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Multidimensional Modeling of Direct Injection Diesel Engine with Split Multiple Stage Fuel Injections

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Abstract

In the present study, an attempt has been made to predict the influence of split multiple stage fuel injection on a DI diesel engine performance and emission characteristics. The predictions have been made for both conventional continuous fuel injection as well as split multiple stage fuel injection. The analysis mainly emphasize on the prediction of SFC and NO_x levels in a diesel engine for the above mentioned two fuel injection modes. These computational predictions can enhance the knowledge of the flow and combustion characteristics, which is of vital importance for the design and development of a high performance DI diesel engine. The aforementioned simulation work has been performed using a CFD code. It has been found that the split multiple stage fuel injection exhibits strong effects on combustion characteristics and provides controlled pressure and temperature inside the combustion chamber. It has also been seen that split multiple stage fuel injection significantly reduces the formation of NO_x compared to that of the continuous fuel injection.

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Keywords: DI Diesel Engine, Split MultipleStage Fuel Injection, Computational Fluid Dynamics, Mesh Motion, Two phase, Combustion

Nomenclature

CFD	computational fluid dynamics
SFC	specific fuel consumption
ATDC	after top dead centre
Φ	conservative and/or transport variable
κ	turbulent kinetic energy
ε	dissipation of turbulent kinetic energy
$S_{i,j}$	source term of conservative and/or transport variable
$\sqrt{g}$	determinant of metric tensor

1. Introduction

The diesel engine pollutants such as soot and NO_x are very sensitive to the gas temperature, fuel injection pressure, injection timing, etc. The conventional continuous injection system does not offer adequate freedom to control these parameters. The high pressure, rate-shaped multiple injection schemes enhances the mixing inside the chamber with controlled pressure and temperature rise thus allowing combustion to continue even at late expansion. The NO_x emissions can be reduced with optimized

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injection characteristics, which enhance mixing and differ from the traditional single injection strategy. In order to analyse the effect of engine performance with continuous and split multiple fuel injection schemes, a detailed three-dimensional CFD analyses have been carried out. The work includes creation of the CAD model of the DI diesel engine, mesh generation with events (for the mesh motion) and combustion analysis.

2. Physical Model

A 4-cylinder DI diesel engine of displacement 2148 cc has been used in the present numerical combustion study. The details of the specifications are presented in the following Table no. 1.

Table 1. Engine Specifications.

No. of Cylinders / arrangement	4, in-line
Engine displacement	2148 cc
Bore and Stroke	88 x 88 mm
Connecting rod length	220.0 mm
Maximum power	105.7 KW @ 4000 rpm
Maximum torque	340 Nm @ 2005 rpm
Mixture formation	Common rail, direct injection
Compression ratio	18:1
Engine CFD analysis speed	2005 rpm
Swirl ratio	1.0
Intake valve diameter	35.5 mm
Exhaust valve diameter	30.0 mm
Fuel	Dodecane ($C_{12}H_{26}$)

3. Mathematical model

The mathematical model has been based on the numerical solution of Favre-averaged transformed conservation equations of mass, momentum, energy, turbulent kinetic energy and its dissipation in the gas phase within the combustion chamber and can be represented by the generalized Equation 1. The $\kappa - \varepsilon$ version of RNG [1,6,7] model has been used to model the turbulence effects. The injection velocity of the liquid fuel strongly influences the atomization, the spray penetration, the inter-phase transfer processes and the droplet-droplet interaction. The process of particle dispersion in the cylinder due to fuel injection has been modelled by the dispersed Lagrangian multiphase model [1,7,8]. The Huh's atomization model [1,7,9] in which a probability is generated based on the minimum distance that occurs between the two particles as they pass each other is chosen for the atomisation and the Reitz and Diwakar [1,7,10] break-up model has been used for the droplet break up process. The O'Rourke collision model [1,7,11] has been used to simulate the inter droplet collisions. A laminar-and-turbulent characteristic-time scale model of Magnussen [1,7] based on the eddy break-up (EDBR) concept has been adopted in the present combustion simulation. This combustion model relates the rate of combustion to the rate of dissipation of eddies and expresses the rate of reaction by the mean concentration of a reacting species, the turbulent kinetic energy and the rate of dissipation of kinetic energy. Auto ignition process has been modelled using shell auto-ignition model [1,7,13]. The rate of formation of NO_x is significant only at high temperatures since the thermal fixation of nitrogen requires the breaking of a strong N_2 bond. This effect is represented by the high activation energy of reaction, which makes this reaction. This effect has been modelled by using the rate-limiting step of the Zeldovich mechanism [1,7,13,14].

