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THE PENETRATION OF OBLIQUE DIESEL SPRAYS UNDER COLD BOMB CHAMBER CONDITIONS

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Abstract

It has been analytically proved that the performance of compression ignition engine is spray characteristics related. The diesel engine performance can be enhanced by achieving improved combustion in the combustion chamber. An efficient combustion depends on the spray formation and ultimate penetration before the onset of combustion. The numerical simulation of fuel spray under varying chamber and injection conditions has been carried out for non-evaporating diesel sprays. The numerical simulations are carried out by using Fluent software. The geometrical modelling is done using gambit and meshing and boundary conditions are also applied in gambit before transferring the model into Fluent. The spray formation and spray penetration rate has been studied under given conditions and the results have been compared with the existing experimental results and theoretical correlations. The simulation is carried out using cold bomb conditions under varying chamber conditions such as gas pressure, nozzle diameter and fuel injection pressure. For comparison purpose the chamber conditions were kept identical as that of the experimental data.

Keywords: Evaporating diesel sprays, penetration rates, hot bomb conditions, numerical simulation, non-evaporating

1. Introduction

The work reported in this paper relates to the computational modelling of fuel sprays under non-evaporating conditions. A spray is defined as non-evaporating if temperature of the chamber air/gas is maintained as atmospheric, magnitude of which is not explicit but varies from 290 to 300 K. In other words, non-evaporating sprays use variable pressure but atmospheric temperature. On the other hand, a spray is defined as evaporating if temperature of the air/gas is also elevated above atmospheric temperature. That is the chamber air/gas pressure and temperature values are variable above atmospheric conditions.

It is important to do the numerical simulations of fuel sprays to understand the formation and penetration of sprays under given conditions. It helps to analyze the combustion process in the engine and hence improve the thermal efficiency. Secondly the emission analysis needs stringent combustion control that depends upon the spray formation in the diesel engine.

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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 (i.e. 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.

A major concern with the diesel engines are the resultant increased level of pollution since they are considered a major source of two key pollutants namely nitrogen oxides (NO_X) and particulate matter [1]. They are also a source of carbon dioxide, one of the most significant greenhouse gases. Nitrogen oxides contribute towards acid rain and ground-level ozone, whilst particulate emissions constitute a major health hazard.

The most favorable 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. This is only possible by carrying out the combustion process at the optimum operating conditions. For this purpose it is highly desirable to develop better understanding of the fuel injection and the subsequent spray formation inside the combustion chamber.

Liquid spray formation concerns a lot of physics, starting from breakup of the liquid core into droplets immediately after the exit from injection nozzle. The secondary breakup happens in a second stage where the formed droplets breakup into smaller droplets. In automotive applications, with high ambient pressures and temperatures, the fuel droplets evaporate along the way 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. Numerous experimental investigations have been directed into understanding the spray formation and penetration in the combustion chamber. Numerical simulations help to understand the dynamics involved in the spray breakup mechanism. In this work the CFD modelling 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. It is not only essential for proficient diesel engine performance but it is also necessary ecologically.

2. Empirical Correlations

2.1. Non-Evaporating Sprays

The mathematical modelling of fuel injection, spray jet formation, its development and dispersion in the combustion chamber is well known to be highly empirical. To develop such models, numerous amount of research work has been carried out on fuel sprays under specific conditions [2, 3, and 4]. For example, in some cases a constant fuel injection pressure was used. Some other experiments involved variable injection pressure where commercially available fuel injection pumps were used to vary the injection pressure. However all such investigations have used the theory of gas jets and applied it to fuel spray jets to derive empirical correlations to describe the spray development. Many empirical and semi empirical correlations for spray penetration history have been published by different authors as discussed later in this work. Apart from some disagreement on the values of constants there exists, however a general agreement among all of them in the set of parameters affecting the spray characteristics.

All such data has been used for comparison with computational fluid dynamics (CFD) simulation results.

