ABSTRACT

In direct injection (DI) diesel engines fuel spray injection orientation plays very important role in fuel air mixing and it is highly an influence on engine performance as well as exhaust gas emissions. A single cylinder four stroke DI diesel engine with having multi-hole nozzle injector is considered for the analysis and a computational fluid dynamics (CFD) code, STAR-CD is used for the simulation. In the present study, various injection orientations are considered for the analysis. In-cylinder fuel spray is discussed through the 3D temperature contour plots at orientations of 95°, 100°, 105° and 110° are considered for the analysis. It is concluded that the peak mass average temperatures are higher as well as that attained well in time in case of 100° orientation it indicates effective combustion in this case.

Key words: Diesel engine, spray, injection orientations, temperature, STAR-CD.

1. INTRODUCTION

In DI diesel engines the fuel is injected into the cylinder against the high pressure and temperature compressed air. In this process of combustion is heterogeneous, unsteady, and 3-Dimensional. Combustion is a chemical reaction where certain elements of hydrogen and carbon molecules are combining with oxygen and liberates heat energy. Liquid fuel is injected through the nozzle by the fuel injection system into the cylinder through the end of compression stroke. The liquid jet leaving the nozzle becomes turbulent and spreads out as it entrains and mixes with the in-cylinder air. The outer surface of the fuel jet breaks up into droplets [1]. The initial mass of fuel evaporates first thereby generating a fuel vapour-air mixture sheet around the liquid containing core. Larger droplets provide a higher penetration but smaller droplets are requisite for quicker mixing and evaporation of the fuel. The sprayed fuel stream encounters the resistance from the dense in-cylinder fluids and breaks into a spray. This distance is called the Breakup Distance. Further they vaporize and mix with compressed high temperature and high pressure in-cylinder fluids. At this stage the in-cylinder fluids have above the self-ignition temperature of the fuel. It causes the fuel to ignite spontaneously and initiate the combustion at various locations, where desired condition is prevailed [3].

The spray impinging on the wall of the cylinder becomes an unavoidable. Because of compact high speed DI diesel engines. In fact due to the short distance between injector nozzle and the cylinder walls and also high injector pressures, fuel spray may impinge on the cylinder walls before vaporization takes place. Accordingly spray wall interaction becomes an important phenomenon in high speed DI diesel engines [4]. The spray impingement has a great influence on the distribution of fuel jet, evaporation and subsequent combustion processes. Accordingly injector orientation angle is highly influencing on the spray impingement and subsequent phenomena [5].

2. Combustion

Chen et al. was applied a commercial computational fluid dynamic package STAR-CD to real engine gas flow. The Volkswagen (VW) two-valve transparent 1.91 DI diesel engine was used to simulate flow field in the inlet port and combustion chamber under engine speeds of 1000 rpm and 2000 rpm. The CFD predictions were verified by Laser-Doppler anemometer measurements under motored condition during three periods that are valve opening, valve closure and the maximum valve lift. The transient flow for the induction stroke in the VW engine has been simulated.
successfully using STAR-CD version 2.21. Comparisons between predicted and measured results have shown reasonable agreement for mean flow features. At the valve exit, differences in velocity magnitude at the locations close to the cylinder head may be attributed to drawbacks of the k-ε model and measurement uncertainties. The fluctuating velocities were under predicted, especially at the recirculation regions. This study is the first step of the VW engine simulation. The results show that the large scale flow structure is dominant, and is highly dependent on the port geometry, bowl-in-piston geometry and their locations relative to the cylinder. The turbulent kinetic energy was created principally at the edges of the inlet jet and the cylinder wall jet flow during the early induction stroke. During second phase of the induction stroke, the flow varies linearly with engine speed. This observation justifies the assumption made for validating the CFD results at 2000 rpm for which directly measured experimental data were not available. The successful simulations of the intake flow encourage further numerical analysis of the processes of compression, spray and combustion. It provides considerable potential for detailed information required by designers to evaluate strategies for optimizing engine performance [6]. Trigui et al. had conducted three-dimensional numerical studies during compression process. For this purpose they have used STAR-CD, a commercial CFD code. First, they have calculated both intake and compression process from IVO to TDC. As a second step, they have used three-dimensional Particle Tracking Velocimetry (PTV) data at IVC as initial conditions and performed 3-D computations during compression up to TDC. Then they compared the velocity field computed from both the cases. Their study indicated that this approach is feasible, and that with adequate vector field yield from 3-D PTV, the overall error could be commensurate with any technique (such as LDV or CFD alone) that could be used for the same purpose, but at much lower cost and time [7]. The investigations were performed in a 6-cylinder SI-engine running at wide open throttle by Holger Peters, Ralph Worret and Ulrich Spicher The numerical calculations were performed using the finite volume CFD code STAR-CD. The mesh generation process, including the description of the piston and the valve motion, was automated using Expert System Internal Combustion Engine (ES-ICE). Combustion in the present study was treated with the one-equation Weller flamelet model. This model was implemented in STAR-CD. For the verification of the combustion model, calculated cylinder pressure data and mass fractions burned are compared to experimental results. The results of the combustion process are discussed for different engine speeds and equivalence ratios are shown, encouraging results for the above combustion model. The comparison of the calculated and measured in-cylinder pressure indicates good agreement for equivalence ratios between 0.87 and 1.25 and engine speeds up to 3000 rev/min. The shape of the predicted flame appears to be reasonable [8]. Chen et al. had studied on engine flow using commercial CFD package STAR-CD. The Volkswagen (VW) two-valve transparent DI diesel engine was used to simulate flow field in the inlet port and combustion chamber under engine speeds of 1000 rpm and 2000 rpm. The CFD predictions were verified by Laser-Doppler Anemometer measurements under motored condition during three periods: valve opening, valve closure and the maximum valve lift. In presentation they discussed the predicted flow structure and its main features under different engine speeds, and examined the validity and accuracy of the CFD predictions in comparison with measured data [9].

