Reducing Particulate and NO_X Emissions by Using Split Injection

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Abstract: ***In order to meet the stringent emission standards significant efforts have been imparted to the research and development of cleaner IC engines. Diesel combustion and the formation of pollutants are directly influenced by spatial and temporal distribution of the fuel injected. The development and validation of computational fluid dynamics (CFD) models for diesel engine combustion and emissions is described. The complexity of diesel combustion requires simulations with many complex interacting sub models in order to have a success in improving the performance and to reduce the emissions. In the present study an attempt has been made to predict the multi pulse injection on a high speed DI diesel engine performance and emission characteristics. The predictions have been made for both continuous (single) and split fuel injection for 75% load of maximum torque with 1600rpm. Fluent is capable of any number of injection per cycle which has inbuilt system like common rail injection. . A 0.6mm and 0.2mm diameter orifice were used to produce injection rise rates. The Predicted results have shown very good agreement with the existing experimental results. The results have shown. Particulate matter has been significantly reduced by a factor of three compared to single injection without any increase in NO_X for a long dwell between two injections***

Keywords: split injection, pilot injection, computational fluid dynamics, combustion, Fluent.

1. Introduction

The problem of diesel engine emissions is exacerbated because of the trade- off feature between NO_X and soot emissions. It is usually impossible to reduce both kinds of emissions simultaneously, since factors that tend to decrease one usually increase the other. For example, retarding the fuel injection timing is effective to reduce NO_X formation by reducing the peak cylinder temperature and pressure. However, this method results in an increase of soot production because more soot formed due to the lower in –cylinder gas temperature has shorter time to be oxidized [Lee, 2002]. Increasing the EGR rate can decrease the NO_X emission level, however less oxygen is available to oxidize soot .Eventually, any change in these engine parameters will unavoidably affect other important engine performance measures , like retarding injection timing causes lower thermal efficiency and higher Brake Specific fuel consumption(BSFC) [Hey Wood,1998].

Increasing environmental concerns and legislated emission standards have led to the necessity of considering both conventional and unconventional means for reducing soot and NO_X emissions in diesel engines, which is also a motivation of the present study. For example, diesel engine manufacturers are facing the challenges of the extremely low diesel engine –out soot emission mandates to be implemented in the near future. Engine simulation, compared to expensive engine experiments, is an efficient way to investigate various novel ideas to improve current engine performance, and hence becomes an essential part of engine research and development. In addition, simulations can investigate the transient properties of physical processes. However, adequate accuracy of modeling particulate matter emission remains a challenge. Soot formation in diesel combustion involves both gas phase and particulate reaction mechanisms, therefore, it is more compared than other pollutant species such as NO_X and CO. Current computing power capabilities and model parameter uncertainties in many of the diesel spray and combustion related mechanisms limit the possibility of using detailed chemistry description of the soot formation process. On the other hand, the widely –applied and highly –efficient empirical soot models have become less sufficient for the emerging demands for accuracy and detailed soot particulate information. For example, newly proposed emission mandates will specifically enforce the emitted soot particulate's size.

2. Literature Review

Multiple injections divide the total quantity of the fuel into two or more injections per combustion cycle. Splitting the injection sequence into two events is called pilot or split injection. A pilot injection is usually defined as an injection where 15% or less of the total mass of fuel is injected which reduces combustion noise and allow the use of poor ignition quality fuel (low cetane numbers) [2].