For cold bomb conditions the point of intersection for initial near nozzle tip zone and final spray tip zone is called break up point by Mirza [5] and the rate of penetration is given by

$$X_{po} = C_1 \left(\frac{2\Delta \mathrm{p}}{\rho_{\mathrm{f}}} \right)^{0.5} t \qquad \text{------------- (1)}$$

$$X_{po} = C_2 \tanh\left(4.1\times10^3 t\right)^{0.5} \left(\frac{\Delta P}{\rho_a} \right)^{0.25} d^{0.5} t^{0.5} \qquad \text{------------- (2)}$$

Where $C_1 = 0.39$ gives an ideal fit to the experimental spray length and $C_2 = 3.8$ and that maintains the uniformity for conversion of the Eq. 2 into the non-evaporating case

$$t_b = \left(\frac{C_1}{\sqrt{2}C_2} \right)^2 \tanh^2\left(4.1\times10^3 t_b\right)^{0.5} \left(\frac{d\rho_f}{\left(\Delta P \rho_a\right)^{0.5}} \right) \qquad \text{------------- (3)}$$

In order to solve Eq. (3) iteratively, it can be written as:

$$t_b - \left[\left(\frac{C_1}{\sqrt{2}C_2} \right)^2 \tanh^2\left(4.1\times10^3 t_b\right)^{0.5} \left(\frac{d\rho_f}{\left(\Delta P \rho_a\right)^{0.5}} \right) \right] = 0 \qquad \text{------------- (4)}$$

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

$$t_b = 0.6 \left(\frac{C_1}{\sqrt{2}C_2} \right)^2 \left(\frac{d\rho_f}{\left(\Delta P \rho_a\right)^{0.5}} \right) \qquad \text{-------------- (5)}$$

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.

$$t_b = 28.7 \left(\frac{d\rho_f}{\left(\Delta P \rho_a\right)^{0.5}} \right) \qquad \text{------------- (6)}$$

There is some variation in fuel line pressure against time as reported by Mirza et al [6].

3. Results and discussion

3.1. Test conditions

For effectual comparison it is important to choose the correct model to run the simulation for the given conditions. Using experimental data of Mirza [5], Khaleghi [12] did numerical simulation of air-borne non-evaporating diesel sprays. The effectiveness of break up models for both evaporating and non-evaporating sprays was compared by Pizza [13] namely E-TAB and wave breakup models. Using either of these models the results revealed that at suitable mesh resolutions the agreement between experimental and simulated data was excellent.

Wave model for spray break-up is used in the present work, based on the findings of Reitz [7]. For the wave breakup model, the default optimized constants are provided by Fluent software documentation. For non-evaporating sprays, the mean fuel injection pressure is kept at 22 MPa while the chamber pressure is kept at 2.25 MPa.

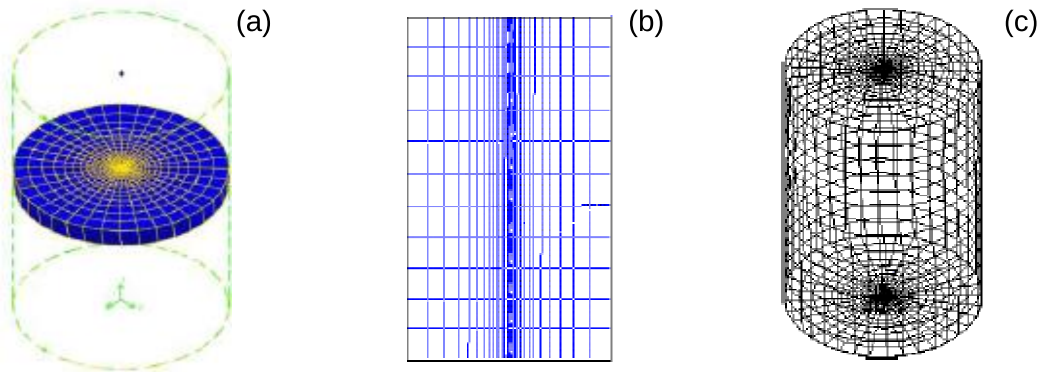

Figure 1: Meshing scheme (a) Cross section of meshed cylinder (b) A slice of meshed cylinder (c) Fully meshed volume

A complex method to control mesh cell size that involve the use of sizing function (S-function) is used for meshing the chamber geometry. A number of independent parameters such as start size, limiting size, growth rate and their limits define the S-function. The typical values of start size, limiting size, and growth rate are taken as 0.0002 m, 0.008 m and 1.05 respectively. The parametric values are applied to the extents of the cylinder. In meshing techniques the sizing function is to be defined at vertices, edges, faces or volumes. The step by step process for the 3D modelling of combustion chamber geometry and its meshing is illustrated in Figure 1.

The simulated results are compared with the correlations of Mirza et al [6, 8]. 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 1.

