. The CFD code projects these curves on the injection period specified to ensure correct total fuel mass is injected in the cycle.

Figure 5 shows the predicted pressure, heat release rate and emission data for the three injection mode cases. According to the figure, the rectangle injection rate results in highest peak pressure which leads to increased mean in-cylinder temperature and thus increasing the NO_x mass fractions (figs. 5, 8, and 11). Due to injection of more fuel at the early parts of the injec-

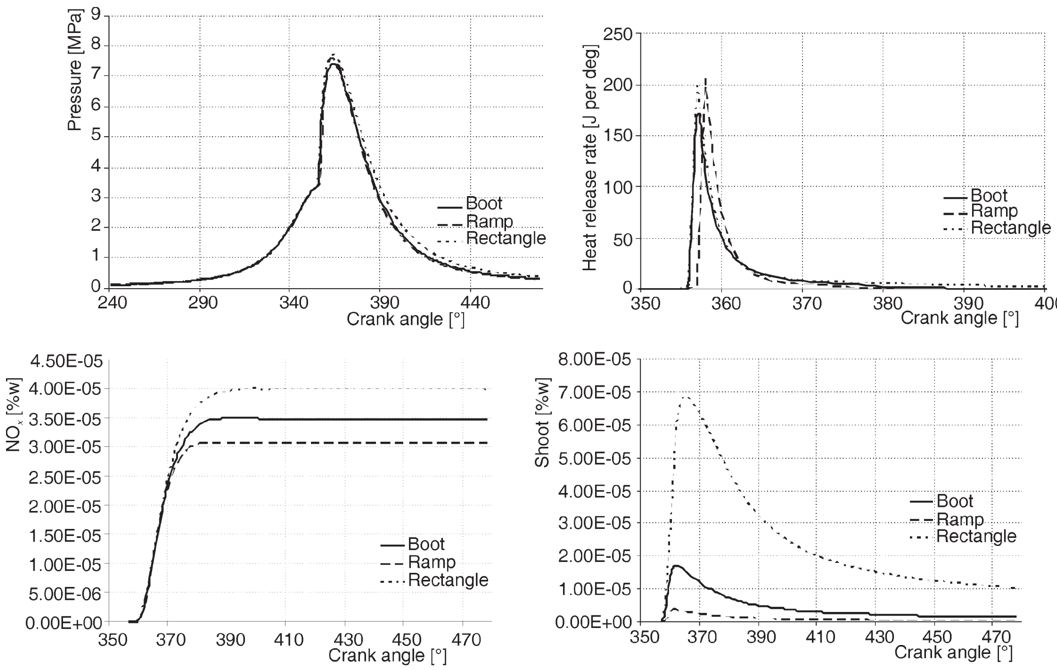


Figure 5. Mean in-cylinder pressure, heat release rate, NO_x, and soot mass fractions for different injection modes

tion period, low mixing and turbulence intensity (see also fig. 6) in the rectangle mode, soot formation is increased considerably comparing with boot and ramp modes. Meanwhile, the ramp injection mode has retarded the combustion thus decreasing the NO_x mass fraction. Because of the fact that the larger portion of fuel is injected later in the injection duration in the ramp mode and due to improved air-fuel mixing (see also fig. 5), the soot mass fraction is considerably decreases at this case compared to the boot and rectangle modes. The boot mode exhibits average results among the three types.

At the end of ramp injection mode high injection pressures are necessary in order to inject the complete fuel mass in the end of injection time. Furthermore, high injection pressures are now advantageous, because they enhance mixture formation and support the reduction of soot at later crank angles. Altogether, a simultaneous reduction of soot and NO_x can be achieved, fig. 5. Also, in the previous work by the authors [19], the effect of split injection on combustion process and emission was investigated at full load state. It was shown that split injection is one of other powerful tools that make the chance to shift the trade-off curve (soot and NO_x) closer to origin.

It is clear that spray wall impingement and the formation of liquid films may have an important effect on spray atomization and mixture formation. In the case of film formation on a hot wall, which can happen, for example, in a small-bore or high pressure injection direct injection diesel engine (ramp mode) if the spray impinges on the hot piston surface, its evaporation strongly influences the mixture formation process in the near wall region and must be studied by CFD models.