Bianchi et al. had investigated influence of different initial condition definition procedures on combustion and emissions predictions in small-bore high-speed DI diesel engines. Using the STAR-CD code for the intake stroke and KIVA-2 for the compression stroke, combustion simulation is performed as analysis. The study considered a detailed assignment of the initial conditions at when Inlet valve is closed (IVC). In the second case the results of a time-resolved calculation of the intake stroke performed using the STAR-CD code. An algorithm was developed in order to properly map the data from STAR-CD into KIVA-2[10]. Marc Zellat et al. were described the recent development of the general multi-purpose code STAR-CD in the field of Internal Combustion Modeling with a special emphasis on DI diesel and Gasoline Direct Engine (GDI) engines operating with very high level of Exhaust Gases Recirculation (EGR) (Higher than 30%). A new combustion model is enhanced in STAR-CD. The Extended Coherent Flame Model-3Z (ECFM-3Z) is described first. This model enable to compute combustion for operating conditions with large Exhaust Gases Recirculation (EGR) amount in the GDI and DI diesel engines has been completed. Predictions are compared with extensive data from a DI diesel engine in production over a wide range of operating conditions. The results show that the combustion model used in combination with a two steps auto-ignition model gives realistic heat release history as well as emission prediction [11]. k-ε model is one of the popular in the turbulence models used in IC engines. The k-ε model assumes a Newtonian relationship between the turbulent stress and mean strain rates. Here ‘k’ is the turbulent kinetic energy and ε is the dissipation rate. Morel and Monsour [12] in their model accounted for changes in ε, which is produced by dilation for a flow with compression and expansion. Tahryl computed the turbulent viscosity appearing in the relationship between local turbulent kinetic energy ‘k’ and its dissipation rate ‘ε’ [13]. Bianchi et al. were investigated the influence of different initial conditions, definition procedures on combustion and emissions predictions in small bore high-speed DI diesel engines. The analysis was performed by using the STAR-CD code for the intake stroke calculation and KIVA-2 for the compression stroke and combustion simulation. A full engine mesh and a 60 deg. sector mesh were used in the compression stroke calculations in order to check the accuracy of the axi-symmetry assumption. Analysis of the evolution of fluid dynamic parameters has revealed that a detailed definition of the initial conditions is required to properly predict the mean and turbulent flow field at the time of injection near TDC. The different numerical methodologies yielded great differences in the mean flow field close to the injector location, proving that the assumption of an axis-symmetric initial flow field, which is often used, is far from being valid. Emission results, particularly NOx are found to be strongly
dependent on local in-cylinder conditions and, therefore, on the numerical methodology adopted to determine the initial conditions [14].

3. Results and Discussion
3.1 Engine Geometry and Specifications
In this section the three dimensional plots of spray distribution in a DI diesel engine cylinder for three injector orientation angles at selected Crank angles are presented [15]. The fuel injection parameters are listed in table 2 are considered for the analysis. It is assumed to begin the fuel injection at 10˚ before TDC and end at 3˚ after TDC in the duration of fuel injection is 7˚ CA and the mass flow rate of fuel is considered as 0.0177 kg/sec [16]. A fuel injector with eight nozzles spaced uniformly is considered for the analysis.