3.2. Results

In figure 2 the simulated results for spray tip penetration is compared with the correlation and figure 3 shows the spray tip velocity histories for cold bomb conditions. When similar results are plotted for hot bomb conditions under similar geometry limitations, a reduction of approximately 22% in penetration rates was observed.

For varying chamber pressure the simulated results have been shown in figure 4. At elevated chamber pressure the penetration rate remains subdued compared to low pressure values. At elevated pressure, the air is far more compressed and therefore offers a lot more resistance to the incoming spray. In the presence of high air resistance the spray velocity is reduced considerably. The penetration rate is directly proportional to the spray velocity.

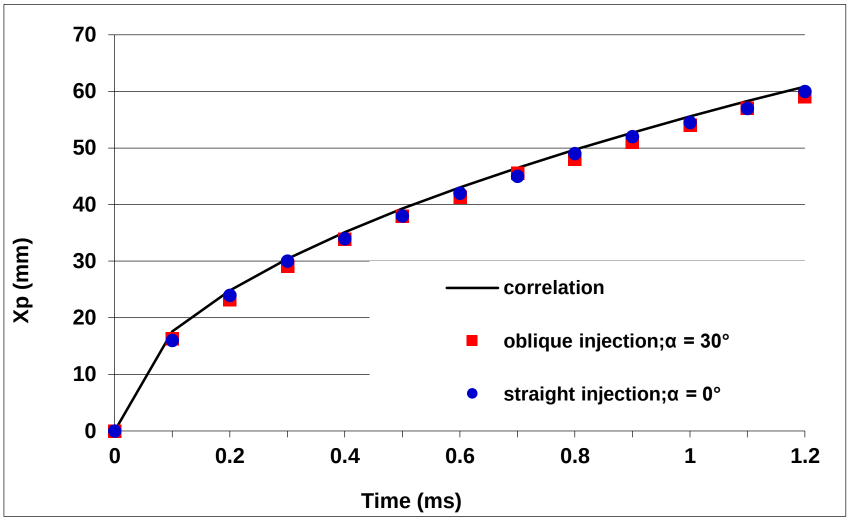

Figure 2: Spray Tip Penetration vs Time

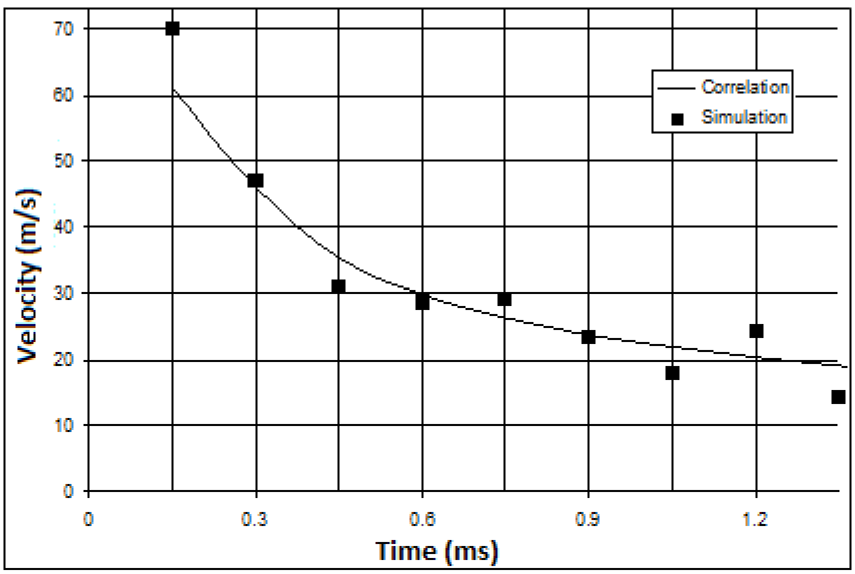

Figure 3: Penetration Velocity (Vp) vs Time

In figure 5 both of the simulated and correlation histories are plotted against each other to reveal the local disparities at each given value. It can be seen that there are negligible local disparities and therefore an excellent match exists between both results.

The injection pressure can also be changed to observe the behavior of spray penetration in the chamber under given conditions. Figure 6 demonstrates the variation of spray penetration rates with the change of injection pressure. At elevated pressure, a far more enhanced penetration rate is achieved. A large injection pressure means an increased spray velocity in the chamber and this eventually increases the penetration rate. Figure 7 is plotted for the spray tip penetration history 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. For a large diameter nozzle the penetration rate is increased considerably because in this case the spray is injected with higher momentum. A higher momentum spray means a large capability to move through high pressure chamber air. This ultimately increases the spray penetration rate in the chamber.