Many researchers are now investigating pilot and split injection as an effective means to simultaneously reduce NO_X and particulate emission. Shundhoh.et al. [3] reported that NO_X could be reduced by 35%, and smoke by 60 to 80 %, without a penalty in fuel economy if pilot injection was used in conjunction with high pressure injection. Yamaki.et al. [4] investigated the effects of pilot injection on exhaust emissions in a turbocharged heavy duty Diesel engine and found that with partial load, when the pilot fuel quantity was increased, Fuel consumption and smoke was increased, but NO_X was found to decrease and then increase. Minami et al. [5] studied the effects of pilot injection s in a turbocharged DI diesel engine and found that the pilot injection was effective to reduce NO_X and HC at low load conditions, through it deteriorated smoke to some degree. Zhang et al. [6] used a single cylinder HSDI diesel engine to investigate the effect of pilot injection with EGR on soot, NO_X and combustion noise, and found that the pilot injection increased soot emission. Nehmer and Reitz.et al. [7] Studied the effect of split injection in a heavy –duty diesel engine by varying the amount of fuel in first injection from 10 to 75% of the total amount of fuel. They found that split injection better utilized the air charge and allowed combustion to continue later into the power stroke than for a single injection case, without increased levels of soot production. Tow et al. [8] found that using a double injection with a relatively long dwell on a heavy duty engine resulted in reduction of particulate emissions by a factor of three with no increase in NO_X and only a slight increase in BSFC compared to a single injection. Han et al. [9] Multidimensional computations carried out to understand the mechanism of soot and NO_X emissions reduction in a heavy – duty diesel engine with multiple injections. The high momentum injected fuel penetrates to the fuel rich, relatively low temperature region at the jet tip and continuously replenishes the rich region, producing soot .However in a split injection, the second injection enters into a relatively fuel –lean and high temperature region that is left over from the combustion of first injection. Therefore, soot formation is significantly reduced. Tow et al. [8] pointed out that the dwell between injections was very important to control soot production and there would exist an optimal dwell at a particular engine operating condition. Durnholz.et al. [10] investigated the influence of pilot injection for a turbocharged and intercooled DI diesel engine for passenger cars. Their optimized pilot injection contained about 1.5 mm^3 of the fuel in the pilot injection independent of engine load and their optimal dwell was 15°CA. Fuchs and Rutland [1] found that high swirl ratios distributed the fuel such that it remained in the bowl, thus depleting almost all of the bowl oxygen during combustion. Therefore, they asserted that in high swirl ratio split injection cases the dwell should be optimized to prevent the second injection from landing in the fuel rich region left in the bowl from the first injection. D.A.Peirpont.et al. [11] Studied multiple injections are effective at reducing particulate. Two nozzle spray angles were used with included spray angles of 125° and 140° the results show that the combined use of EGR and multiple injections is very effective at simultaneously reducing particulate and NO_X. D. T. Montgomery et al. [12] observed the emissions and performance effects of exhaust gas recirculation (EGR) and multiple injections on the emission of oxides of nitrogen (NO_X), particulate emissions, and brake specific fuel consumption (BSFC) over a wide range of engine operating conditions. NO_X and particulate could be simultaneously reduced to as low as 2.2 and 0.07g /bhp-hr, respectively.

3. Objectives

The main objective of the present work is to improve the performance and simultaneous reduction of NOX and soot levels in the exhaust of high speed direct injection (HSDI) CI engine through simulation and experimentation by using split injection.

- Performance improvement of high speed direct injection (HSDI) CI engine
- Simultaneous reduction in NOX and soot levels in the exhaust

In order to achieve the above objectives the following methodology has been adopted.

4. Methodology

- Geometric model is created in GAMBIT (pre-processor)
- Mesh creation
- Exporting the model from GAMBIT to FLUENT
- Defining the models to be used for the simulation
- Applying boundary conditions
- Applying material properties
- Activate the species transport and include the diesel species with PDF
- Activate second order upwind scheme for iterations
- Perform the iterations to converge
- Post processing the results

5. Computational Fluid Dynamics

CFD is a sophisticated analysis technique that the analyst to predict transfer of heat, chemical reaction, and fluid flow behavior etc. CFD is based on the fundamental governing equations of fluid dynamics- the continuity, momentum, and energy equation. It is a powerful tool to carry out numerical experiments. This project uses the Computational Fluid Dynamics –FLUENT 6.3 software package. The process of utilizing FLUENT can be assumed in Firstly, the geometry and gird is created using GAMBIT. T Grid can be used to generate 2D triangular, 3D tetrahedral or 2D and 3D hybrid volumes mesh from an existing boundary mesh. Another alternative of creating grids for FLUENT is using ANSYS or IDEAS and Geo Mesh are the names of FLUENT Pre-processors that were used before the introduction of GAMBIT. Once a grid has been read into FLUENT, all remaining operations are performed with in the solver. These include setting the boundary conditions, defining fluid properties, and material properties, executing the solution, refining the grid, viewing and post- processing the results.