In the present work, 45˚sector is taken for the analysis due to the symmetry of eight-hole injector. The computational mesh when the piston is at Top Dead Center (TDC) is shown in fig: 1. the computational domain comprises of the combustion chamber with piston crown. The number of cells in the computational domain at TDC is 10608. Piston bowl dimensions are given in fig: 2. Engine details are given in table 1 and fuel and injection details are given in table 2.

Figure: 1 Computational Sector mesh used in the engine simulation at TDC

Fig: 2 Geometric dimensions of Piston Bowl

<table>
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<tr>
<th>Table 1: Engine Specifications</th>
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<td>Bore</td>
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<th>Table 2: Injection Parameters</th>
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<td>End of injection</td>
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<td>Spray orientation angle</td>
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<td>Total fuel injection per cycle</td>
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<td>Nozzle hole Diameter</td>
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<td>Number of injector holes</td>
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3.2 Initial and Boundary Conditions
It is important to study the in-cylinder fluid dynamics during the later part of combustion and initial part of expansion strokes in DI diesel engines. Analysis is carried out from 40° before TDC (bTDC) to 80° after TDC (aTDC), as fuel injection combustion and pollutant formations are taken place during this period.
The initial swirl is taken as 2 m/s and the constant absolute pressure and temperatures as 9.87 bar, 583 K respectively. The turbulent model has the Intensity length scale as 0.1 and 0.001 respectively and it is assumed that there is no traces of fuel and exhaust gases. The initial surface temperatures of combustion dome region and piston crown regions are taken as 450 K and the cylinder wall region has temperature of 400 K.

3.3 Effect of Injection Orientation on Mass-Averaged Temperatures Variation with Crank Angle
The effect of injection orientation angle on mass average temperature prevailing in the combustion chamber with respect to crank angle is shown in figure 4. The figures are drawn with the data generated from the computational fluid dynamics (CFD) simulation of combustion process. Figure 4, shows the variation of in-cylinder temperature at different crank angles for four injection orientation angles varies from 95°-110°. It is observed that the mass-averaged temperatures are gradually rise from 580 K to 846 K till start of injection and follows the same path in all the three cases from 40° before TDC to 10° before TDC. This is attributed to in-cylinder fluid temperature rise due to compression of in-cylinder fluids during compression stroke. There is a slight drop in mass-average temperatures, noticed from 10° before TDC, during a short span, this is due to endothermic reaction taking place during evaporation of fuel, as the fuel injection starts from 10° before TDC to 3° before TDC. A sharp rise in mass-average temperature profiles was noticed, this indicates combustion has taken place.

In case of 100° injection orientation angle the peak mass average temperatures are attained little earlier than the other two cases. The average mass temperature of 1890 K was noticed at 9° after TDC. In case of 95° injection orientation angle it is observed that a little delay to attain the peak mass average temperature of 1928 K at 14° after TDC. In case of 105° and 110° injection orientation angle mass average temperature of 1759/1758 K was observed at 8° after TDC. It is very clear from the figure 4, that the mass average peak temperatures are attained well in time for the case of 100° orientation i.e., 3° crank angle earlier than that of other two cases and its peak mass average temperatures are also higher than that of other two. From the figure it was inferred that at 100° injection orientation angle good combustion taken place at right time.

![Pressure Vs. Crank Angle](image)

**Figure: 3** Comparison of injection orientation of Mass-Averaged Temperature vs. Crank Angle Profiles for the four spray orientation

From the above analysis it can be concluded that in case of 100° injection orientation angle there is an early start of combustion and early reach of peak mass average temperatures and sustain a good mass average temperatures throughout the combustion process.

From the graph it is noticed that in case of 95° injection orientation angle there is a higher temperature in the exhaust stages. It indicates that there is delay in combustion phase. Due to more wetting area, the fuel that is impinged on the walls will be adhere on it during initial part of combustion, the same is participated in the late combustion stage. This contributes to higher temperatures in the late combustion phase. In case of 100° injection orientation angle the late combustion is observed to be optimum. At 80° after TDC higher mass-average temperatures of 1900 K, 1905 K 1750 and 1750 K are noticed for the cases of 95°, 100°, 105° and 110° injection orientations respectively. From the graph it was observed that 80° after TDC there is no drastic change in mass average temperatures of exhaust gases in all the three cases.