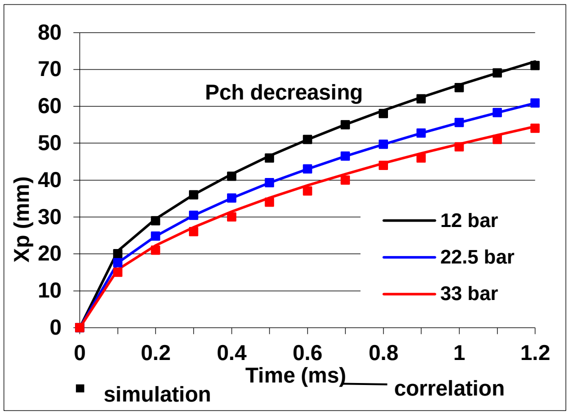

Figure 4: Spray Tip Penetration vs Time, P_{inj} = 22 M Pa, d = 0.25mm, P_{ch} varying

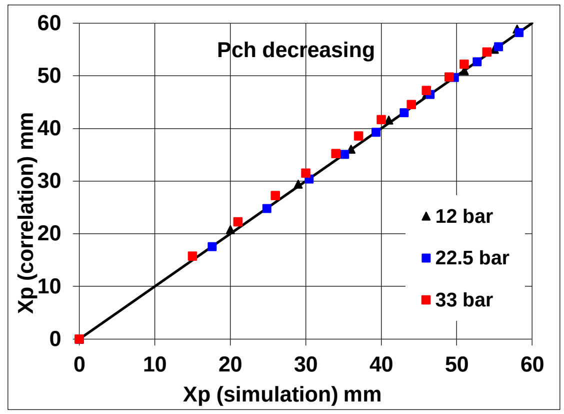

Figure 5: Comparison of the experimental results

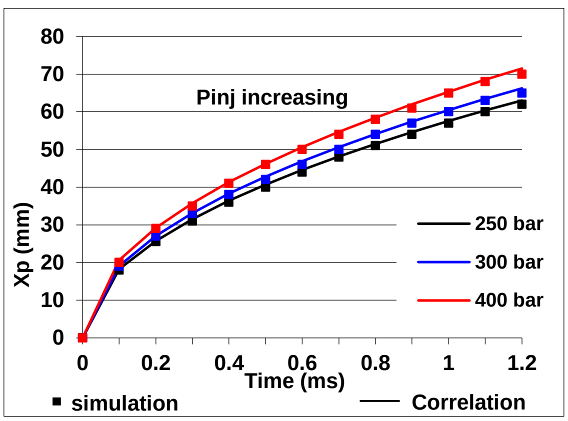

Figure 6: Spray Tip Penetration vs Time, P_{ch} 2.25 M Pa, P_{inj} varying

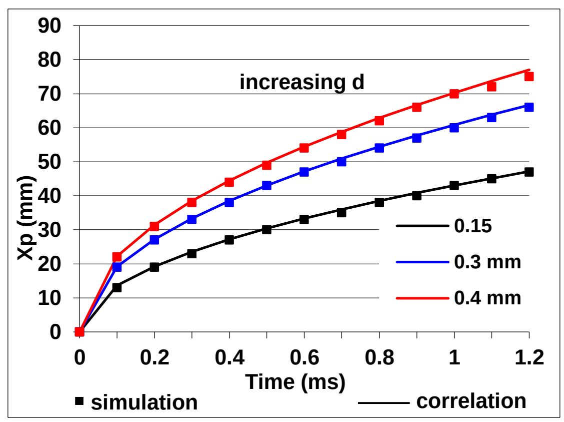

Figure 7: Spray Tip Penetration vs Time, d varying

In a naturally aspirated compression ignition engine, of course depending upon the compression ratio, both temperature and pressure of the air/ gas are high. Chamber pressure may be as high as 50 bar; and temperature may be as high as 850 K. The injection of fuel takes place in the second half of the compression stroke. The injected fuel interacts with the high pressure/temperature chamber air to break into droplets to make a combustible fuel-air mixture. The displaced air, by the injected quantity of fuel, forms convection currents to entrain into the injected fuel. Resultantly, the injected fuel assumes the shape of a conical fuel spray jet. With the lapse of time, usually measured from the onset of fuel injection, the fuel spray jet gets fully developed with a well-defined cone angle as shown in Figure 8 [11]. Length of the spray jet parallel to the injection axis is defined as spray tip penetration distance X_p, as defined in Figure 8.

