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Effects of Injection Timing and EGR on DI Diesel Engine Performance and Emission-using CFD

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Abstract: The aim of present study was to investigate the effect of early injection and EGR on engine performance and emission. This entire computational study was modelled by using the commercial CFD code - STAR-CD. Using this code, the timing of early injection and EGR fraction was varied to investigate their effect on pollutant emissions. The injection timing and EGR fraction was also optimised for better engine performance and Emissions. Due to stringent emission norms, there is an immediate need to address the issue of emission control from automobile exhaust and related technologies. NOx and PM are the major pollutants from Diesel engine exhaust. The fuel injection system is the heart of the diesel engine and with the advance of electronically controlled fuel injection, multiple injection, High injection pressure, retarded injection timing, EGR and high swirl ratio have all been used to improve performance and reduce emissions. Combustion of fuel can also be optimised leading to leaner operation and lesser emissions. Each technology has its own merits and demerits and there exists strong interaction between these methods when they are simultaneously employed.

Key words: Computational fluid dynamics, nitrogen oxide, particulate matter, top dead centre, bottom dead centre

INTRODUCTION

DI Diesel engines, having the evident benefit of a higher thermal efficiency than all other engines, have served for both light- duty and heavy-duty vehicles. However direct injection diesel engines emit more particulates and oxides of nitrogen than their counterpart and hence a reduction of such emissions is most urgent. As a result, many technologies such as high injection pressure, multiple-injection (Okude *et al*., 2007) retarded injection timing, EGR, HCCI mode operation (Miyamoto *et al*., 1999) and high swirl ratio have been used in high efficiency DI diesel engines in order to reduce the pollutant emissions. The Common Rail (CR) fuel injection system offer very high injection pressure which can reduce the emission of particulate matters due to improved spray atomization and air-fuel mixing (Flaig *et al*., 1999). However, the high injection pressure will increase the NOX emission due to high peak temperature and oxygen rich regions near the beginning of the combustion process. The application of both Exhaust Gas Recirculation (EGR) and retarded injection timing in a diesel engine can significantly reduce NO_x emission but increase the soot (Han *et al*., 1996). The coupling of these technologies like multiple injection and EGR (Mingfa *et al*., 2009), common rail injection with EGR (Millo *et al*., 2009) can improve the engine performance and reduce the emission but there is always tradeoff between NO_x and smoke level.

The interaction between injection timing and EGR needs to be considered when they are applied to a HSDI diesel engine. This combination of injection timing and EGR achieve simultaneously reductions in both NO_x and smoke level.

Multi-dimensional engine models or Computational Fluid Dynamic (CFD) models have progressed to the point where three-dimensional analyses of the cold flow within the combustion chamber are becoming practical enough to be employed as part of the actual engine design process. The modeling approach to be described is relatively simple in nature, which offers advantages related to reduce computational expense and ease of implementation within existing engine CFD methodologies (Jennings, 1992). Multi-dimensional modeling has been shown in several previous studies to be a useful tool for engine design and optimization, as well as for gaining a better physical understanding of the in cylinder combustion process (Reitz *et al*., 2001). Multi-dimensional models provide detailed geometric information on the flow field based on solution of the governing flow equations

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(Heywood, 1988). Multi-dimensional CFD codes solve the full set of differential equations for species mass, momentum and energy conservation and also account for the effects of turbulence. These models are best suited to analyze the various processes of mixture formation and combustion in greater detail (Stiesch, 2003). In multi-dimensional models the time dependent, instantaneous conservation equations are time averaged and the turbulence correlations are considered to be proportional to the gradients of the mean flow. Multi-dimensional models of diesel engine combustion, account for temporal and spatial variations of the flow field, pressure, temperature, composition and turbulence within the combustion chamber (Ramos, 1989; Kong *et al*., 1995).

This study presents a effect of injection timing and EGR on DI diesel engine combustion performance and Pollutants formation using the Computational Fluid Dynamics code STAR-CD.

