Acta Mechanica Malaysia (AMM)

DOI : http://doi.org/10.26480/amm.1.2019.23.28

ZIBELINE INTERNATIONAL PUBLISHING

ISSN: 2616-4302 (Online)
CODEN : AMMCFL

RESEARCH ARTICLE

CFD ANALYSIS ON DIFFERENT PISTON BOWL GEOMETRIES BY USING SPLIT INJECTION TECHNIQUES

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ARTICLE DETAILS

Article History:

Received 01 April 2019
Accepted 15 May 2019
Available online 21 May 2019

ABSTRACT

A better spatial distribution is required for the injected fuel throughout the entire space of combustion geometry in DI diesel engine, to obtain a better combustion with lesser emission. In order to effectively make use of gas flows it is mandatory to match the piston bowl geometry with fuel spray characteristics. For obtaining better combustion, matching of combustion chamber geometry, fuel injection and gas flow plays prominent role. The model was developed in computational fluid dynamics (CFD) code, ANSYS FLUENT. The simulations were conducted for the combinations of swirl ratio, three split injection and four piston bowl geometries (Case A,B,C and D). The simulation results revealed that, the pilot injection creates favorable condition for the upcoming main fuel injection in the case of multiple injection. In case of multiple injections there is a considerable reduction in NO formation. The final NO formed is 24% lower than that of normal injection and 22% lower than that of the retarded injection. The combination of split injection with suitable combustion chamber configuration would greatly enhance the engine performance, besides reducing emission level to a greater extent.

KEYWORDS

DI Diesel engine, Split injection, Combustion, piston bowl configuration, in-cylinder air motion, NOx formation

1. INTRODUCTION

Apart from different piston bowl configurations and fuel injection timings, split injection impact a significant role in the DI diesel engine operation. The Common rail injection approach has the ability to control the injection pressure, temperature and heat release rate independent of the engine speed and load. The CR injection with its adaptable classification of fuel injection in bowl-in-piston geometry into several advanced, conventional and retarded injections, allows the fuel injecting system and engine would match each other in a best possible way. The pollutant formation in diesel engine cylinder greatly depends on fuel injection timing and pattern, in addition to the in-cylinder flow dynamics. The effect of piston bowl configurations and various injection timings i.e., Early, Conventional and Retarded) on in-cylinder fluid flow characteristics and formation of the pollutants are discussed elaborately in the previous cases [1]. It is therefore necessary to analyze the effect of multiple injection and fuel injection pattern on the pollutant formation. The central bowl is identified as the best-suited piston blow, with regard to fuel air mixing and NO formations, is considered for the analysis.

Yao Mingfa et al (2009) conducted experiments on a multi-cylinder diesel engine to emphasize as well as the effects of coupling of multi-injection and EGR (exhaust gas recirculation) on diesel emissions and performance. At three loads, BMEP were found to be 1.55 MPa, 1.17 MPa and 0.38 MPa, which were tested at 100%, 75% and 25% of the maximum load, when the engine ran at 1849 rpm. However, it seems that double pilot injection could reduce smoke and increase NOx. In this sense, it seems that double-pilot injection can improve in-cylinder air utilization, promoting fuel-air mixing process and obtains better fuel-air mixture. Mingfa et al, an obvious reduction of smoke and CO can be achieved through post-injection. NOx emission can also be improved at the same time. Higher post injection fuel mass can get better results, but with a penalty on BSFC. Pilotmain-post injection could further reduce emissions slightly. Tobias Husberg et al (2008) by using optical instruments and with simulation analysis on heavy-duty engine investigated the imapct of strategies with multiple injections. CFD code STAR-CD was used in numerical computations with diesel surrogate chemistry implemented for the multiple representative interactive flameless models. The purpose of carrying the above exercise is to reduce the exhaust emissions in order to make sure that the injected fuel should not ignite before the start of the combustion process as the fuel-air mixture is not properly mixed. As a consequence, prevention of heat release rate is possible prior to the main injection. Genzale et al (2007) carried out a computational investigation to study the effects of bowl geometry, fuel spray targeting, and swirl ratio for low-temperature combustion in a heavy-duty diesel engine. To simulate the process on combustion engine a modified version of KIVA-3V code was used to optimize the designed parameters multi-objective algorithm an optimization code was used. The effect of bowl geometry was manifested mainly as an interaction with swirl ratio levels. Large diameter bowls were observed to require higher swirl levels in order to achieve similar soot reduction benefits seen in small bowl designs [2-5]. Small bowl designs were observed to be more sensitive to variations in swirl, than large bowl designs.