Figure 6 shows the spray-wall impingement data and the piston wall temperature for the three injection modes. The impinged mass tends to increase in the ramp mode due to larger

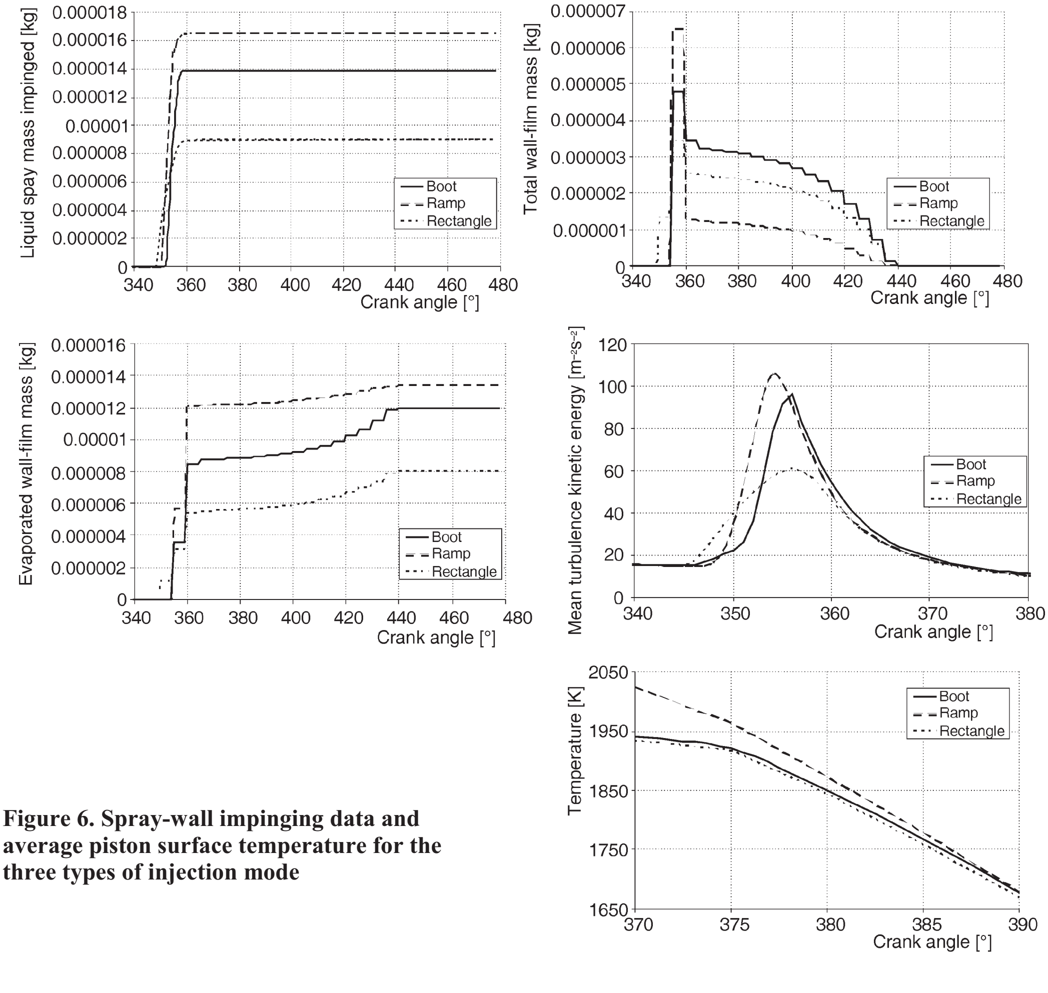


Figure 6. Spray-wall impinging data and average piston surface temperature for the three types of injection mode

injection rates at this case. However, due to increased pressure and heat release rate for the delayed combustion and increased piston surface temperature the wall-film evaporation is increased in this mode and this result in decreasing the soot mass fractions at the impinging zones which are primary zones for soot formation according to data in literature (see figs. 9 and 10).

In the previous work by the authors [12], the effect of spray impingement was also investigated on combustion process and emission formation at various speeds by the same multi-dimensional model. To establish a reference to the results of spray impingement data for the previous work which is relevant to this paper, fig. 7 indicates the increase in engine speed shortens the available time for the spray evaporating and leads to more impinged mass and high wall temperature. Also at high engine speed, because of high temperature in cylinder and low heat loss, piston surface temperature will increase. Furthermore, contact with a hot wall at high engine speeds intensifies wall-film evaporation.