MATHEMATICAL FORMULATION

The model is based on the numerical simulation of time averaged conservation equations of the transport processes of heat, mass and momentum of the gas phase within the cylinder. A standard RNG k-ε model for the turbulence with wall function treatment for near wall region has been adopted for the solution of conservation equations. The process of combustion in the cylinder after injection of fuel is modelled by dispersed Lagrangian multiphase model. The chemical reaction rate is modelled using global turbulent controlled reaction kinetics (eddy break-up model). The formation of thermal NOx is modelled by the Zeldovich mechanism. The mass, momentum and energy conservation equations solved for general incompressible and compressible fluid flows in Cartesian tensor notation are expressed as:

Continuity equation:

$$\frac{\partial \rho}{\partial t}+\frac{\partial(\rho u)}{\partial x}+\frac{\partial(\rho v)}{\partial y}+\frac{\partial(\rho w)}{\partial z} \quad (1)$$

Momentum equation:

$$\rho\frac{Du}{Dt}=\frac{\partial(-p+\tau_{xx})}{\partial x}+\frac{\partial \tau_{yx}}{\partial y}+\frac{\partial \tau_{zx}}{\partial z}+S_{Mx} \quad (2)$$

Energy equation:

$$\frac{1}{\sqrt{g}}\frac{\partial}{\partial t}\left(\sqrt{g}\rho h\right)+\frac{\partial}{\partial x_j}\left(\rho \tilde{u}_j h-F_{h,j}\right)=\frac{1}{\sqrt{g}}\frac{\partial}{\partial t}\left(\sqrt{g}\rho\right)+\frac{\partial}{\partial x_j}\left(\tilde{u}_j p\right)-p\frac{\partial u_j}{\partial x_j}+\tau_{ij}\frac{\partial u_i}{\partial x_j}+s_i \quad (3)$$

$$h\equiv \bar{c}_p T-c_p^o T_o+\sum m_m H_m = h_t+\sum m_m H_m \quad (4)$$

COMBUSTION MODELS

The laminar-and-turbulent characteristic-time combustion model of Magnussen based on the eddy break-up concept is adopted. This 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. As per this model, the combustion rate (R_f) is described as:

$$R_f=\frac{\rho\varepsilon}{k}A_{ebu}\min\left(m_F,\frac{m_O}{s_O},B_{ebu}\frac{m_P}{s_P}\right)kg/m^3s \quad (5)$$

Where:

$$s_O\equiv n_O M_O/n_F M_F$$
$$s_P\equiv n_P M_P/n_F M_F$$

A_{ebu}, B_{ebu} are dimension less empirical coefficients and k/ϵ is the turbulent time scale. The Shell auto-ignition model is employed in the present analysis with the following assumptions RH hydrocarbon fuel of nominal composition C_nH2_{n+2}

R* = Radical formed from the fuel (R* denotes radical)
B = Branching agent
Q = Intermediate species
P = Products consisting of CO, CO_2 and H_2O

The chemical model consists of the following set of equations:

Primary initiation:

$$RH+O_2\rightarrow 2R^* \quad (6)$$

Main propagation:

$$R^*\rightarrow R^*+P \quad (7)$$

$$R^*\rightarrow R^*+B \quad (8)$$

$$R^*\rightarrow R^*+Q \quad (9)$$

$$R^*+Q\rightarrow R^*+B \quad (10)$$

Branching:

$$B \rightarrow 2R^* \tag{11}$$

Linear termination:

$$R^* \rightarrow \text{inert products}$$

Quadratic termination:

$$2R^* \rightarrow \text{inert products}$$

The non-reactive species (inert products) in the termination equations are assumed to be equivalent to nitrogen. The rate coefficients of the above reactions take the Arrhenius form.

$$k = Ae^{(-E/RT)} \tag{12}$$

The modelling of fuel injection processes is an essential part of DI diesel engine simulation. The existing fully coupled stochastic Lagrangian-Eulerian approach used in STAR-CD has been enhanced to avoid the necessity to empirically tune coefficients or other inputs of the spray model. The velocity of the liquid fuel (injection velocity) as it exits the nozzle and enters the combustion chamber, is one of the most important parameters in a spray calculation. It strongly influences the atomisation (break-up processes), the spray penetration, the inter-phase transfer processes and the droplet-droplet interaction. In the present analysis, the effective nozzle model simulates the nozzle. This model determines the injection velocity based on the injector parameters like nozzle hole cross-sectional area, nozzle hole diameter, roughness and the discharge coefficient:

$$C_d = \frac{\dot{Q}}{A_o\sqrt{\frac{2}{\rho_d}\Delta p}} \tag{13}$$

where, $\dot{Q}$ is the volumetric flow rate through the injector, ρ_d is the fuel density and Δp is pressure drop across the nozzle.

No_x model: In the present analysis Zeldovich mechanism was used for NOx prediction.