Takashi Kaminaga and Jin Kusaka (2006) presented a combined experimental and numerical study on combustion and exhaust gas emissions in a passenger car diesel engine by optimizing the combustion chamber design. The results showed that, by introducing Pa-PCI combustion to medium load, up to 88% improvement of NOx emission was achieved. The EGR application improved the thermal efficiency and emission at full load condition. The larger boost pressure is required as the EGR amount increases to improve the intake air quantity. Fontanesi et al (2005) carried out both experimental and numerical investigation on mixture formation in a heavy-duty direct injection diesel engine (HDDI) for marine application. The results showed that the modified combustion chamber increases the swirl intensity and turbulence level. The Modified combustion chamber improved air usage and better air fuel mixing. The modified combustion chamber geometry improved the BSFC by around 10 % and reduced the soot by 16% and slightly increased in NOx (3%) emission. Kulkarni et al (2005) achieved the optimization of flow capacity by conducting paramedic study on inlet port design. Different helical port with different geometrical models were studied and validated with CFD code. They concluded that larger valve inner seat diameter and lesser fire-deck angle are good for better flow capacity of the port. Finite Element Technique (FET) is used to study the contact stress between valve and seat assembly with variation in valve seat angle. It is observed that 300 seat angles is better for seat wear. Few of the vital conclusions drawn from the literature survey include: Swirl is a key parameter that impacts on the air-fuel mixing ratios, combustion quality, heat release, overall engine performance and emissions. For a specific combustion chamber, optimal swirl ratios are determined. These swirl levels can be changed with respect to the suitable variations in design of intake system. In this work, an effort is made to arrive at better injection strategies with focus on emission reduction. Basically, three fuel injection strategies viz. Normal-injection (conventional), Retarded-injection and Multiple injections are considered. From the previous analysis, it is clear that in the case of early injection, there is an ignition delay of 50 crank angles. The timings for pilot injection and main injection events are carefully designed, such that the internal pressure rise due to combustion starts a little before TDC.

2. MODEL DEVELOPMENT

In order to analyse the process of combustion, it is attempted to solve an first order implicit transient state with highly disorganised effects to analyze the process of combustion for compression ignition, Direct injection diesel engine [6-9]. A species transport model with discrete phase injection with finite rate chemistry and species transport equation has been used for solving non pre-mixed combustion with non-reversible adiabatic equilibrium phase model. Injection of fuel is specified at crank angle position at which fuel injection start and stop. During the process of injection itself, the fuel parameters like size of injector, location of injector, injection temperature, injection pressure and mass flow rate are defined in the figure 1. Based on the complexity of governing equations, Ansysfluent15 converts governing equations into algebraic equations by following control volume based technique which can solve numerically. In order to solve any heat transfer problem analysis following conservation principles were used (i.e., continuity, I law of thermodynamics and Newton's II law of motion) and initiated with appropriate boundary conditions for the combustion analysis.

2.1 Geometrical Model

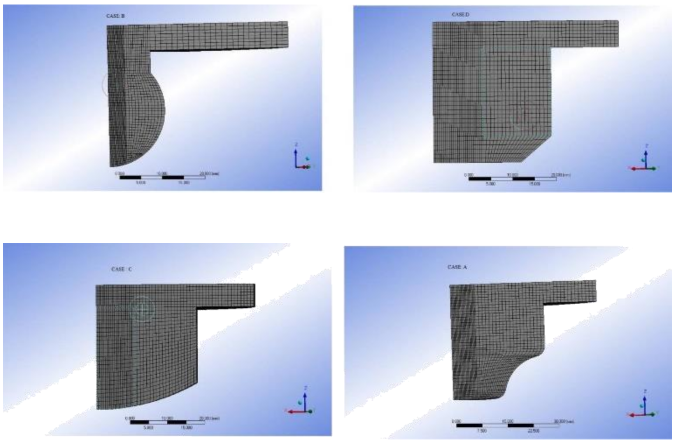

Figure 1: Computational Domain for different piston Bowl geometries

2.2 Turbulence model

Fluctuations in the velocity field is ascertained by using turbulence model. The two-equation model such as the K-ω model and K-Ɛ family use the Boussinesq approximation. Along with the Specific dissipation rate ω, or the turbulence dissipation rate Ɛ, transport equation is solved for the turbulence kinetic energy K. The turbulent eddy viscosity is then concluded as a function of K and Ɛ or K and ω. The great advantage of the two equation models over one equation model is that they can compute both the turbulent velocity and the turbulent length scales, thus they are vastly more applicable to flows with variances in length scales.