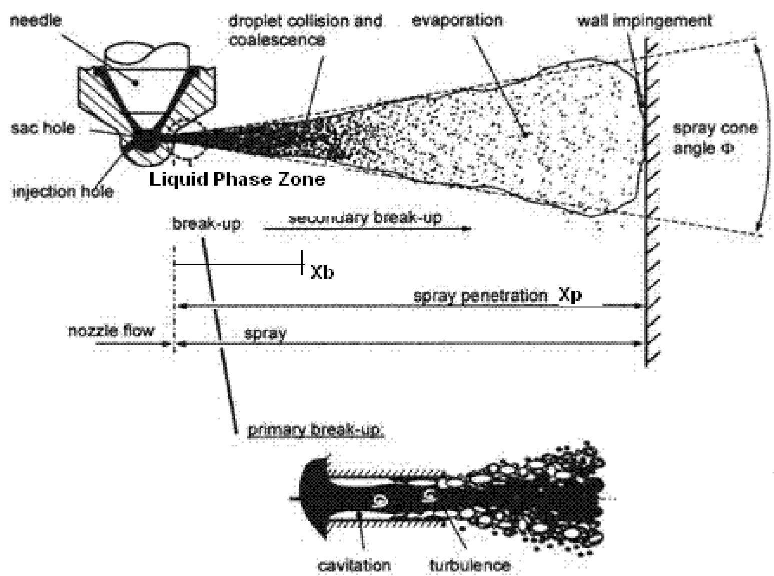

Figure 8: Free hand sketch of a Diesel Spray

In figures 9–11 the spray shapes have been shown as the spray travels in the chamber for non-evaporating spray simulation at given conditions. Figure 9 shows evaporating case while figure 10 shows the non-evaporating case under similar conditions. For non-evaporating case the radial spread for spray shape is a lot more than the evaporating case. At low chamber temperature the spray tends to spread radially a lot more than travelling longitudinally. Similar trends were also in [9, 10] where it appears that at hot bomb conditions the spray shape tends to inflate radially at the tip that is the spray shape of larger diameter at the tip is achieved. Figure 11 illustrates that the spray shape can be refined by using smaller time interval

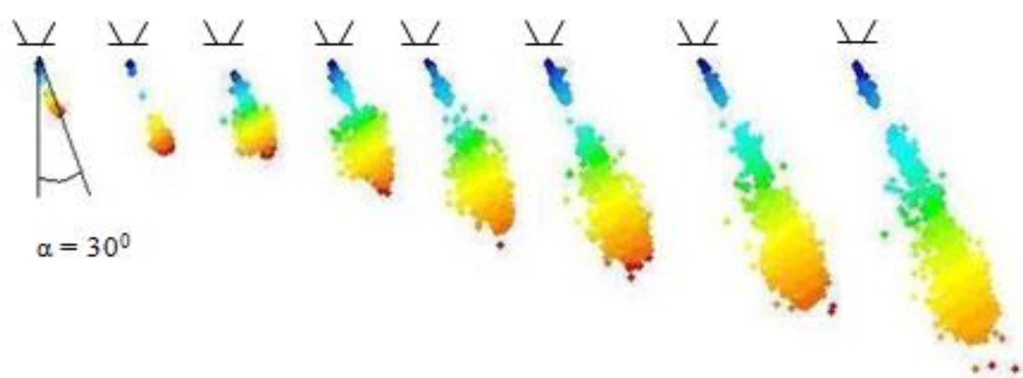

Figure 9: Spray shape, at 0.30 ms interval, starting from fuel injection till end of injection, colored by penetration, Pch 2.25 MPa, Pinj 22MPa, Nozzle diameter 0.25 mm, Temperature 800k

