

Numerical Simulation of Diesel Sprays under Hot Bomb Conditions

Ishtiaq A. Chaudhry, Zia R Tahir, F. A. Siddiqui, F. Noor, M. J. Rashid

Abstract—It has experimentally been proved that the performance of compression ignition (C.I.) engine is spray characteristics related. In modern diesel engine the spray formation and the eventual combustion process are the vital processes that offer more challenges towards enhancing the engine performance. In the present work the numerical simulation has been carried out for evaporating diesel sprays using Fluent software. For computational fluid dynamics simulation “Meshing” is done using Gambit software before transmitting it into Fluent. The simulation is carried out using hot bomb conditions under varying chamber conditions such as gas pressure, nozzle diameter and fuel injection pressure. For comparison purpose, the numerical simulations the chamber conditions were kept the same as that of the experimental data. At varying chamber conditions the spray penetration rates are compared with the existing experimental results.

Keywords—Evaporating diesel sprays, Penetration rates, Hot bomb conditions.

I. INTRODUCTION

IN order to catch up the ecological obligations for greener environment the impact of air pollution caused by our own vehicles remain under considerable scrutiny. Over the last three decades or so there has been ever increasing concern over the composition of the fuel and especially the combustion by-products. Exhaust emissions as they are generally known are the unwanted by-products of fuel combustion. In this regard firstly the petrol engines were put under greater concern since in their original form, these machines, excluding smoke or particulates, were found to produce significantly higher emissions than compression ignition (diesel) engines. Much directed research and subsequent development on emission controls on petrol engines has had reduced the undesirable elements from the exhaust gases to levels below that of their counterpart typical C.I. engines [1].

A major concern with the increasing popularity of diesel-powered vehicles is the resultant increased level of pollution since the diesel engines are a source of two major pollutants; nitrogen oxides (NO_x) and particulate matter, which both have an undesirable effect on public health and the environment [2]. They are also a source of carbon dioxide, one of the most important green house gases. Nitrogen oxides contribute towards acid rain and ground-level ozone, whilst particulate

emissions constitute a major health hazard.

This is a growing problem as more stringent legislation on emissions is introduced. To comply with such legislation, more effective and environmentally-friendly combustion systems need to be designed and manufactured. There are several ways of tackling this problem. Some engine designers favor exhaust gas after-treatment, whilst others prefer the introduction of more sophisticated fuel-injection systems. In some cases, both technologies are applied side by side. However, the most beneficial way of reducing emissions is at source, and to develop fuel-injection systems capable of meeting the requirements over the complete range of engine operating conditions. Diesel fuel-injection equipment is known to lend itself to the control and quality of the emerging spray. Engine load conditions have a direct influence on the combustion process, which is dependent upon the quantity, the quality and the timing of the fuel spray emerging from the injector nozzle.

Numerous experimental investigations have been directed into understanding the spray formation and its penetration in the combustion chamber. Liquid spray formation concern a lot of physics, starting from breakup of the liquid core into droplets short after the nozzle exit, called primary breakup. In a second stage the formed droplets breakup into smaller droplets, called secondary breakup. In automotive applications, with high ambient pressures and temperatures, the fuel droplets evaporate during their path until the liquid length is reached. From then on the evaporated fuel penetrates further into the surrounding gas, and at some point the spray auto-ignites.

For multiple-injections, it was found that the injected mass of the first of the split was approximately 19% less than that of the single injection strategy for the same injection duration. The second split reduction was less than 4% in comparison to the single injection strategy [3]. In this work the CFD modeling of the diesel fuel sprays is carried out to understand the formation and propagation of the spray under the given chamber conditions. The purpose is to investigate the fuel spray formation so that an efficient combustion process could be achieved. An efficient combustion process is not only essential for proficient diesel engine performance but it is also necessary ecologically.

II. EMPIRICAL CORRELATIONS FOR EVAPORATING SPRAYS

For evaporating sprays, [4] defined the break up point as point of intersection for initial near nozzle tip zone and final spray tip zone respectively.