The generalized transformed conservation and transport equation can be presented as below.

$$\frac{1}{\sqrt{g}}\frac{\partial(\sqrt{g}\rho\varphi)}{\partial t}+\frac{\partial}{\partial xj}\quad \rho u\varphi - \Gamma j \quad = S i,j \text{................................(1)}$$

Where, ϕ represent any conservative and/or transport variable such as velocity, enthalpy, κ, ε, species mass fraction etc. and Γj is the diffusion flux in j direction, $\sqrt{g}$ is the determinant of metric tensor and Si,j is the source term of the subsequent variable.

To get the solution of above generalized equation for any variable in the calculation domain, the equation is to be discretized using several standard techniques to linearize the second order terms in mass, moment, energy, κ, ε and species conservation equations. And subsequently the linear equation is been solved using the iterative solver.

4. Computational domain

The physical domain has been discretised accordingly for the mesh motion and 180 degree sector has been considered for simplicity and modelled using software for the simulation of engine cycle. The model consists of 1, 20,000 cells and are shown in Figure 1.

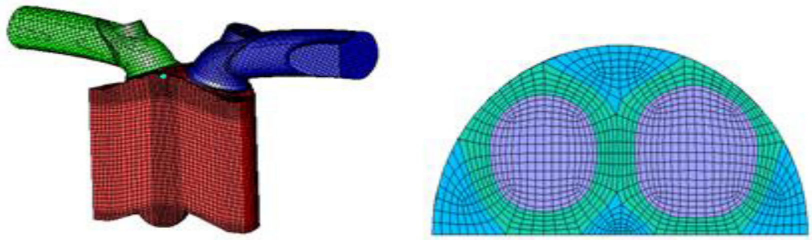

Fig.1. Engine model with computational mesh of complete domain (left) and at valve cross-section (right).

The grid spacing in axial and radial directions has been smoothened in order to minimize the deterioration of the formal accuracy of finite volume technique due to variable grid spacing in a way that a higher concentration of nodes occurred near the cylinder head, cylinder liner, piston surface and the valve faces.

5. Boundary conditions

No slip boundary condition has been provided at all the wall boundaries (solid surfaces like cylinder head, cylinder liner, piston face, valve stem and the valve faces). Pressure boundaries with temperature and turbulence quantities are specified at the intake and exhaust ports. Respective temperature values are specified on all the wall boundaries. The typical boundary values assigned for the present work are shown in Table 2. The injection timings in the continuous and split cases are varied as shown in Figure 2.

Table 2. Boundary Conditions.

Boundary Region	Pressure (Pa)	Temperature (K)	Turbulence	
			Intensity	Length scale
Intake Port	1.0e+5	300	0.1%	0.001
Exhaust Port	1.0e+5	300	0.04%	0.001
All other faces	No-slip wall boundary condition with respective temperature			

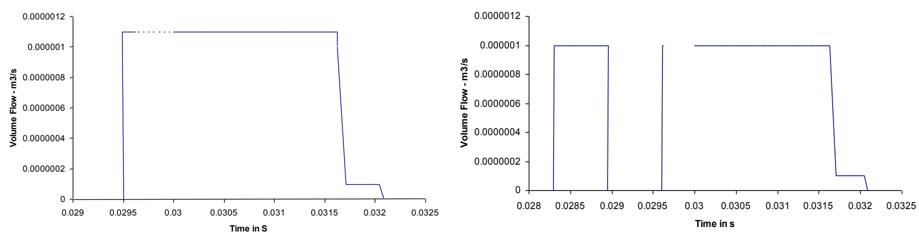

Fig 2. Injection profile for the cases of continuous (left) and split (right) injections.

