Investigation of the Effect of Injection Timings on the Performance of an Internal Combustion Engine using Computational Fluid Dynamics

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Abstract:

Almost all the vehicles on road today are powered by Internal Combustion (IC) engines. Internal combustion engine is a heat engine in which the chemical energy of the fuel released in the combustion chamber is directly used to produce mechanical work. The performance of an IC Engine depends on many parameters like the time scales of the Intake Airflow, Fuel Injection Timings, Dynamics of turbulent reacting flows, Air Fuel ratios and so on. It is well known that fuel injection strategies including fuel injection timings play a vital role in the performance of an IC engine.

Computational Fluid dynamics (CFD) has emerged as an inevitable tool in the design of IC engines. Unlike the conventional experimental techniques, CFD predicts the detail insight into the spatial temporal variations of all the variables, without modifying or installing the components. Advent of powerful hardware, parallel processing techniques, cloud computing further enhanced CFD to significantly reduce the cost and turnaround time in the design process.

In this study Computational Fluid Dynamics is used to investigate the effect of split injection system in comparison with uniform injection on the performance and emission components of a compression ignition engine.

Keywords:

Internal Combustion Engine, Computational Fluid Dynamics, injection strategies, Performance of a compression ignition engine.

Acronyms:

IVC = Intake Valve Close; IVO = Intake Valve Opening; ATDC = After Top Dead Centre; ABDC = After Bottom Dead Center; CO = Carbon Monoxide; CO_2 = Carbon Dioxide; NO_2 = Nitrogen Dioxide EVC = exhaust valve close; EVO = exhaust valve opening;

Introduction:

The objective of the present study is to analyse the effect of two injection strategies on the performance of Compression Ignition engine. In first simulation uniform injection is given and in the second simulation same amount of fuel is injected to the multi injection system.

Many studies concluded that the performance of a multi injection depends on the timing and dwell between injections. Shundoh et al. (1992) concluded that nearly 35% reduction in NOx can be achieved using the multi injection system without any compromise on fuel economy. Tow et al. (1994) concluded from an experimental study a double injection with a significantly long delay between injections reduced particulate by as much as a factor of three over a single injection at 75% load with no increase in NOx. Zhang (1999) showed that by increasing the dwell between pilot and main injection and reducing the fuel in pilot injection resulted lower soot. Raouf Mobasheri et al. (2012) discussed about using pilot injection accompanied with an optimized main injection has a significant beneficial effect on combustion process so that it could form a separate Second stage of heat release which could reduce the maximum combustion temperature, which leads to the reduction of the NOx formation.

In the current study, along with pilot and main injections advanced post injection is also introduced. It was of interest to check the emission reduction capability of a multi injection (Three Pulses) system in comparison to uniform injection system using CFD. In multi injection system three pulses are given i.e Pilot Injection, Main Injection, Advanced Post Injection. Comparisons of maximum static temperature, Maximum static pressure, Penetration length, Mass averaged turbulent kinetic energy, heat release rate, Mass fraction of Carbon monoxide (CO), Carbon dioxide (CO_2), and Nitrogen dioxide (NO_2) were presented.

Geometry & Meshing:

Connecting rod length of 165mm and crank radius of 55mm with 0.0 piston offset is used to create the geometry. Compression ratio was set to 15.75:1. Engine speed is set to 1500 RPM. Deep Re-Entrant bowl is considered for piston bowl as shown in figure 1.

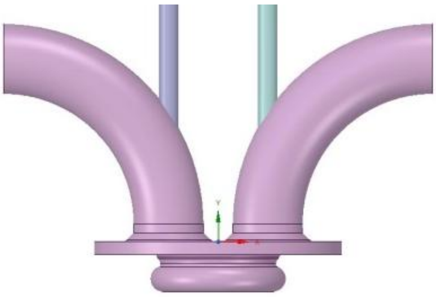


Figure 1: Deep re-entrant bowl.

In both the simulations approximately 0.5 Million elements were used to discretize the geometry. The present mesh density was found to give sufficiently grid independent results. Injector is located at the centre of the combustion chamber, and because of this symmetrical location, 60^0 Sector was used for the simulation. The valves are not considered because this simulation is carried out between closures of inlet valve (IVC) and opening of outlet valve (EVO). For the purpose of simulation beginning of crank angle is considered as 360^0. In this case, the piston reaches the bottom dead centre at a crank angle 540°. The Inlet valve closes at 570° i.e. 30° ABDC (after BDC). Piston reaches the TDC at 720° and BDC at 900°. The Outlet valve opens at 60° (approx) before BDC i.e 833^0 of crank angle. Spray starting crank angle is 712 ° (8^0 before TDC). Spray angle is 70^0 .