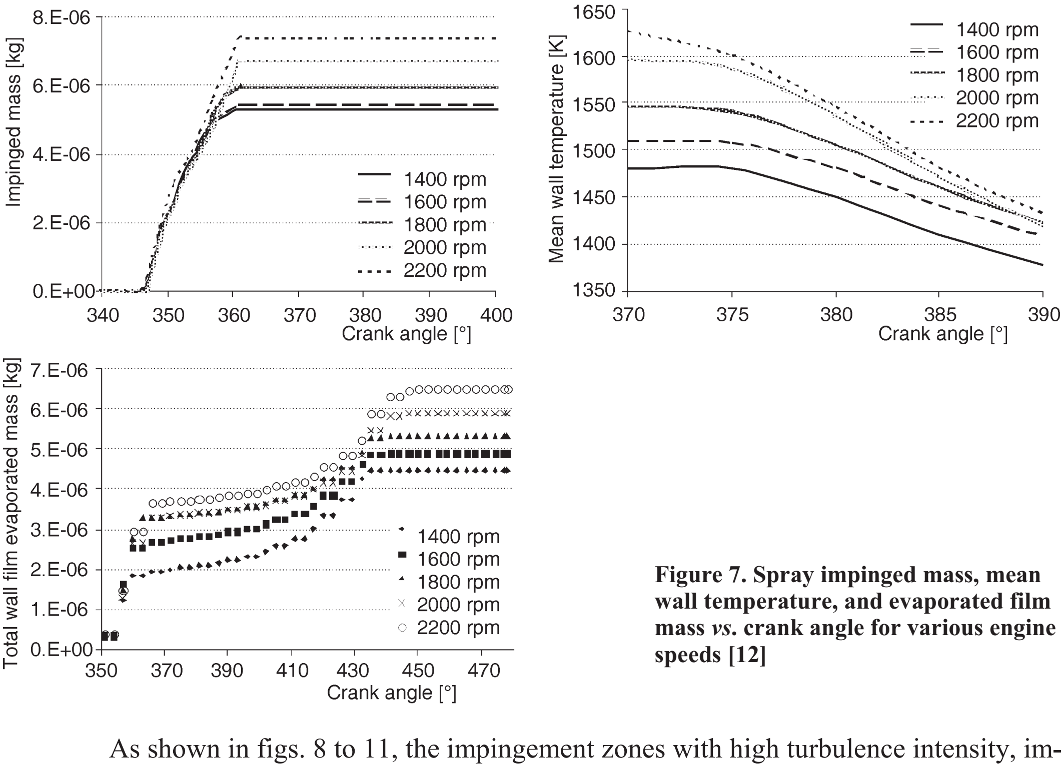


Figure 7. Spray impinged mass, mean wall temperature, and evaporated film mass *vs.* crank angle for various engine speeds [12]

As shown in figs. 8 to 11, the impingement zones with high turbulence intensity, improved mixture formation and the higher temperature than 2000 K is the NO_x formation area. In addition, the area with high equivalence ratio, low turbulence intensity and the temperature approximately between 1600 K and 2000 K is the soot formation area. A local soot-NO_x trade-off is evident in these contour plots, as the NO_x and soot formation occur on opposite sides of the high temperature region. The prediction of wall spray height (fig. 9) at three states captures the trend that an increase of injection pressure (ramp mode) results in larger values of wall spray height. The droplets in the wall spray get larger tangential velocities due to the increased gas velocities along the wall, and larger normal velocities because of a more intense splashing.

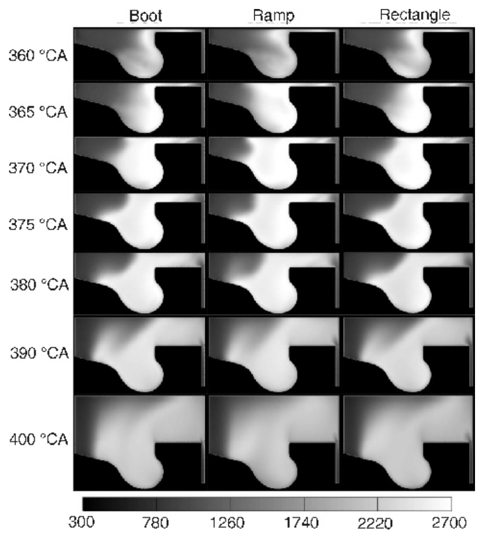


Figure 8. Plots of temperature in cylinder at three injection mode over various crank angle