5.1 Governing Equation

In CFD, fluid flows are stimulated by numerical solving partial differential equations that governs the transport of flow quantities also known as flow variables. These variables include mass, momentum, energy, turbulent quantities, and species concentrations. In designing the POME- nozzle, the basic governing equations that will be used are the conservation of mass, momentum and energy equations.

5.2 Discritization Method

The method contains settings that control the discritization of the convection terms in the solution equations. It is a numerical method to solve the above equation by discritization to the partial differential Equations on a computational grid, the formation of a set algebraic equations and the solution of the algebraic equations. FLUENT allows choosing the discritization scheme for the convection terms of each governing equation. The numerical method is a discrete solution of the flow field, which is comprised of the values of the flow variables at the grid points. One of the most important terms that need to be discritized is convection. Second- order accuracy is automatically used for the viscous terms. The mathematical code uses a control volume technique to covert the governing equations that can be solved numerically. It consists of integration the governing equations about each control volume.

5.3 Upwind Scheme

Due to the computational domain, the initialized values are quite different from those expected in the final solution after the iteration process has begun. For this reason, first order UPWIND scheme is utilized until a more realistic solution is achieved, after which a more accurate second order UPWIND scheme could be implemented.

5.4 Application of CFD

CFD is useful tool in performing theoretical experimental validation. It solves all problems concerning fluid flows such as incompressible and compressible flow. Newtonian or non Newtonian flow , swirl , transfer of heat , in viscid , laminar and turbulent flow , radiation ,mixing , chemical reaction , spray models etc. CFD can be applied to solve industrial flow problems due to rapid growth of powerful computer resources and the development of CFD software packages. In engineering applications, it is much cheaper to use CFD than conventional design process. In CFD simulation, we can simulate different set of parameters for the same design without any additional cost. This reduces the time and cost of experimental work.

6. Mathematical Modeling and Simulation

To predict the parameters cycle peak pressures, heat release rate, temperature and the influence of different parameters on the formation of oxides of nitrogen, carbon monoxide, and soot using CFD technique, the following flow governing equations are to be solved.

6.1 Continuity and Momentum Equation

For all flows, FLUENT solves conservation equations for mass and momentum. For flows involving heat transfer or compressibility, an additional equation for energy conservation is solved. For flows involving species mixing or reactions, a species conservation equation is solved or if the non premixed combustion model is used, conservation equations for the mixture fraction and its variance are solved. Additional transport equations are also solved when the flow turbulent. The conservation equations relevant to heat transfer, turbulence modeling and species transport will be discussed here.

6.1.1 The Mass Conservation Equation

$$\frac{\partial p}{\partial t}+\nabla.\left(\rho\vec{v}\right)=S_m \quad (1)$$

Equation (1) is the general form of the mass conservation equation and is valid for incompressible flows. The source S_m is the mass added to the continuous phase from the dispersed second phase and any user –defined sources.

For 2D Axi -symmetric geometries, the continuity equation is given by

$$\frac{\partial p}{\partial t}+\frac{\partial}{\partial x}\left(\rho v_x\right)+\frac{\partial}{\partial r}\left(\rho v_r\right)+\frac{\rho v_r}{r}=S_m \quad (2)$$

Where x is the axial coordinate, r is the radial co-ordinate, v_x is the axial velocity, and v_r is the radial velocity.

6.1.2 Momentum Conservation Equations

Conservation of momentum in an inertial (non – accelerating) reference is given by

$$\frac{\partial}{\partial t}\left(\rho\vec{v}\right)+\nabla.\left(\rho\vec{v}\vec{v}\right)=-\nabla p+\wp\nabla.\left(\overline{\overline{T}}\right)+\rho\vec{g}+\vec{F} \quad (3)$$

Where p is static pressure, $\overline{\overline{T}}$ is the stress tensor (described below), and $\rho\vec{g}$ and $\vec{F}$ are the gravitational body force and external body forces respectively. $\vec{F}$ also contains other model –dependent source terms such as porous media and user defined sources.