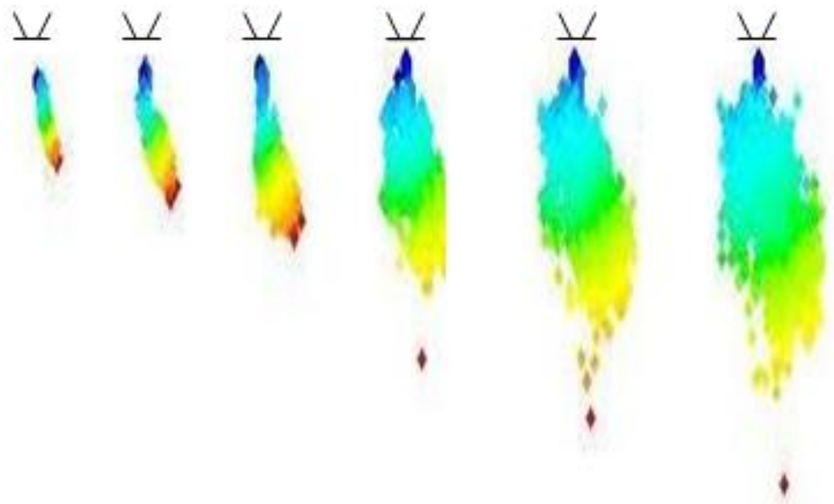

Figure 10: Spray shape, at 0.30 ms interval, starting from fuel injection till end of injection, colored by penetration, Pch 2.25 MPa, Pinj 22MPa, Nozzle diameter 0.25 mm, Temperature 290k

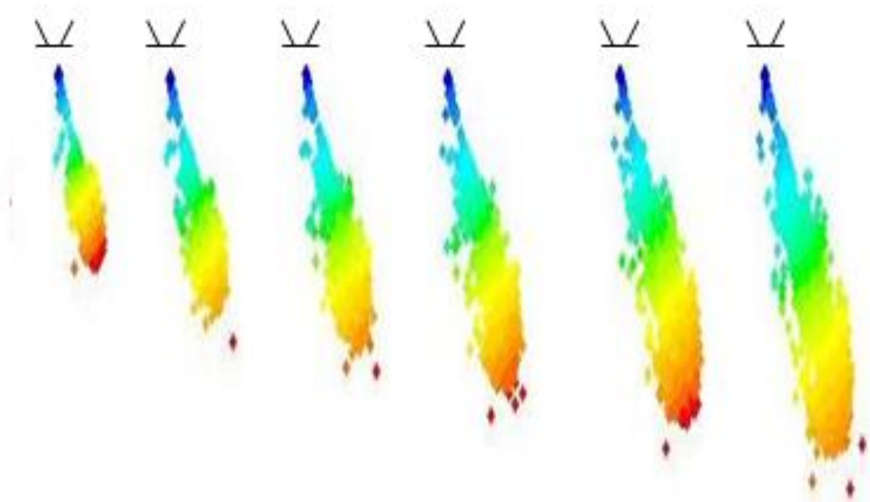

Figure 11: Spray shape, at 0.15 ms interval, Pch 2.25 MPa, Pinj 22 M Pa, Nozzle diameter 0.25 mm, Temperature 290k

Table 1: Fuel and air properties

	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	

4. Conclusion

The CFD simulation has been carried out of diesel fuel sprays by using WAVE model [7] for variable injection pressure, injector nozzle diameter and chamber gas pressure. The simulation reveals that the axial spray tip penetration decreases with an increase in chamber pressure. At increased chamber pressure the air inside the chamber becomes more compressed and offers a lot more resistance to the incoming fuel spray. The increased chamber pressure is beneficial to increase the power output but it certainly decreases the penetration rate. High penetration rate is desirable to achieve more efficient combustion and hence an improved thermal efficiency. On the other hand the penetration rate increases as the injection pressure is increased for constant chamber pressure. This means that the penetration can be increased without having to decrease chamber pressure by other means. The fuel injected at higher momentum will also result in increased penetration and this can be done by increasing the nozzle diameter.

The radial spread of sprays is increased at cold bomb conditions as the smaller temperature tends to enlarge the spray radially. At cold bomb conditions the spray shapes are not as refined as they were for non-evaporating conditions [10]. There is considerable deviation from the cone shape shown in Figure 8. For each case shown in figure 9, the deviation from the cone shape is perceptible. As the time interval is decreased to 0.15 m s, the spray shapes become more conic in shape and conform closely to the predicted shape. Similarly for the spray penetration rate the deviation from experimental results is smaller than the evaporating case. This means that there is need to modify the simulation model for more refined and reliable results. 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

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