Three different mechanisms have been identified for the formation of nitric oxide during the combustion of hydrocarbons, namely Thermal NOx, Prompt NOx and Fuel NOx. Among these, estimation of thermal NOx important in diesel engine combustion. Thermal NOx is strongly temperature dependent. It is produced by the reaction of atmospheric nitrogen with oxygen at elevated temperatures. For thermal nitric oxide, the principal reactions are generally recognized to be those proposed by the following three extended Zeldovich mechanisms.

$$N_2 + O \underset{K_{-1}}{\overset{K_1}{\Leftrightarrow}} NO + N \tag{14}$$

$$N + O_2 \underset{K_{-2}}{\overset{K2}{\Leftrightarrow}} NO + O \tag{15}$$

$$N + OH \underset{K_{-3}}{\overset{K_{31}}{\Leftrightarrow}} NO + H \tag{16}$$

PRE PROCESSING AND GRID GENERATION OF THE GEOMETRY

The preprocessing mainly involves in the creation of basic 3D model, grid generation and fixing of the boundary conditions. The creation of the geometry is done in GAMBIT, the mesh generation package of FLUENT. The partially generated grid in GAMBIT is exported to STAR-CD for completing the mesh. In this analysis a complete hexahedral structured mesh was created for the ports and cylinder. For performing transient flow simulations it is necessary to use moving grids that incorporate the piston and valves motion. The CFD package STAR-CD possesses the above capabilities required for the simulation of transient flow cases. Cell layer addition and removal are controlled by event and moving mesh commands. Before starting the flow simulation, the correct valve and piston movement for entire cycle at any crank angle position is verified from the mesh preview (Fig. 1). The entire computational work was done at IC engine Simulation laboratory. Pentium IV processors computer was used for the computational work and it took 6 months to complete the entire work.

The engine studied in this work is a direct-injection Diesel engine. Table 1 shows the configuration of the engine . The inlet valve axis is offset from the cylinder axis by 18.5 mm in the x direction and 2.0 mm in the y direction.

Table 1: Engine specification

Specification	Values
No. of cylinders	One
Bore	79.5 mm
Stroke	95.5 mm
Connecting rod length	144 mm
Compression ratio	19.5:1
Engine speed	1000 rpm
Injection pressure	270 bar
Start of injection	12 deg BTDC
Injection duration	20 deg CA
No. of holes	3

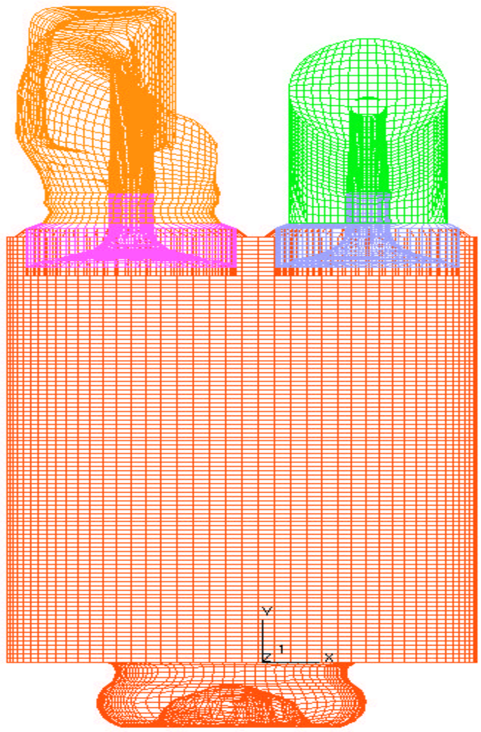

Fig. 1: Computational domain of the engine

RESULTS AND DISCUSSION

Effect of early injection on peak pressure and temperature: Simulations were carried out to study the effect of early injection on combustion performance and emission characteristic of the engine operating at a speed of 1000 rpm. A single step profile was considered for study. The start of injection (SOI) is 12°, 16° and 20° bTDC. Figure 2 below shows the effect of early injection on average cylinder pressure with respect to crank angle.

Figure 3 shows the variation of average cylinder temperature. From the graph peak pressure and temperature shows an increase with advancing the injection timing. The high peak pressure 13.8% (115 bar) and 5.9% (109 bar) and Peak temperature 2.3% (1197 K) and 1.3% (1185K) obtained when the fuel is injected at 20° bTDC and 16° bTDC respectively. Since the fuel is injected at early stage of compression stroke, at this point in-cylinder temperature and pressure is very less. It increases the ignition delay or more specifically physical delay. Due to long ignition delay large parts of fuel made premixed mixture and give more aggressive combustion (Zhu *et al*., 2003). It results in increase in peak pressure and temperature. It is also observed that the occurrence of peak pressure advances with early injection.