The stress tensor $\overline{\overline{T}}$ is given by

$$\overline{\overline{T}}=\mu\left[\left(\nabla\vec{v}+\nabla\vec{v}^{T}\right)-\frac{2}{3}\nabla.\vec{v}I\right] \quad (4)$$

where μ is the molecular viscosity, I is the unit tensor, and second term on the right hand side is the effect of volume dilation.

For 2D axi-symmetric geometries, the axial and radial momentum conservation equations are given by

$$\frac{\partial}{\partial t}\left(\rho v_x\right)+\frac{1}{r}\frac{\partial}{\partial x}\left(r\rho v_x v_x\right)+\frac{1}{r}\frac{\partial}{\partial r}\left(r\rho v_r v_x\right)=-\frac{\partial p}{\partial x}+\frac{1}{r}\frac{\partial}{\partial x}\left[r\mu\left(2\frac{\partial v_x}{\partial x}-\frac{2}{3}\left(\nabla.\vec{v}\right)\right)\right]+$$
$$\frac{1}{r}\frac{\partial}{\partial r}\left[r\mu\left(\frac{\partial v_x}{\partial r}+\frac{\partial v_r}{\partial x}\right)\right]+F_x \quad (5)$$

And

$$\frac{\partial}{\partial t}\left(\rho v_r\right)+\frac{1}{r}\frac{\partial}{\partial x}\left(r\rho v_x v_r\right)+\frac{1}{r}\left(r\rho v_r v_r\right)$$
$$=-\frac{\partial p}{\partial r}+\frac{1}{r}\frac{\partial}{\partial x}\left[r\mu\left(\frac{\partial v_r}{\partial x}+\frac{\partial v_x}{\partial r}\right)\right]+\frac{1}{r}\frac{\partial}{\partial r}\left[r\mu\left(2\frac{\partial v_r}{\partial r}-\frac{2}{3}\left(\nabla.\vec{v}\right)\right)\right]$$
$$-2\mu\frac{v_r}{\partial r}+\frac{2}{3}\frac{\mu}{r}\left(\nabla.\vec{v}\right)+\rho\frac{v_z^{\,2}}{r}+F_r \quad (6)$$

Where

$$\nabla.\vec{v}=\frac{\partial v_x}{\partial x}+\frac{\partial v_r}{\partial r}+\frac{v_r}{r} \quad (7)$$

and v_z is the swirl velocity.

7.0 Computational Mesh

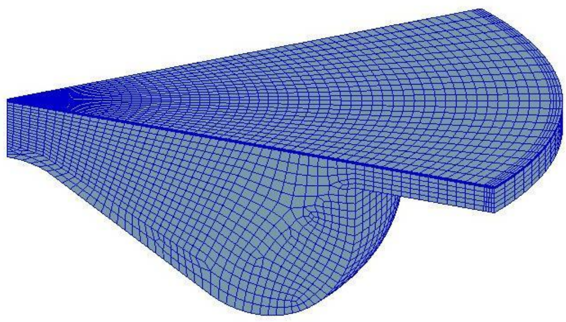

Figure 1: Mesh used for Simulation

7. Engine Specifications

Table 1

Injector type	Electronically controlled common rail injector
Injection pressure	Variable up to 120 M pa
Number of Nozzles	6
Nozzle hole diameter	0.2 mm and 0.6mm
Spray included angle	140°
Injection Approach	La-grangian
Turbulence model	RNG K-ε
Engine Type	Caterpillar 3406, Single cylinder Direct injection ,4 valve
Bore	137.2 mm
Stroke	165.2 mm
Compression ratio	15.1:1

8. Results and Discussions

AT 75% load condition - Figure 2 summarizes emissions results from the engine obtained using a single injection at 75% load. The particulate vs. NO_X tradeoff is seen as the injection timing was varied for both the fast rise rate of rise (S1) and the slower ramped rate of rise (S2) injections. Injection timing was varied over a range of -2° to -12° ATDC to generate a tradeoff curve. In all of the injection cases studied the NO_X always increased with advanced timing. However, the fast rate of rise (S1) produced lower particulate and the