Injection profiles:

The main aim of this work is to understand the effect of split injection on emission reduction capability without compromising fuel economy. In both the situations injection is given from the crank angle 712^0 to 738^0. In split injection same amount of fuel is given in three pulses.

In split-Injection simulation, approximately 22% of the total fuel is injected from crank angle 712^0 to 716^0 as pilot injection. Main injection consisting of 67% of the total fuel is injected from crank angle 725^0 to 738^0. Remaining 11% fuel is injected between the crank angles 736^0 to 738^0. Figure 2 shows the profile chart for uniform injection field and split injection field.

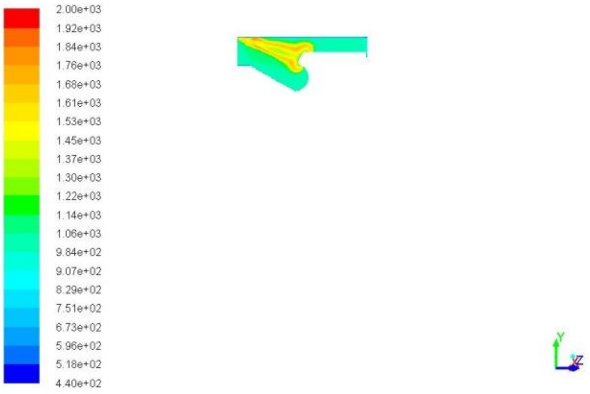
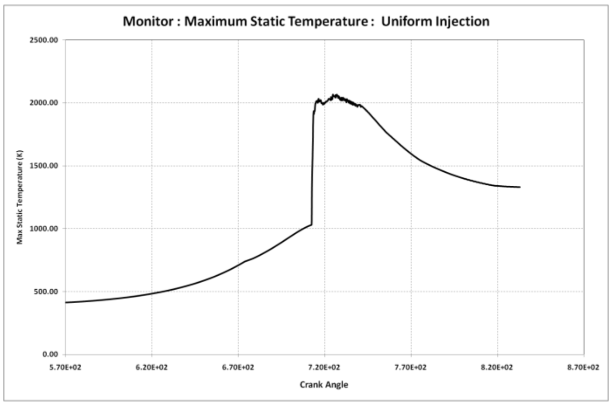


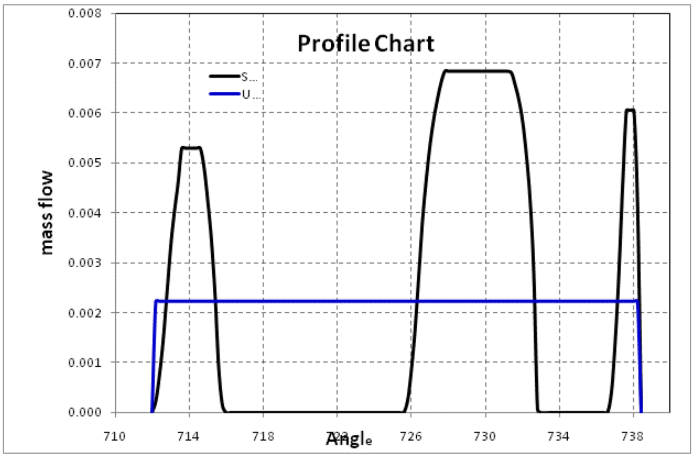
Figure 2: Profile chart

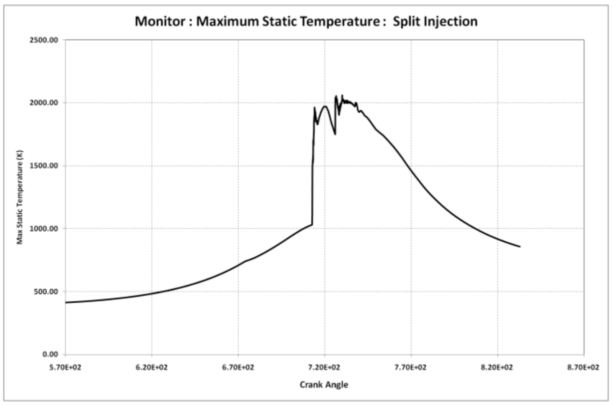
Boundary conditions & Solver settings:

Uniform wall temperature of 440 K is imposed on cylinder wall and 560 K is specified on piston top.



For species model, the standard Diesel Unsteady Flamelet model based on the work of Pitsch et al. and Barths et al. is used with 2 unsteady flamelets. Two equation k-e model is used to account the turbulence effects with standard wall functions based on the work of Launder and Spalding.





Results and Discussions:

Uniform injection profile shows a maximum static temperature and also significantly larger temperature at the start of EVO. Uniform injection also shows the spread of high temperature zone for longer crank angle duration.

Figure 3 & 4 shows the maximum peak temperature profiles.