Table 2: Comparison of ignition delay, peak heat release rate and combustion duration for injection timing

Start of Injection (deg bTDC)	Ignition delay (deg)	Peak heat release rate (J deg^{-1})	Combustion duration (deg)
20	10.3	158.5	49.4
16	8.2	125.2	50.2
12	6.6	97.1	55.5

Effect of early injection on heat release rate: Figure 4 shows the variation in cylinder averaged instantaneous heat release rate. The ignition delay, combustion duration and peak heat release rate obtained for different injection timing are summarised in Table 2. The early injection timing 20° bTDc and 16° bTDC shows higher peak heat release rate a compared to base case of 12°. bTDC. Early injection timing leads to longer ignition delay which results in accumulation of large amount of evaporated fuel before the start of combustion. Longer ignition delay is due to lower values of pressure and temperature inside the cylinder during the initial period of fuel injection at advanced injection timings. The longer ignition delay leads to rapid burning rate and the pressure and temperature inside the cylinder rises suddenly. Hence, most of the fuel burns in premixed mode causing higher peak heat release rate and shorter combustion duration. Whereas the baseline case 12° bTDC the ignition delay is short causing accumulation of relatively less amount of evaporated fuel. Shorter ignition delay is due to pressure and temperature inside the cylinder during the initial period of fuel injection being high. The shorter ignition delay shortens the mixing time which leads to slow burning rate and slow rise in pressure and temperature. Hence, most of the fuel burns in diffusion mode rather than premixed mode resulting in lower peak heat release rate, longer combustion duration.

Effect of early injection on NO_X and soot emission: Figure 5 shows the cylinder averaged NO_x emission at various crank angles. From the graph it is observed that the NO_x emission increases with early injection timing .In the early injection more quantity of fuel burns instantly in premixed combustion period. It gives very high combustion temperature and hence NO_x formation rate increases.

Figure 6 shows the cylinder averaged soot emission at various crank angle. From the graph it is observed that the soot emission decreases with early injection timing. In the early injection combustion temperature increases which increases the oxidation reaction and hence Soot formation decreases.

Effect of exhaust gas recirculation on peak pressure and temperature: The Start of Injection (SOI) is 20°