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$$X_{po} = C_1 \left(\frac{2\Delta p}{\rho_f} \right)^{0.5} t \quad (1)$$

$$X_{po} = C_2 \left(\frac{\rho_g}{\rho_a} \right)^{0.25} \left(\frac{\Delta P}{\rho_a} \right)^{0.25} d^{0.5} t^{0.5} \quad (2)$$

$C_1 = 0.39$, with minor variation in the third decimal place, gives an ideal fit to the experimental spray length and $C_2 = 3.8$ and that maintains the uniformity for conversion of (2) into the non-evaporating case.

$$t_b = \left(\frac{C_2}{\sqrt{2} C_1} \right)^2 \left(\frac{\rho_g}{\rho_a} \right)^{0.5} \left(\frac{d \rho_f}{(\Delta P \rho_a)^{0.5}} \right) \quad (3)$$

For the specific values of T_a and T_g , the correlation for the evaporating jet break up time becomes

$$t_b = 0.605 \left(\frac{C_2}{\sqrt{2} C_1} \right)^2 \left(\frac{d \rho_f}{(\Delta P \rho_a)^{0.5}} \right) \quad (4)$$

C_1 and C_2 relate to the straight-line fit and the $t^{1/2}$ type fit, respectively. Now for the jet break up time and break up length the combination of constants $C_2 = 3.9$, and $C_1 = 0.40$; or $C_2 = 3.8$, and $C_1 = 0.39$, yield the following dimensionally consistent correlations

$$t_b = 28.7 \left(\frac{d \rho_f}{(\Delta P \rho_a)^{0.5}} \right) \quad (5)$$

$$X_b = 0.39 \left(\frac{2\Delta p}{\rho_f} \right)^{0.5} t_b \quad (6)$$

Alternatively by taking C_2 as 3.8 and keeping the nozzle coefficient C_1 as variable, to accommodate variable value of the coefficient of discharge of the nozzle the generalized form of (3) gives

$$t_b = \frac{4.346}{C_1^2} \left(\frac{d \rho_f}{(\Delta P \rho_a)^{0.5}} \right) \quad (7)$$

Mirza et al. [5] reported the variation in fuel line pressure with time as shown in Fig. 1. The dotted line shown in Fig. 2 for non-evaporating case is prediction of the correlation 2 for the reference case of [5]. This prediction, however suffers from the disadvantage of over prediction of the initial liquid phase and hence in agreement with the findings of [6] and [7].

The initial liquid phase is shown in Fig. 3 where the straight line follows the correlation equation (1). The jet break up time is plotted against the density ratio (ρ_a / ρ_g) in Fig. 4 where the solid line is the prediction of the correlation of Hiroyasu and Arai [8] and the dotted line represents the small variation due to disparity in the density ratio.