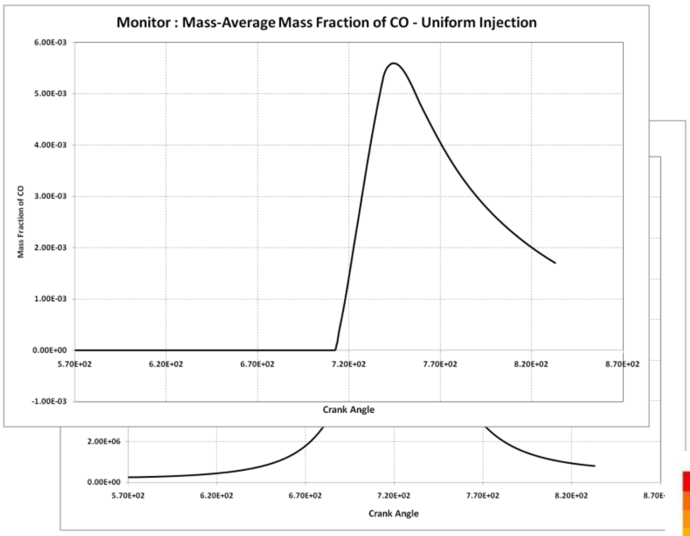


Figure 3: Maximum Static Temperature Uniform Injection

Figure 4: Maximum Static Temperature Split Injection

Figure 5: Maximum Static Pressure – Uniform Injection

In close similarity Uniform Injection leads to higher static pressure comparing to Split Injection.

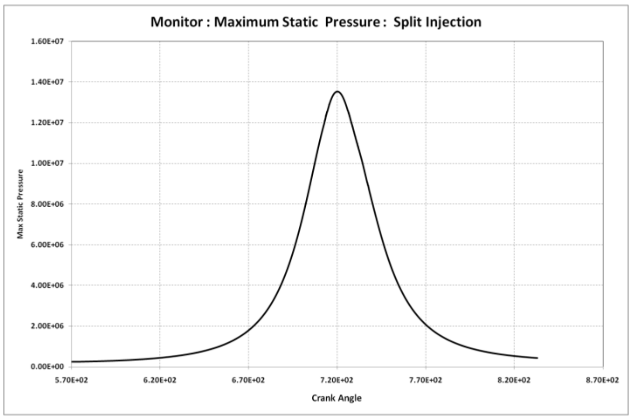


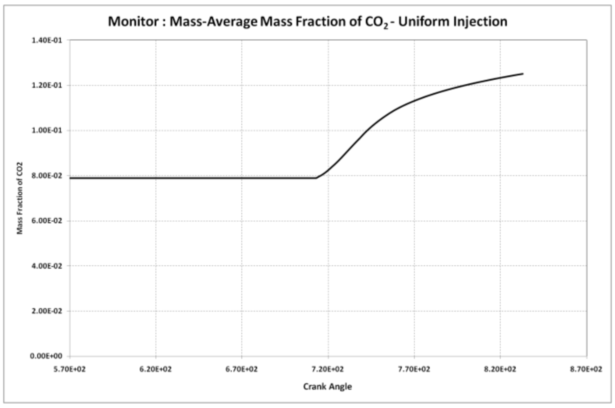
Figure 5 & 6 shows the variation of maximum static pressure in both uniform and split injections.

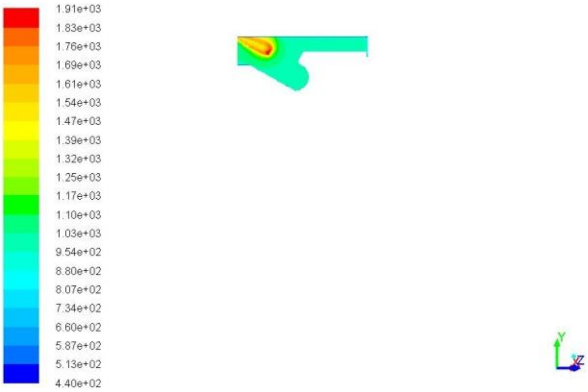
Figure 6: Maximum Static Pressure – Split Injection

Figure 7 & 8 shows the contour of temperature at 728^0 of crank angle i.e 16^0 after the start of injection.