$$\frac{\partial}{\partial t}(\rho k)+\frac{\partial}{\partial x_j}(\rho k u_j)=\frac{\partial}{\partial x_j}\left[(\mu+\frac{\mu_t}{\sigma_k})\frac{\partial k}{\partial x_j}\right]+G_k+G_b-\rho\varepsilon-Y_M+S_k \quad (1)$$

$$\frac{\partial}{\partial t}(\rho\varepsilon)+\frac{\partial}{\partial x_j}(\rho\varepsilon u_j)=\frac{\partial}{\partial x_j}\left[(\mu+\frac{\mu_t}{\sigma_\varepsilon})\frac{\partial \varepsilon}{\partial x_j}\right]+\rho C_1 S_\varepsilon-\rho C_2\frac{\varepsilon^2}{k+\sqrt{v\varepsilon}}+C_{1\varepsilon}\frac{\varepsilon}{k}C_{3\varepsilon}G_b+S\varepsilon \quad (2)$$

2.3 Combustion Modeling

Based on the implementation of modified eddy dissipation conept into the CFD code, combustion/ignition model is considered for the simulation analysis. Simultaneously due to the occurrence of several reactions in bulk phase on wall surfaces, chemical reactions can also be resolved. Conservation equation in general form can be expressed as:
Where Ri is the net production rate of species i, Si is creation rate from the dispersed phase. the rate of creation by addition from the dispersed phase.

$$\frac{\partial}{\partial t}(\rho Y_i)+\nabla.(\rho\bar{v}Y_i)=-\nabla.\vec{J_i}+R_i-S_i \quad (3)$$

2.4 Engine Ignition Modeling

The most suitable method for simulating a direct injection diesel engine is Hardenberg model (auto-ignition model). The transport equation an ignition species is given by:

$$\frac{\partial}{\partial t}(\rho Y_{ig})+\nabla.(\rho\vec{v}Y_{ig})=-\nabla.\frac{\mu_t}{S_{ct}}(\nabla Y_{ig})+(\rho S_{ig}) \quad (4)$$

here Yig is a "mass fraction" of a passive species representing radicals which form when the fuel in the domain breaks down.

2.5 Spray Break Up model

In the present analysis, Taylor Analogy Breakup (TAB) model was used. Oscillating, distorting droplets and spring mass system are some of the analogies considered for the analysis in the TAB model. In the present scenario of research work on split injection, the droplet effect with distortion concept is considered. The equation governing is damped, force oscillator is

$$F-kx-d\frac{dx}{dt}=m\frac{d^2x}{dt^2} \quad (5)$$

$$\frac{F}{m}=C_F\frac{\rho_g\mu^2}{\rho l^r},\frac{k}{m}=C_k\frac{\rho}{\rho l r^3},\frac{d}{m}=C_d\frac{\mu_l}{\rho l r^2} \quad (6)$$

Where I and a are the discrete phase and continuous phase densities, Vr was droplets relative velocity, r was radius of the droplet undisturbed, was surface tension.

2.6 Droplet Collision Model

For estimating the number of collision of droplets, droplet collision model is considered for tracking of droplets and the net outcomes in a computationally efficient manner. The O'Rourke's method is applied on droplet collision model which assumes collisions are based on stochastic approximation. When two groups of particles collide an algorithm further establishes the type of collision. Only commingle and indefatigable outcomes are measured. The possibility of each output was evaluated from the dimensionless number, Weber Number and fit to experimental observations. The Weber number can be expressed as:

$$W_e=\frac{\rho V_r^2 l}{\sigma} \quad (7)$$

Where Vr is the relative velocity between two parcels and l is the arithmetic mean diameter of the two parcels.