6. Combustion analysis

The commercial CFD code is used for the present simulation has been validated for modelling diesel engine combustion by Gosman. A time step (Δt) of 8.31331×10^{-6} s (corresponding to 0.1^{o} of crank rotation at an engine speed of 2005 rpm) is used for the entire simulation. Fuel is assumed to react irreversibly in the vapour phase with the available oxygen to form products. Also the diffusivities of mass, momentum and energy are taken to be equal, which is a reasonable assumption in a turbulent flow. The combusting mixture therefore consists, in general, of fuel, oxygen and products whose mass fractions are to be calculated. This is supplemented by solving two conservation equations for the fuel concentration and mixture fraction, resulting from the fact that both quantities have similar initial and boundary conditions.

7. Results and discussion

7.1 Gas Phase Velocity

The gas phase velocities at 90^{o} and 345^{o} crank angles are shown in Figure 3 and 4 respectively. It is observed that the mean gas phase axial velocity hits the cylinder bottom wall and takes upward motion, known as tumble gas motion and the secondary velocity becomes negative in the central region of the cylinder due to a typical axial gas re-circulation, which forms the swirling flow inside the cylinder. The axial velocity then increases and reaches a high positive value away from the cylinder axis and then again decreases to a negative value close to the wall because of a near wall circulatory flow.

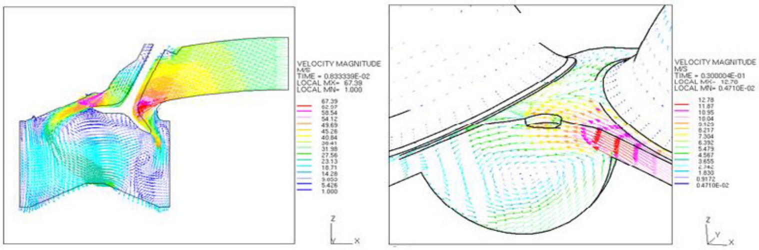

Fig 3. Velocity vector at 90^0(left) and 345^0(right) crank angle.

The gas phase velocity at 345^o of crank angle shows the amplified swirling and strong re-circulatory flow inside the piston bowl. This gives better fuel air mixing and fuel evaporation. The predicted velocity pattern assures that the whole charge will be inside the bowl. The burning of smaller size droplets leads to a uniform temperature field inside the entire combustion volume.

7.2 Fuel Concentration and Spray

The fuel concentration for continuous injection, at 10^o of crank angle shows that the predicted trajectory is straight path during early part of injection, followed by the clear evidence of deflection of the tip by the swirling motion in the later stages, and the impingement on the bowl wall, leading to droplet breakup due to splashing. Also the effects of the spray on the gas flow mainly manifested as a result of strong induced velocities in the direction of the spray motion. In the early stage of injection, the droplet penetration is higher than that of vapour, because the bulk of the droplets are cold. After some time, the droplets are decelerated due to the drag force and evaporation causes the vapour penetration to exceed that of the liquid.

7.3 Cylinder Pressure and Temperature

During motored condition, the maximum average cylinder pressure and temperature occur at 360^o of crank angle and the corresponding values are 57 bar and 1050 K respectively. These values are in-line with the ideal gas equation of state for the isentropic process. In case of continuous injection, at the end of fuel injection, the average cylinder gas pressure and temperature are 51 bar and 940 K respectively. The combustion is initiated after 3^o of injection crank angle (ignition delay) due to auto ignition process. This increases the cylinder pressure and temperature to their peak values of 135 bar and 2422 K respectively at 5^o ATDC. The pressure and temperature at 700^o crank angle are 3.5 bar and 750 K respectively.

In multiple fuel injection 25% percentage of the fuel is injected in the first stage of fuel injection. The predicted cylinder pressure and temperatures at this crank angle are 29 bar and 800 K respectively. In the second stage of injection, the remaining 75% of the fuel is injected. The obtained cylinder pressure and temperature at this crank angle are 51 bar and 940 K. It can be seen that the combustion is started after 3^o of injection crank angle due to auto ignition process. This increases the cylinder pressure and temperature to its peak value of 120 bar and 2276 K respectively at 8^o ATDC. This is slightly lower than that predicted for the continuous injection case.