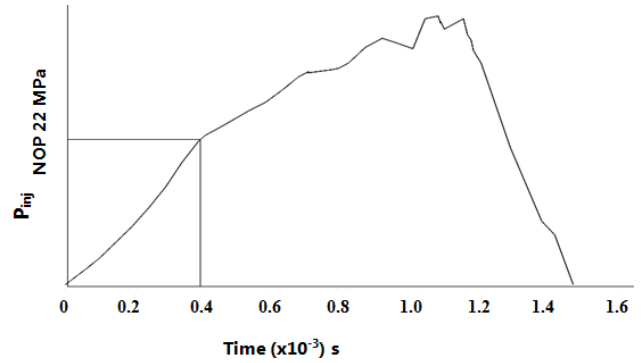


Fig. 1 Fuel line pressure diagram

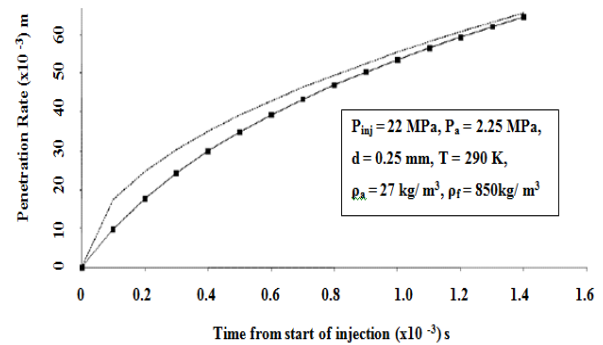


Fig. 2 Distance-Time history of fuel spray

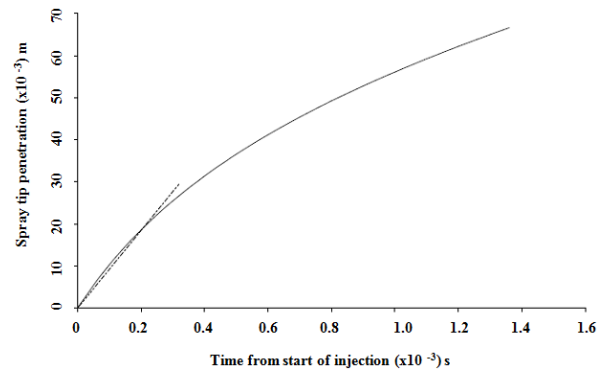


Fig. 3 2-Line fit to spray penetration history

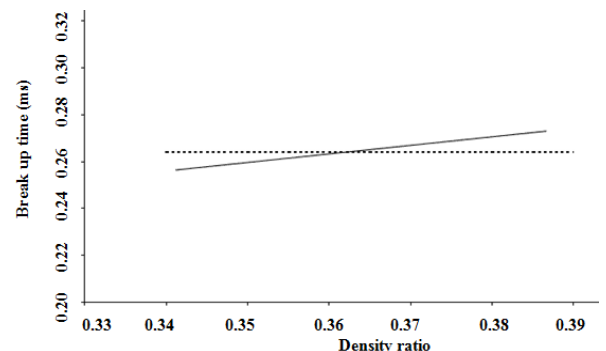


Fig. 4 Jet breakup time and density ratio

III. TEST CONDITIONS AND GEOMETRY

For CFD simulation of diesel fuel sprays the WAVE model

of Reitz [9] is used. The numerical results are compared with the experimental findings of [4] who performed experiments on sprays produced by a commercial Bosch distribution type pump and a Fiat single hole 0.25 mm diameter orifice type nozzle. For hot bomb conditions, the mean fuel injection pressure and the chamber pressure were kept at 22 MPa and 2.25 MPa respectively and the chamber temperature was maintained at 800 K. These conditions are taken to check validity of the model under chamber conditions closest to the real engine case. To investigate the behavior of the sprays at varying conditions some of the variables are changed, for instance fuel injection pressure, injector nozzle diameter, chamber pressure, chamber air/gas temperature.

The published correlations of [5], [10] are chosen to be compared with the numerical simulations. Some other correlations are also available in the published work having inherited discrepancies as described earlier. A time step of 1.36 ms is taken during the injection period for the sake of simplicity and the spray tip penetration rates are recorded for the purpose of plotting the results. Physical properties of the diesel fuel and that of air, used in the present work, are listed in Table I.

TABLE I
 FUEL AND AIR PROPERTIES

Property	Fuel	Air
Density (kg/m ³)	850	27.03
Viscosity (kg/m.s)	0.00332	1.789 x10 ⁻⁰⁵
Surface tension (N/m)	0.0190355	

In the present work, a 1600 cc, 4-cylinder, 4-stroke compression ignition engine is taken as the reference. The bore length to diameter ratio is taken as 1.1 such that each cylinder has a swept volume of 400 cc. The origin is the center of the base geometrical figure with its specified radius and the direction of its axis on which the cylinder is constructed. Height of cylinder is defined as the distance between the top and bottom circular faces of the cylinder.