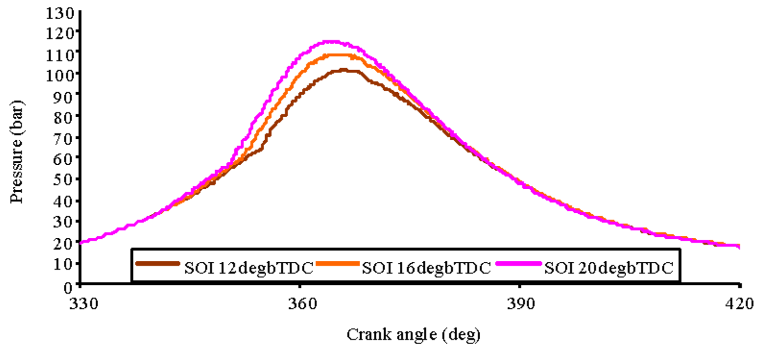

Fig. 2: Cylinder average pressure

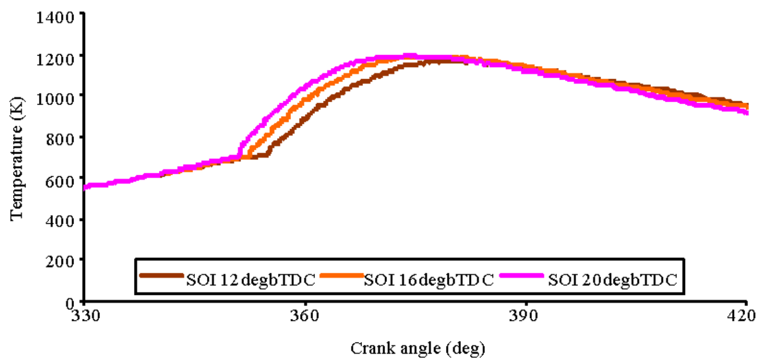

Fig. 3: Cylinder average temperature

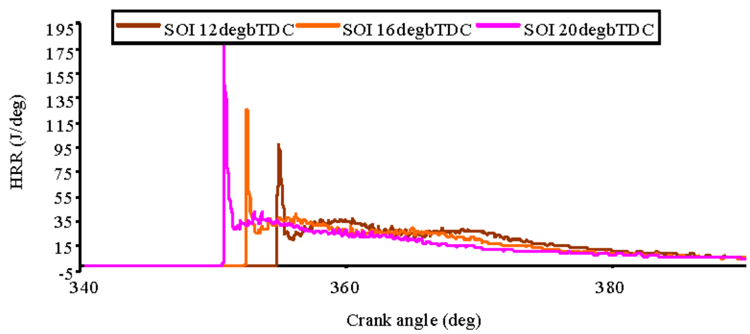

Fig. 4: Average heat release rate

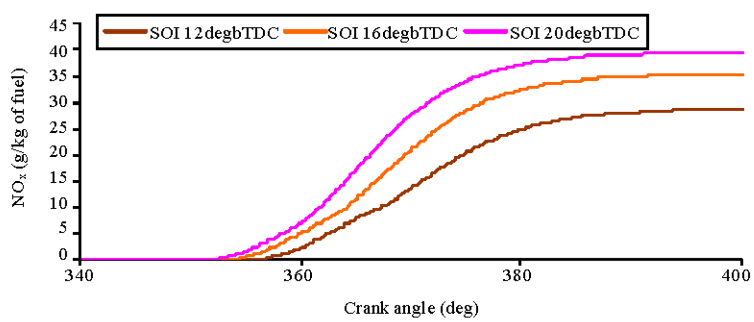

Fig. 5: Cylinder average NO_x emission

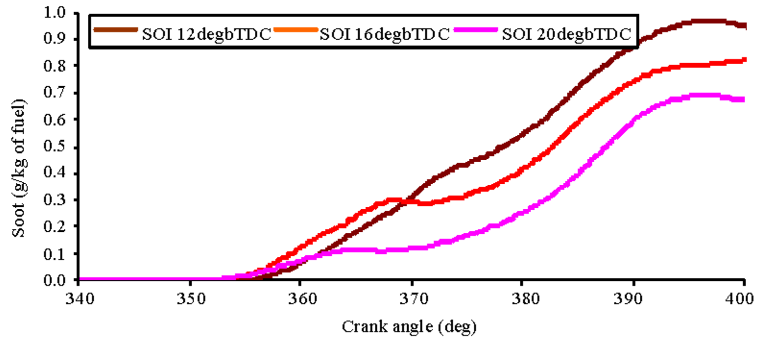

Fig. 6: Cylinder average soot emission

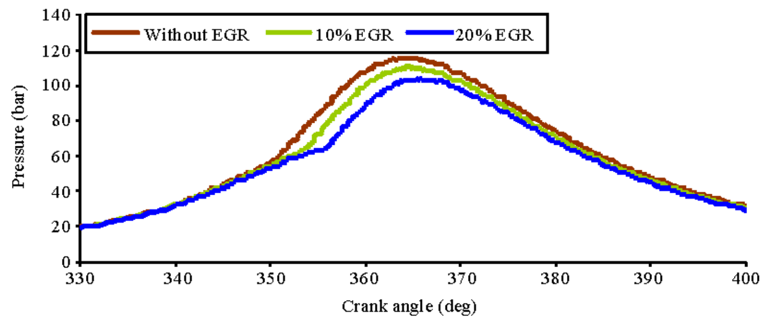

Fig. 7: Cylinder average pressure

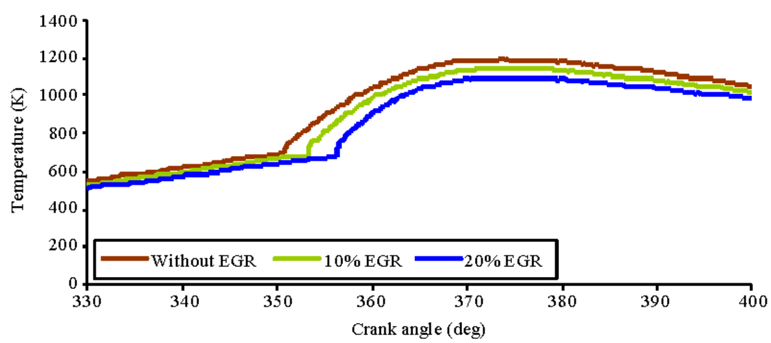

Fig. 8: Cylinder average temperature

bTDC, 10% EGR, 20% EGR and a single step profile is considered for study. Figure 7 shows the effect of EGR on cylinder average combustion pressure.