3.4 Effect of Injection Orientation on In-Cylinder Temperature distributions
The effect of injection orientation angle on in-cylinder temperature distribution for the four different fuel injection orientation angles viz 95°, 100° 105° and 110° are shown in figure 4 (a-d).
From the figure 4, it is observed that at 2\(^\circ\) after TDC initiation of combustion taking place at selected pockets, where the fuel air mixture is near to stoichiometric or slightly higher. The first ignition is due to the temperature rise associated with the strong pressure wave which emanates from each ignition site due to local rapid chemical energy release. At this stage near lip of the bowl as well as at the bottom sides of the bowl regions initiation of combustion starts in all the three cases; however the high temperature zones are small in the combustion chamber. The minimum temperatures are noticed in the squish region in all the three cases. This is due to higher surface heat transfer in this region.

At 4\(^\circ\) after TDC from the figure 5(a-d), it is observed that the spreading of high temperature flame takes place in all the three cases, this indicates, that the combustion of fuel has been taken place at multiple locations. This is due to spontaneous ignition at regions, which are very close to the region where the first ignition occurs. In case of 95\(^\circ\) injection orientation figure 5(a) along the side walls of the bowl as well as squish region there is a strip of narrow band of high temperature flame propagation. In case of 100\(^\circ\) injection orientation as shown figure 5(b), it is observed that the flame propagates vertically along with side walls of the bowl and horizontally along the piston head toward the axis of the bowl. In case of 105\(^\circ\) and 110\(^\circ\) injection orientation as shown in figure 5(c-d), it is observed that there is a delay in flame spreading.

This is attributed to that the spray orientation. As the spray orientation angle is more the fuel spray travelled distance is more. In case of 95\(^\circ\) and 100\(^\circ\) injection orientation the spray direction towards the strong vortex which is situated at the lip of the bowl, whereas in the case of 105\(^\circ\) and 110\(^\circ\) injection orientation the spray is directed away from the strong turbulent zone, hence the delay in combustion occurs. At 6\(^\circ\) after TDC shown in figure 6(a-d), combustion related high temperature zones of temperatures 2757 K, 2798 K, 2711 K and 2711 K, are noticed for the four fuel injection orientation angles viz 95\(^\circ\), 100\(^\circ\), 105\(^\circ\) and 110\(^\circ\) cases. This is attributed that the high heat release rates due to rapid combustion during this period. At this stage peak mass average temperature are noticed and are represented in figure 6. At this stage flame front which is at the cylinder head appears to propagating towards the squish region i.e. positive x-direction, another wave flame front moving negative x-direction at the bottom of the bowl is noticed.
At 20º after TDC shown in figure 7(a-d), in case of 95º injection orientation shown in figure 7(a), the flame front fully occupies the squish region, in case of 100º and 110º injection orientation it is noticed that the flame propagation progresses in squish region. Another flame front which is situated at the bottom side of the bowl is noticed which is rolling upwards and moving towards the squish region. This is attributed to the expansion stroke, a space is created due to the piston moving away from the cylinder head. This resemble to its corresponding velocity vector plots which contain higher velocities. In the velocity vector plots and its corresponding temperature contour plots contains higher temperatures in the same region and propagation. At this stage cylinder occupying the higher temperatures of the order of 2757 K, 2798 K, 2711 and 2711 K respectively.

**Fig. 7 Temperature Contour Plots at 20º after TDC at Various Injection Orientation**

### 4. Conclusion

It is observed that the mass-averaged temperatures are gradually rise from 580 K to 846 K till start of injection and follows the same path in all the three cases from 40º before TDC to 10º before TDC. In case of 95º injection orientation angle there is a delay to attain the peak mass average temperature, the attained temperature at 14º after TDC is 1928 K. It is observed that the mass average peak temperatures are attained well in time in case of 100º orientation i.e., 3º crank angles earlier than that of other two cases and its peak mass average temperatures are also higher than that of other two cases. It indicates that in case of 100º injection orientation there is good combustion taken place at right time. In case of 105º and 110º orientation, it attained maximum temperature of 1758 K at 8º after TDC. It is noticed that in case of 95º orientation there is a higher temperature in the exhaust stages. It indicates late combustion.

### REFERENCES


AUTHOR

Dr. P. Manoj Kumar has 13 years of teaching experience. Previously worked in St John’s College of engineering college for 7 years as Associate Professor and HOD, Stananley Stephen college engineering & Technology as a Professor and HOD of mechanical engineering department for 5 years. Presently working in Guru Nanak Institutions Technical campus, Ibrahimpatnam (R.R.Dist) as a Professor and HOD in the department of Mechanical Engineering. He has completed research in the area of IC engines under JNT University, Anapatur, Andhra Pradesh, India. He has published 1 national, 7 international papers.