IV. RESULTS AND DISCUSSION

Fig. 5 shows the comparison of the simulated and correlation spray tip penetration and Fig. 6 shows the spray tip velocity histories. Both the plots describe the evaporating diesel sprays under elevated chamber air temperature of 800K. When these results are compared to non-evaporating cases it shows a reduction in penetration rates of the evaporating spray tip of approximately 22%.

Fig. 7 shows the comparison of simulated and correlation results for spray tip penetration histories for different values of chamber pressure. Chamber air temperature is elevated to 800 K but maintaining other conditions as unchanged. It seems that the penetration is far more enhanced at decreased chamber pressure. When the pressure is increased, it offers a lot more resistance to the spray penetration and subsequently its velocity is reduced. In Fig. 8 both the simulated and correlation histories are plotted against each other to reveal the local disparities at each given value. It appears that the

simulated results approach the correlation histories with reasonable accuracy.

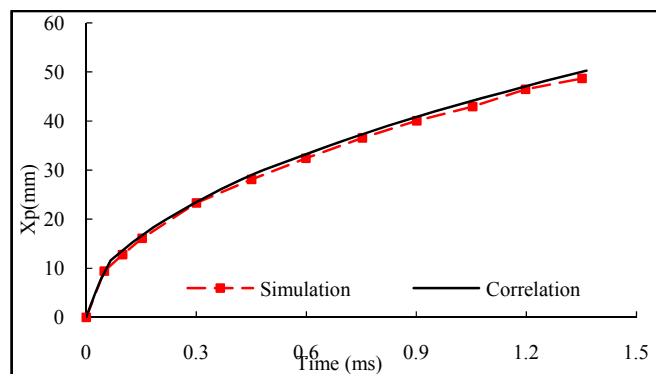


Fig. 5 Spray tip penetration vs time

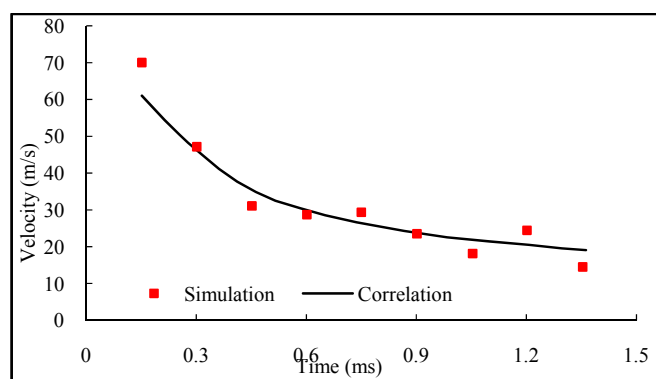


Fig. 6 Penetration velocity (Vp) vs time

Along with the chamber pressure the injection pressure is also vital parameter for spray penetration rate. Fig. 9 demonstrates the variation of spray penetration rates with the change of injection pressure. At elevated injection pressure, a far more enhanced penetration rate is achieved. Fig. 10 shows the comparison of the correlation predictions and the simulated penetration rates of the sprays. Fig. 11 is plotted for the spray tip penetration history and Fig. 12 is plotted for the comparison of the correlation prediction and the simulation results, for three different values of injector nozzle diameter but maintaining the other parameters as unchanged.

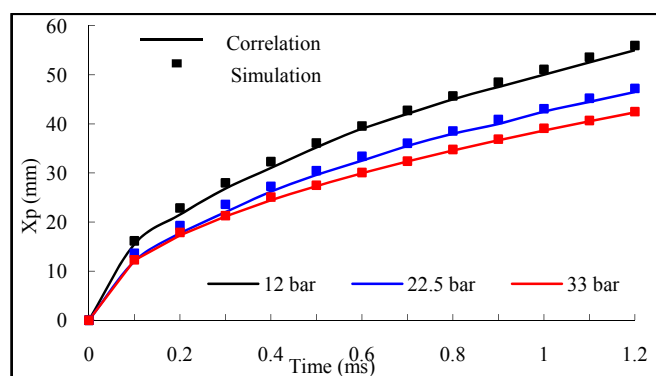


Fig. 7 Spray tip penetration vs time, $P_{inj} = 22 \text{ MPa}$, $d = 0.25 \text{ mm}$, P_{ch} varying