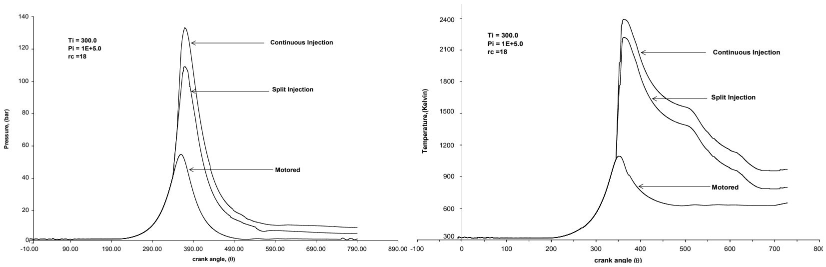

Fig 4. Variation of mean cylinder pressure and temperature.

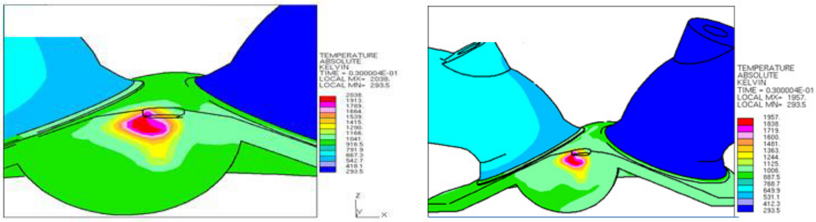

Fig 5. Temperature contour at 350^0 of crank angle during continuous (left) and split (right) injections.

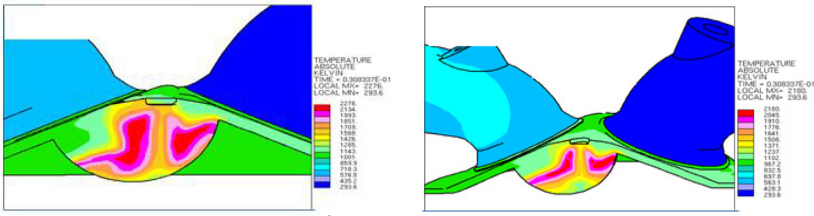

Fig 6. Temperature contour at 360^0 of crank angle during continuous (left) and split (right) injections.

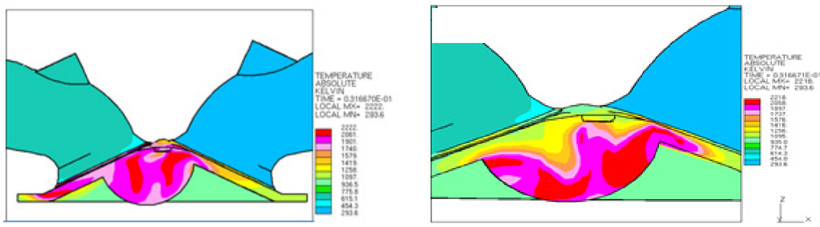

Fig 7. Temperature contour at 370^0 of crank angle during continuous (left) and split (right) Injections.

7.4 Pollutant (NO_x) Formation

The NO_x contours obtained from the analysis shown in Figures 7 and 8 depicts that the concentration level is higher in case of combustion with continuous injection. This is because of the fact that the charge is highly inhomogeneous due to insufficient time for thorough mixing that resulted in higher combustion temperature. While in case of multiple fuel injections, NO_x concentration is lower due to the formation of a homogeneous charge because of early injection in split form which enhances the mixing intensity till second stage fuel injection and controls the temperature in the combustion volume which reduces NO_x level.