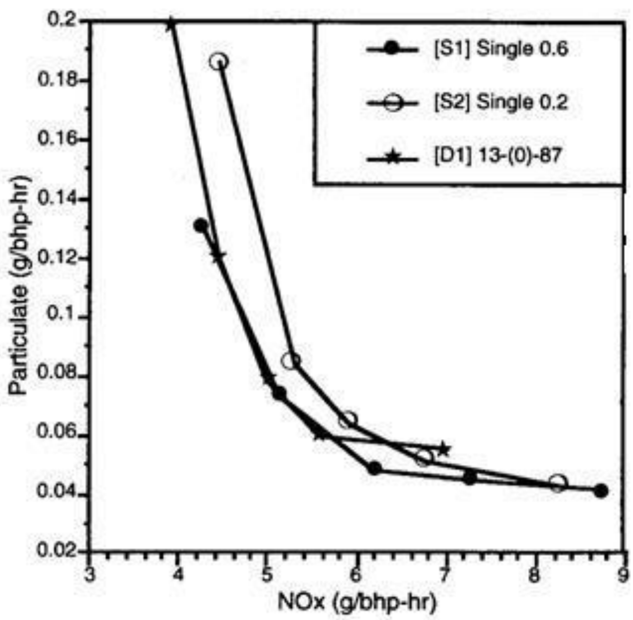

Figure 2: Particulate vs. NO_X tradeoff curves for single and double injections at 75% engine load

Slow rate of rise (S2) produced lower NO_X at a given start of injection condition. Double injections as noted above, one way to reduce NO_X emissions and peak cylinder pressures is by slowing the initial rate of fuel injection. Figure 2 shows the particulate vs. NO_X emissions for a double injection with a zero dwell (D1) after 13% of the fuel is injected. Neither the slow rate of rise (S2) nor the split case (D1) improved the emissions tradeoff as compared to the fast rate of rise single injection (S1). Figures 3a, 3b, & 3c show the cylinder pressure, apparent heat release rate, injection rate for these cases. As can be seen by comparing the cylinder pressures and the size of the peaks in the apparent rate of heat release diagrams, both S2 and D1 reduced the peak cylinder pressure and magnitude of the premix burn (a source of noise and NO_X formation). However S2 produced significantly higher particulate levels at lower NO_X levels.

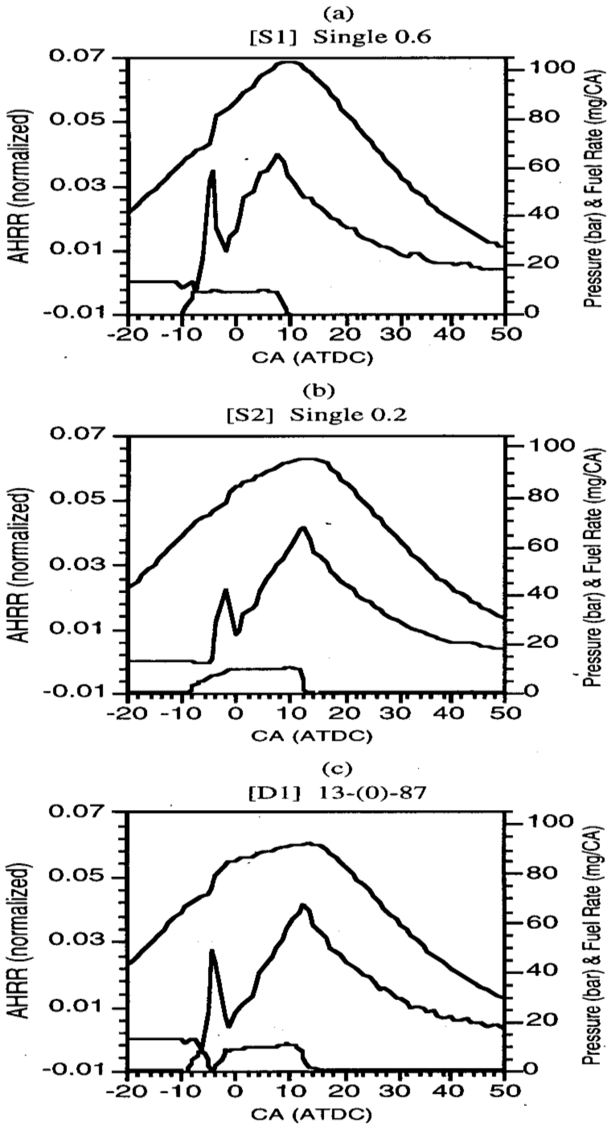

Figure 3 Cylinder Pressure (top curve), Apparent Heat Release Rate (middle curve), and Injection Rate (bottom curve) for single and double injections at 75% load