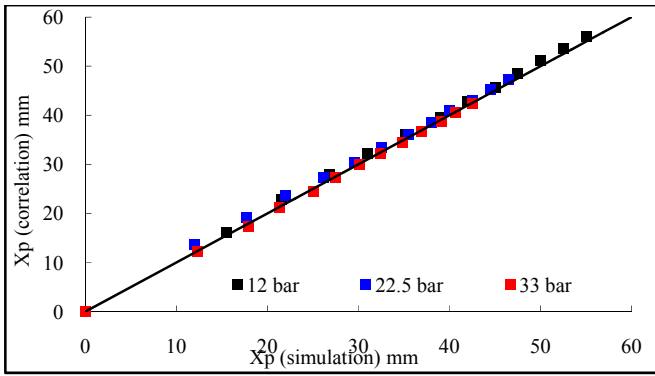


Fig. 8 Penetration (correlation vs simulation)

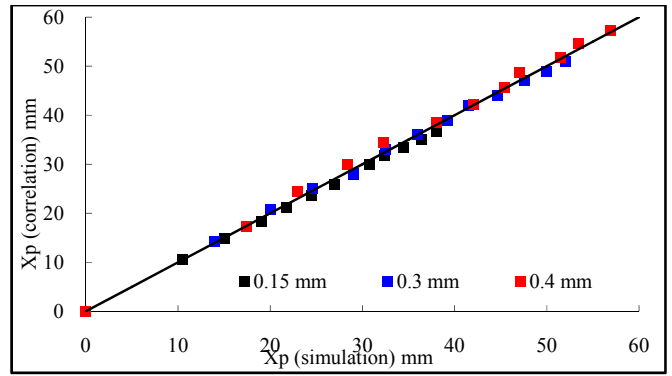


Fig. 12 Penetration (correlation vs simulation)

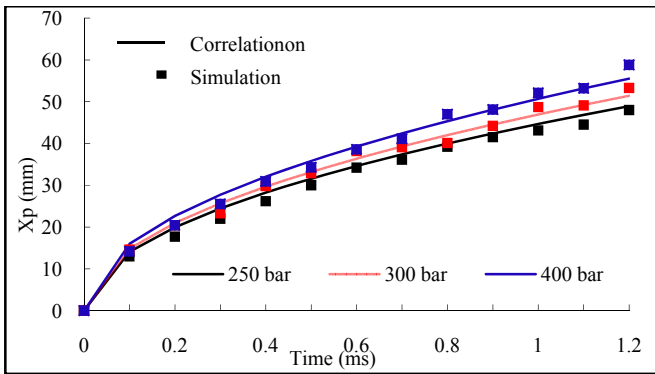


Fig. 9 Spray tip penetration vs time, P_{ch} 2.25MPa, P_{inj} varying

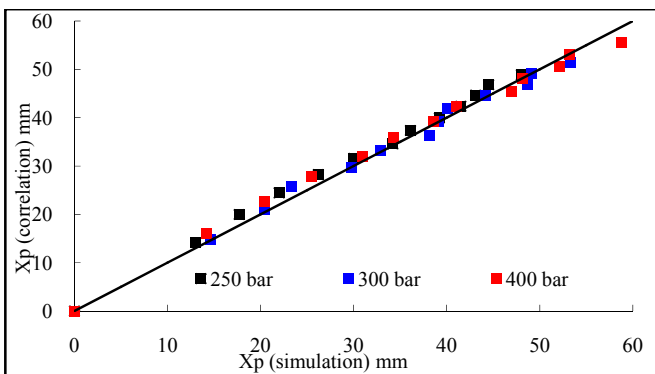


Fig. 10 Penetration (correlation vs simulation)