Figure 8 presents the variation of cylinder temperature with EGR.

From the Fig. 8 it is observe that peak pressure and temperature decreases with increasing the EGR fraction. The peak pressure is decreased by 5.5% (109 bar)and 13.86 % (101 bar)with 10% EGR and 20% EGR respectively. The peak temperature is decreased by 4.5%(1145 K) and 9.4% (1094 K) with 10% EGR and 20% EGR respectively. This is due to the fact that, exhaust gas has a higher specific heat than air and amount of oxygen available for combustion is comparatively less in case of 10 and 20% EGR. Also, observed that start of combustion was retarded with increasing EGR fraction, due to diluting the air.

Effect of exhaust gas recirculation on NO_x and soot emission: Figure 9 and 10 show the cylinder averaged NO_x and soot emission at various crank angle. From the graph it is observed that the NO_x emission decrease with

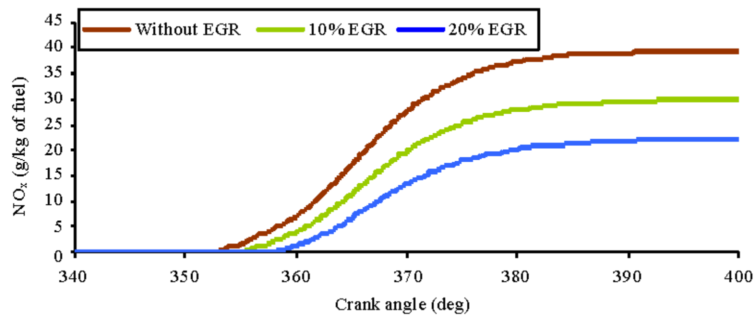

Fig. 9: Cylinder average NO_x emission

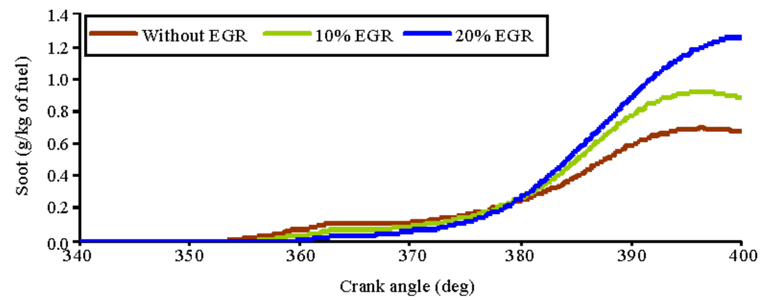

Fig. 10: Cylinder average soot emission

increasing EGR fraction .The presence of EGR will act as heat sink and also diluting the charge, there by reducing the combustion temperature which will reduce the formation of NO_x

From the graph it can be observed that the soot emission increases with increase in EGR fraction. The presence of EGR reduces combustion temperature which decreases the oxidation reaction and hence soot formation increases.

CONCLUSION

A computational investigation of the combustion process was performed using STAR-CD. The following conclusions drawn based on this analysis,

Peak pressure increases with early injection. Peak pressure is 109 bar and 115 bar when the fuel is injected at 16° bTDC and 20° bTDC respectively. Occurrence of peak pressure also advances with early injection. Since the combustion duration decreases with early injection, it is an indication of HCCI mode of operation.

No_x emission increases with early injection (without EGR) the value is 22.5 and 36.8% higher than the conventional combustion at 16° bTDC and 20° bTDC respectively.

Soot emission decreases with early injection and the value obtained is 19.7 and 44.7% lesser than the conventional combustion when the fuel is injected at 16° bTDC and 20° bTDC, respectively.

Due to supplementation of EGR peak pressure decreases compared with early injection without EGR. Cooled EGR works as heat sink and drops the temperature avail inside the combustion chamber. It results in drop in combustion pressure. EGR also retard the occurrence of peak pressure.

Due to the presence of EGR, NOx emission decreases. From the simulation it is also found that the 10% EGR fraction simultaneously reduce the NOx and soot emission. The engine peak pressure also improved by 7% compare with base line 12° bTDC injection timing.

From the above simulation, combination of 20° bTDC injection timing and 10% EGR fraction improve the engine performance and simultaneously reduce the NOx and soot emission.

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