2.7 Engine Specifications

On account that of the symmetrical vicinity of the injector on the middle of the combustion chamber considered in figure 1, the CFD calculations are carried out on a 600 sector. Exhaust and inlet valves aren't incorporated in the computational mesh by concentrating this simulation on in cylinder drift and combustion approaches. Simulations are carried out from the inception of suction stroke (0 CAD) and end at the analysis of combustion stroke. Similar parameters and boundary conditions are used for the entire computations as mentioned in table 1.1.

Table 1: Engine Specifications

Engine specifications	Kirloskar Single cylinder, 4S diesel,
Cylinder bore	87.5 mm
Stroke length	110 mm
Displacement volume	2440 cm3
Compression ratio	17.5: 1
Power	5.2 kW
Engine speed	1500 rpm
Connecting rod length	234 mm
No: of fuel injectors	01
No: of nozzles	04
Injection approach	Lagrangian
Injector nozzle bore	0.3 mm
Fuel mass flow rate	0.1361 Kg/min
Injection duration	20 CAD

3. RESULTS AND DISCUSSION

3.1 Effect of Injection Strategies on Temperature

Figure 3-6 presents the predicted mass-averaged temperature vs. crank angle Degree for the three multiple injection strategies viz., 10%, 75%, 90%. The rise in temperature due to compression process is observed to be gradual and the same in all the three cases for different piston bowl configurations until combustion is initiated. Drop in temperature in the case of 10% and 75% is due to the evaporation of liquid fuel. In the case of 90%, this drop is not noticed for the reason that the quantity of pilot fuel injected is small to affect the mass averaged temperature through evaporation.

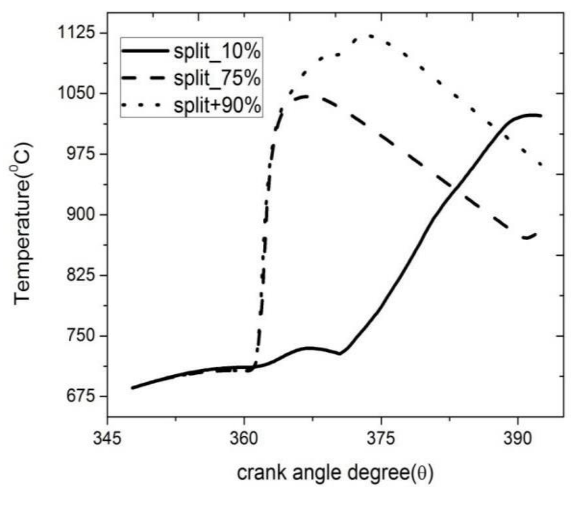

Figure 3: Effect of Injection Strategies on Mass Averaged Temperature Vs Crank angle degree for CASE A

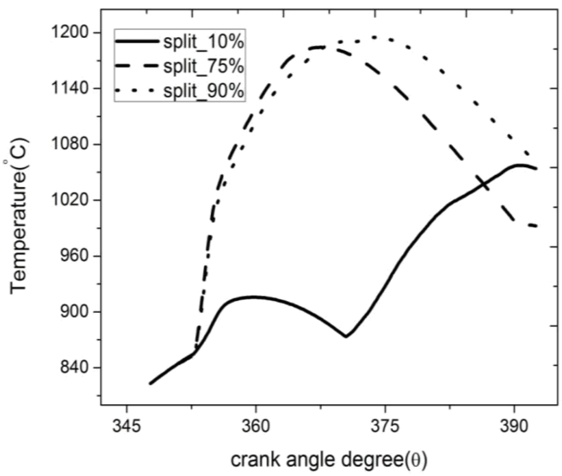

Figure 4: Effect of Injection Strategies on Mass Averaged Temperature Vs Crank angle degree for CASE B

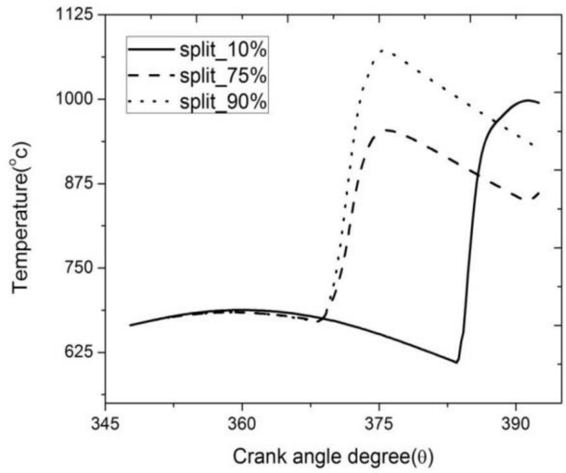

Figure 5: Effect of Injection Strategies on Mass Averaged Temperature Vs Crank angle degree for CASE C