Figure 7: Contours of Temperature – Uniform Injection -728^0 of crank angle

Figure 8: Contours of Temperature – Split Injection - 728^0 of crank angle



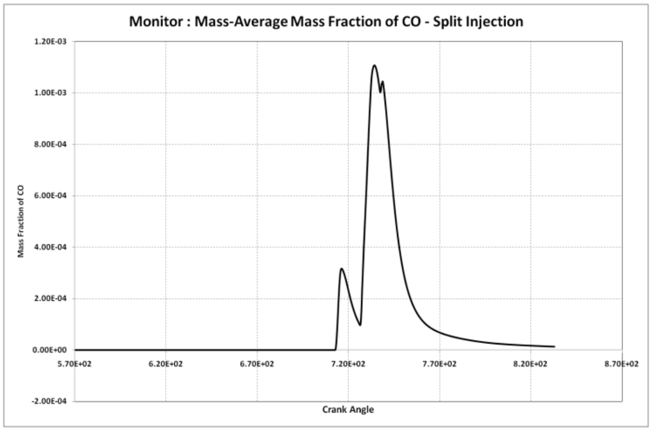


In a landmark study, laser-based diagnostics of Sandia's Combustion Research Facility (CRF) showed that smoky particulate matter, or PM, was formed in regions where fuel concentrations were too high. Another serious pollutant, nitrogen oxides, or NOx, arose from a high-temperature flame inside the engine. NOx emissions are toxic, and also they react with other pollutants to create ground-level ozone, or smog. In this numerical study also there is a good agreement is observed with peak temperatures and associated NOx content.

Figure 9 & 10 shows the comparison of the production of Carbon Monoxide (CO). Observation shows an approximate 80% reduction of CO in split injection without any compromise in fuel consumption.

Figure 9: Mass Fraction of CO – Uniform Injection

Figure 10: Mass Fraction of CO – Split Injection



From figure 11 to 14, it is clearly understood that here is a significant reduction in emission quantities (CO_2, NO_2) while using split injection system.

Figure 11: Mass Fraction of CO_2 – Uniform Injection

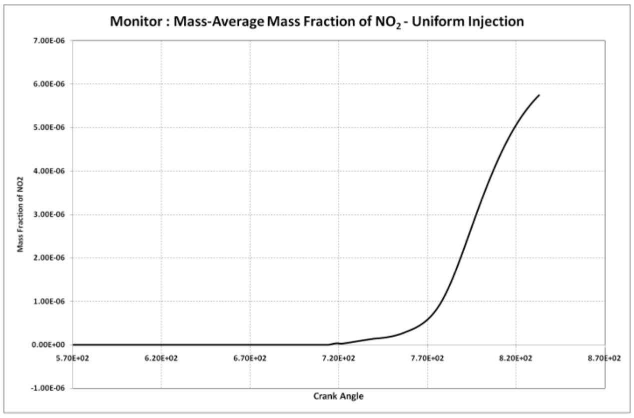


Figure 12: Mass Fraction of CO_2 – Split Injection

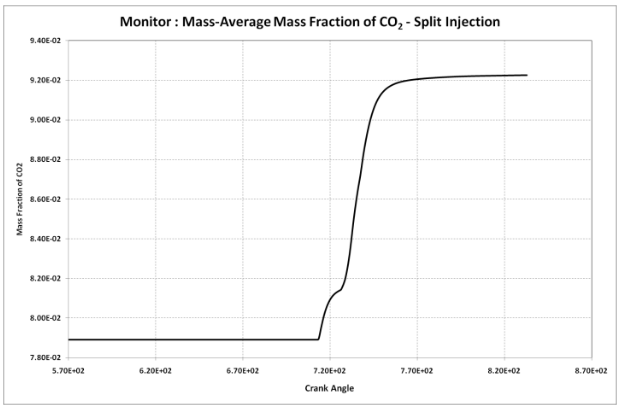


Figure 13: Mass Fraction of NO_2 – Uniform Injection

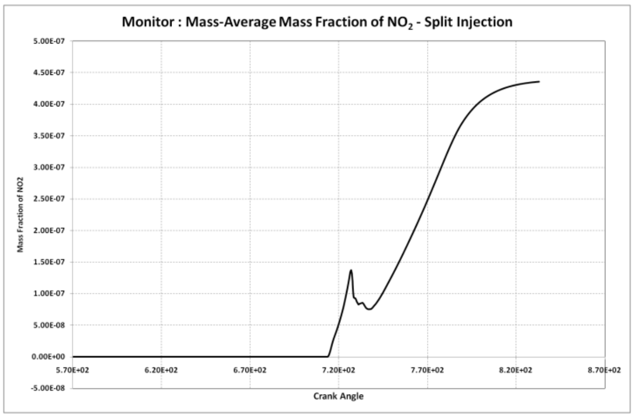


Figure 14: Mass Fraction of NO_2 – Split Injection

Conclusions:

CFD is used to investigate the emission reduction capability of a multi injection system in comparison to uniform injection system. Apart from the two conventional injections, advanced post injection is also introduced. Higher peak temperatures and also higher temperatures at the time of exhaust valve opening, were observed in uniform injection system in comparing with the split injection system. It was also observed that the volume of high temperature zone is more in uniform injection. Lower peak temperatures and a smaller volume of high temperature zone, resulted a significant reduction in emission constituents in multi injection system.

It is further advised to investigate the optimum split timings for the injection to further reduce the emission constituents.

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