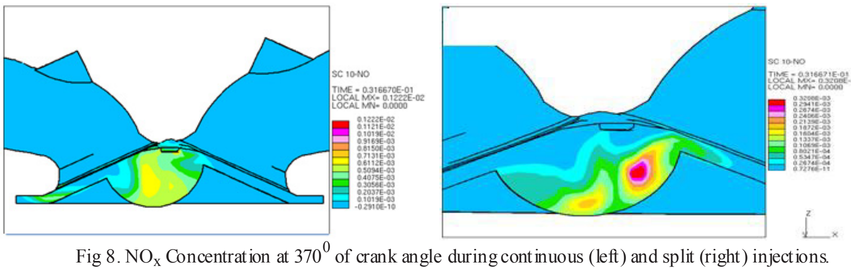

Fig 8. NO_x Concentration at 370^0 of crank angle during continuous (left) and split (right) injections.

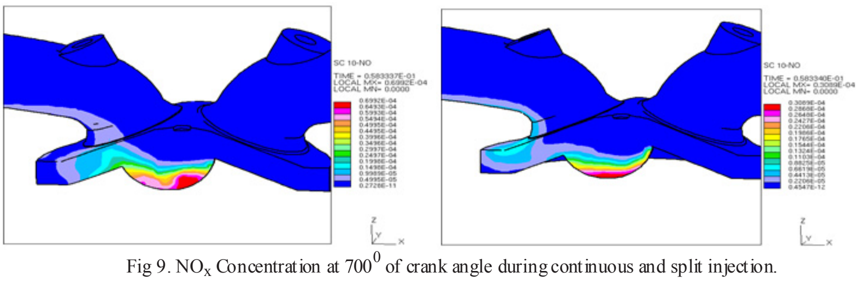

Fig 9. NO_x Concentration at 700^0 of crank angle during continuous and split injection.

7.5 Engine Performance Characteristics

The predicted engine performance is given in Table 3. The combustion with multiple fuel injections shows 45% reduction in NO_x value compared to that of the continuous injection, which is because of formation of homogeneous charge due to early and split injection that enhances the mixing intensity with controlled combustion, reduced temperature and subsequently the NO_x.

Table 3. Engine performance characteristics (Ti= 300 K, P_i= 101325 Pa, r_c=18)

Fuel Injection	Volumetric Efficiency, $\eta_{v\%}$	Specific Fuel Consumption, SFC, gm/ kW-hr	NO_x Pollutant gm/ kW-hr
Continuous	84.54	295.1	7.5
Split (25-75)	84.54	295.1	4.1

Conclusion

The cases of continuous injection and split multiple stage fuel injections have been numerically investigated using a set of injection model schemes. The study finally depicts the following pertinent conclusions.

1. At motored condition, the maximum average cylinder pressure and temperature occur at 360^o of crank angle and that corresponds to 57 bar and 1050 K respectively, while in case of continuous injection it corresponds to 51 bar and 940 K respectively at the end of fuel injection and the combustion is initiated just 3^o ignition delay after fuel injection due to auto ignition process delay leads to peak values of 135 bar and 2422 K respectively at 5^o ATDC. The pressure and temperature at 700^o crank angle are 3.5 bar and 750 K respectively.
2. In multiple fuel injections, just after first stage of 25% fuel injection the pressure and temperatures are found out to be 29 bar and 800 K respectively, while after second stage of 75% fuel injection it corresponds to 51 bar and 940 K respectively and the combustion is initiated just 3^o ignition delay after fuel injection due to auto ignition process delay leads to peak value of 120 bar and 2276 K respectively at 8^o ATDC, which is slightly lower than that predicted for the continuous injection case. It can be concluded from study that the combustion with split multiple stage fuel injection exhibits strong effects on combustion and provides controlled pressure and temperature inside the combustion chamber.
3. It is also observed that a multiple injections with 25% fuel in the first pulse and 75% in second pulse can significantly reduce the NO_x formation by 45% compared to the continuous fuel injection combustion. This is because of the fact that in case of continuous fuel injection the charge is highly inhomogeneous due to insufficient time for thorough mixing that resulted in higher combustion temperature, while in case of multiple injection, the charge gets more homogeneous because of early injection in split form which enhances the mixing intensity till second stage fuel injection and the combustion temperature reduces drastically, which is the most responsible factor for NO_x formation.

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