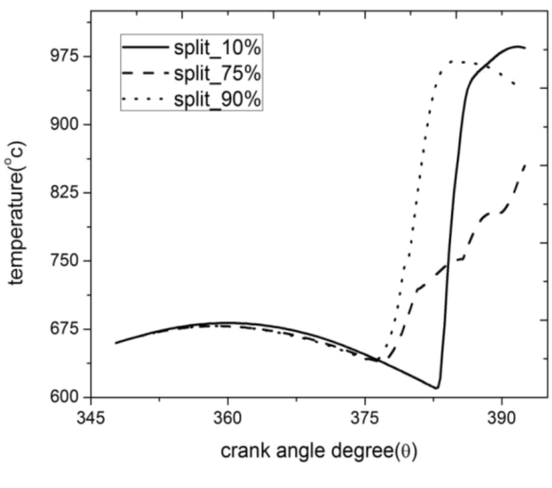

Figure 6: Effect of Injection Strategies on Mass Averaged Temperature Vs Crank angle degree for CASE D

In the case of Multiple-injection, the rise in temperature is noticed to start at about 50 bTDC due to the combustion of pilot fuel, in the case of Normal-injection at about 10 bTDC and in the case of Retarded-injection at about

40 aTDC.

The pilot injection appears to shorten the ignition delay period of the main fuel and creates favorable conditions for mixing-controlled combustion of the main fuel, resulting in lower peak heat release rates in turn peak temperatures in the case of Multiple-Injection. Higher temperature noticed beyond 260 aTDC in the case of Multiple-injection is due to the continued tail combustion.

3.2 Effect of Injection Strategies on Mass-Averaged Pressure

Figure 7-10 presents predicted volume-averaged cylinder pressures vs. crank angle profiles from 200 bTDC in compression stroke to 500 aTDC in expansion stroke for the three split injection phenomena.

The pressure rise due to compression process is noticed to be gradual increasing and the same in all the three cases until 45 bTDC. This can be attributed that the evaporation of liquid fuel (endothermic reaction). A similar trend could not be noticed in the case of Multiple-injection for the reason that the quantity of pilot fuel injected is very small and its evaporation cannot greatly influence on the in-cylinder temperatures and in turn on pressure[10-12]. An early pressure rise, at about 40 bTDC, is noticed in the case of Multiple-injection indicating combustion of pilot fuel injection.

After attaining the peak pressures, the pressure in Multiple injection case is noticed to be higher during the rest of the expansion stroke than the other two cases. This clearly indicates that Multiple injection strategy is beneficial in limiting the peak pressure while increasing the mean effective pressures.

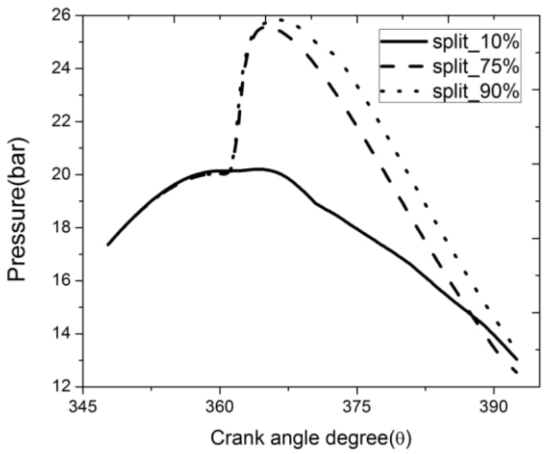

Figure 7: Effect of Injection Strategies on Volume Averaged Pressure Vs Crank angle degree for CASE A