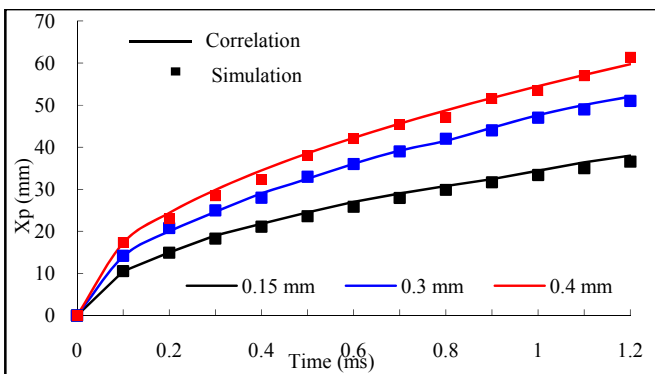


Fig. 11 Spray tip penetration vs time, d varying

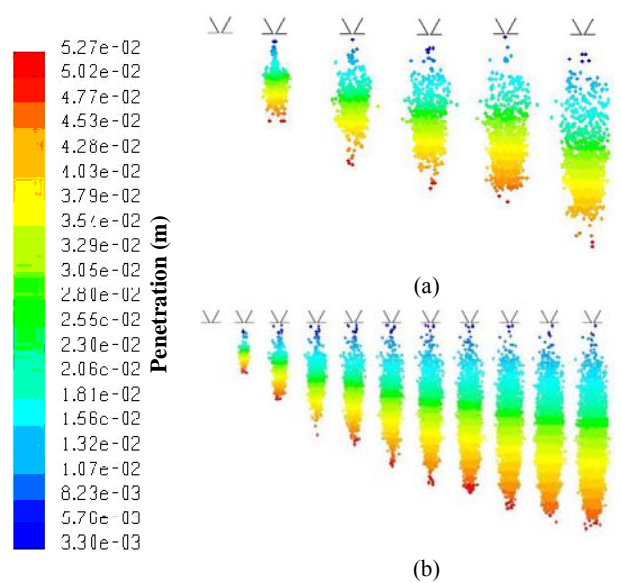


Fig. 13 Spray shape, at (a) 0.3ms interval, (b) 0.15 ms interval

Fig. 13 shows the comparison of spray shapes for evaporating spray simulations of the reference case as described above. For the purpose of comparison the simulated spray images are measured from the commencement of fuel injection and time step is kept 0.3 ms.

When compared to non-evaporating case [11] it appears that at hot bomb conditions the spray shape tends to inflate radially that is the spray shape of larger diameter is achieved. A similar radial inflation is also seen for oblique injection cases at hot bomb conditions [12]. Fig. 13 (b) is the plot for spray simulation shapes using a refined time-step of 0.15 ms between the two consecutive spray images where the customary radial expansion is seen with more refined details.

V. CONCLUSION

The CFD simulation using WAVE model of Reitz [9] has been carried out of diesel fuel sprays using variable injection pressure, injector nozzle diameter and chamber gas pressure. The simulated results are plotted along with the empirical correlation and an excellent match is met. Axial spray tip penetration of diesel sprays decreases with an increase in chamber pressure. On the other hand the penetration rate

increases as the injection pressure is increased for constant chamber pressure. The radial spread of sprays is increased at hot bomb conditions as the higher temperature tends to enlarge the spray radially. With regard to the use of simulation software, number of injected fuel particles are observed to increase as simulation time step is reduced / refined. Increase in number of particles produces well defined shape of the simulated spray.

NOMENCLATURE

P_{inj}	Fuel Injection Pressure
P_{ch}	Chamber Pressure
X_p	Spray Penetration Distance
Δp	Difference between injection pressure and chamber pressure
ρ_a	Chamber air density at given pressure but maintaining the atmospheric temperature
ρ_g	Gas density in the chamber at given pressure and temperature
t	Time span measured from the onset of fuel injection

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