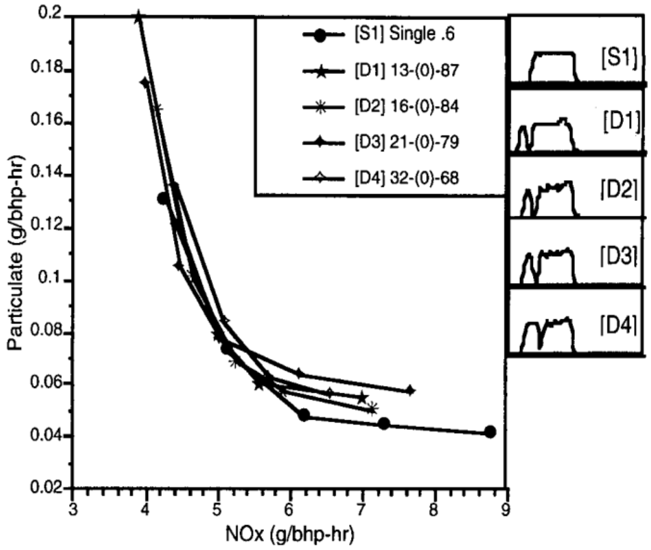

Figure 4 Particulate vs. NO_X tradeoff curves for double injections at 75% load. Start-of-injection timings varied **from -14° to -2° ATDC**

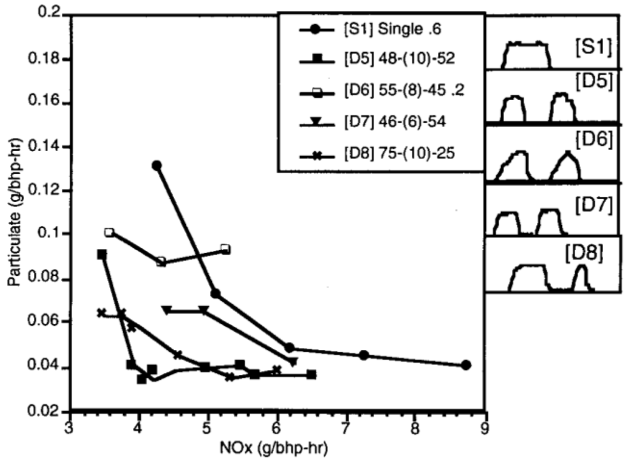

Figure 5 Particulate vs. NO_X tradeoff curves for double injections at 75% load

Fig 4 shows the effect on particulate and NO_X emissions when the fuel quantity in the first injection was varied. Within simulation repeatability, the quantity is seen to have little effect on the overall particulate vs. NO_X tradeoff. From these results, it appears that a double injection with zero dwell between injections behaves very similarly to a single injection, regardless of the fuel amounts in the first and second pulses. However, when the dwell time between injections, was increased, particulate levels were found to be lowered significantly.

Figure 5 illustrates the considerable reduction in particulate levels that resulted when a longer dwell.

Start-of-injection timings varied from -12° to 1° ATDC between injection periods was used. A ratio of approximately 50/50 fuel mass in the first to second injection was used in cases D5, D6, and D7 whose dwells were 10, 8, and 6 crank angle degrees, respectively. The difference in particulate level between D5 and D7 shows the effectiveness of a relatively long (10° CA) dwell which produces a combustion process in which the particulate emissions do not increase significantly with timing retard.

9. Summary and Conclusions

The results of this study show the following specific conclusions:

- Particulate reductions by a factor of three with no increase in NO_X and only 2.5% increase in BSFC compared to single a injection were found at 75%, load using a double injection with a relatively long dwell between injections (case D5)

- Reduction in NO_X and peak pressure at 25% and 75% load were found with small first quantity double injection (D1). Ramped injection (case S2) was not as effective

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