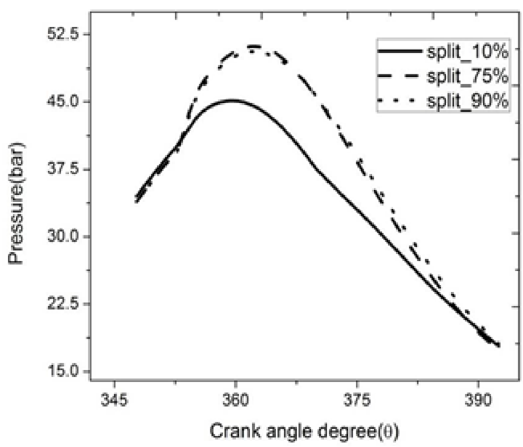

Figure 8: Effect of Injection Strategies on Volume Averaged Pressure Vs Crank angle degree for CASE B

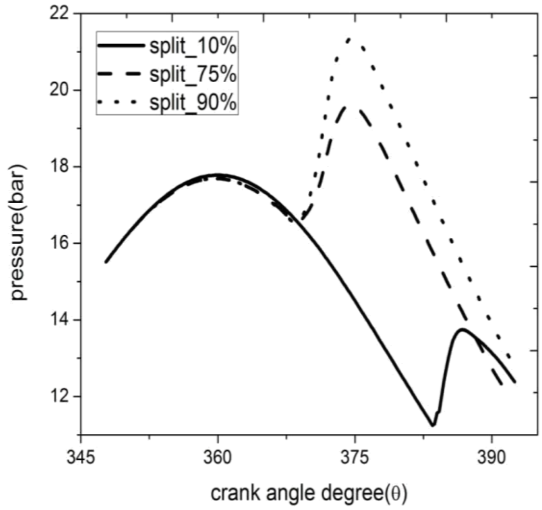

Figure 9: Effect of Injection Strategies on Volume Averaged Pressure Vs Crank angle degree for CASE C

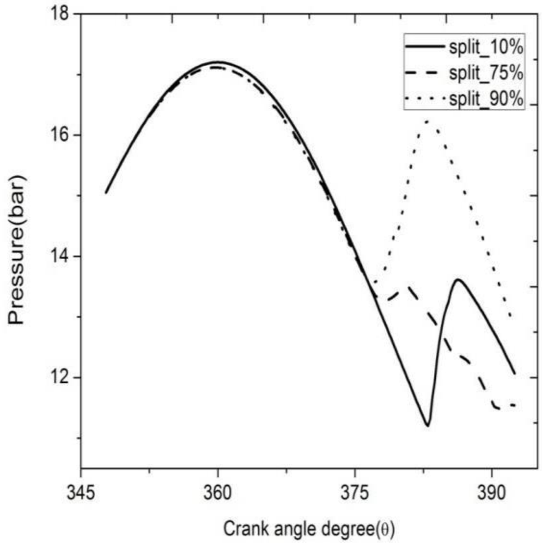

Figure 10: Effect of Injection Strategies on Volume Averaged Pressure Vs Crank angle degree for CASE D

3.3 Effect of Injection Strategies on NO

The present work is focused mainly on study of NO reductions. Figure 11-14 presents the predicted NO vs. crank angle profiles for the three injection strategies.

From the NO plots, it is also evident that a small portion of NO is getting decomposed after the peak value is attained, in the cases of Normal-injection and Retarded-injection. Whereas in case of multiple injection there is no reduction in NO concentrations. This is mainly due to the extension of tail combustion.

Fuel that burns early in the combustion process (premixed combustion) especially causes a higher local temperature, in turn causes higher NO formations higher in the cases of Normal-injection and Retarded-injection. But in the case of Multiple injection, the lowest NO formation is mainly due to suppression of premixed combustion and enhanced diffused combustions.

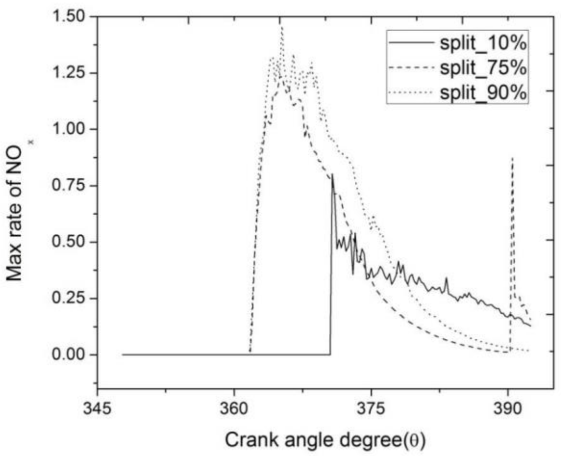

Figure 11: Effect of Injection Strategies on Rate of NOx Vs Crank angle degree for CASE A