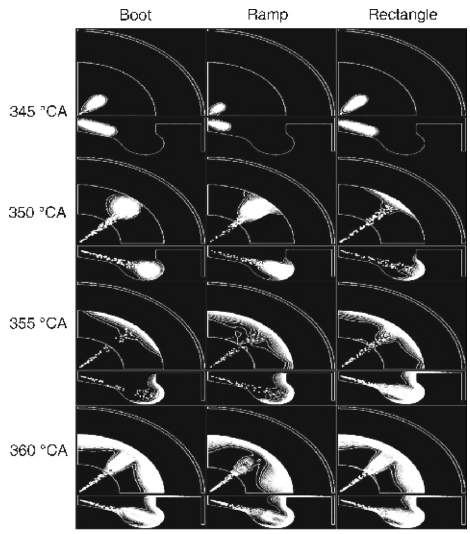


Figure 9. Plots of fuel spray, vapor, and wall impingement at three injection mode over various crank angle

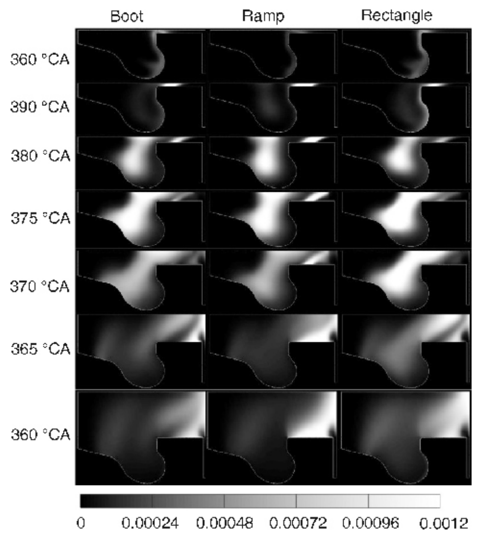


Figure 10. Plots of soot mass fraction in cylinder at three injection mode over various crank angle

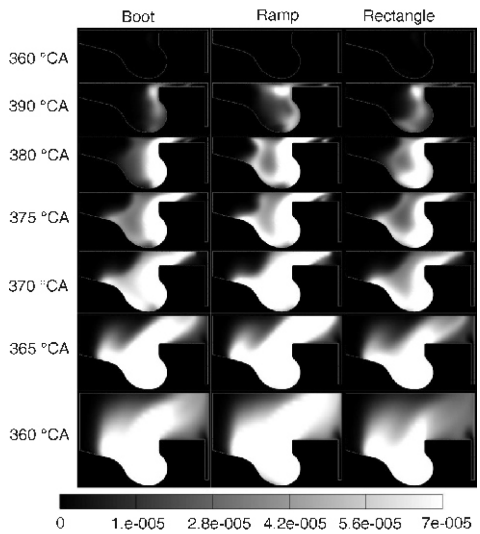


Figure 11. Plots of NO_x mass fraction in cylinder at three injection mode over various crank angle

Summery and conclusions

This paper was dedicated to study the effect of injection rate mode on impingement of spray and the combustion process in a DI diesel engine using AVL Fire 8.31 code and the following results were obtained:

Results of model for in cylinder pressure and exhaust emission at 1400 rpm and full load state (maximum emission point) was compared with the corresponding experimental data and show good agreement. Such verification between the measured and computed result for this operating point gives confidence in the model prediction, and suggest that the model may be used to explore new engine concepts.

The ramp injection mode has shown to retard the combustion process and reduce the NO_x mass fraction due to decreased in-cylinder temperature, also the soot mass fraction decreases considerably at this case because of high pressure injection and high turbulence intensity.

The rectangle injection mode has highest peak pressure and temperature and increases the NO_x and soot mass fractions compared to other cases.

The use of different injection rate curves affects the combustion related properties and can be used as an alternative way to control combustion and emission in the diesel engines.

Spray impinging was increased in the ramp mode due to larger injection rates and high injection pressure.

At the ramp mode, due to increased turbulence intensity and average piston surface temperature, the wall film evaporation is intensified and mixture formation is optimized which resulted in considerable decrease in soot mass fractions.

Nomenclature

C – empirical coefficient
d_o – parcel diameter
m – mass, [kg]
$\dot{r}$ – fuel consumption rate, [kgs^{-1}]
S – stoichiometric oxygen requirement; source term
$\bar{y}$ – mass fraction

Greek letters

μ – viscosity
$\dot{\rho}$ – density, [kgm^{-3}]
σ – surface tension
τ_R – turbulent mixing time scale

Subscripts

fu – fuel
ox – oxidizer
pr – product
R – reaction

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