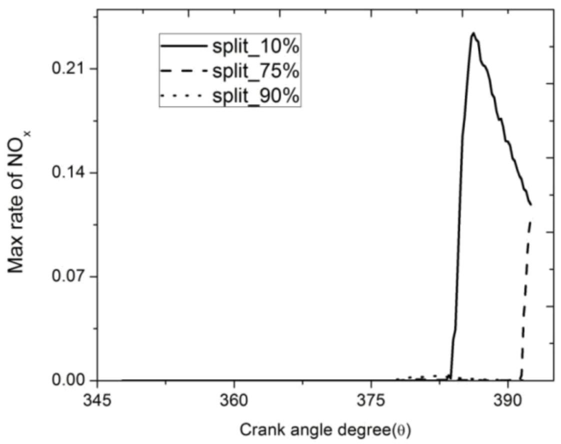

Figure 12: Effect of Injection Strategies on Rate of NOx Vs Crank angle degree for CASE D

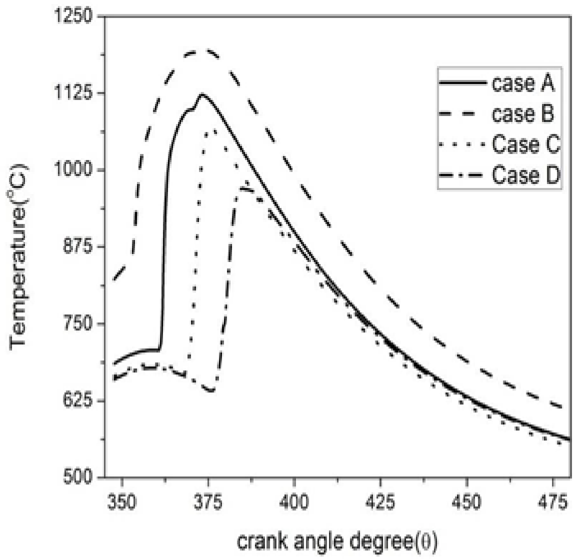

Figure 13: Effect of Injection Strategies on Mass Averaged Temperature Vs Crank angle degree for different Piston Bowls

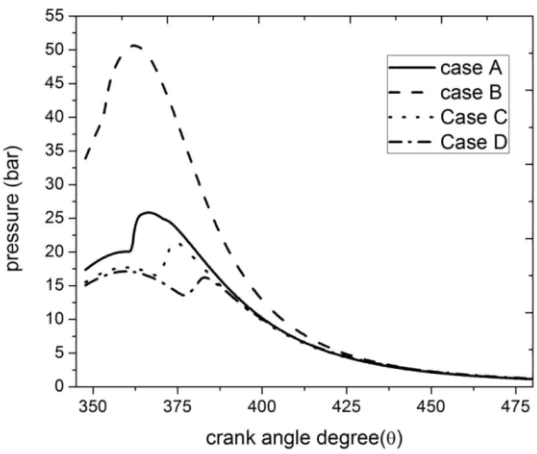

Figure 14: Effect of Injection Strategies on Volume Averaged Pressure Vs Crank angle degree for different Piston Bowl

4. CONCLUSIONS

In Multiple-injection strategy, fuel injection is split into two pulses of injections - a small quantity of fuel injected prior to main fuel injection is called as pilot injection followed by the main injection with the small time delay.

- Pilot injection creates favourable conditions for the upcoming main fuel injection thereby increasing combustion quality for the main injection.
- In Multiple-injection strategy a smooth rise in pressure profile is noticed and it indicates smooth engine operation.

- The overall performance with Multiple-injection is comparable to that of the other two cases as observed in the P-θ diagram.

- From the heat release rate profiles, it is observed that the case of Multiple-injection, the premixed combustion phase is suppressed and diffusion combustion phase is enhanced.

- It is also observed that in the case of Multiple-injection there is a considerable reduction in NO formations. The final NO formed is 24% lower than that of Normal-injection and 22% lower than that of Retarded-injection.

Finally, it can be concluded that the Multiple-injection strategy results effective combustion and more uniform in-cylinder the temperature distribution. Hence, the overall output performance is higher and formations of NO is could be reduced considerably.

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