INCORPORATION OF EXHAUST GAS RECIRCULATION AND SPLIT INJECTION FOR REDUCTION OF NO_x AND SOOT EMISSIONS IN DIRECT INJECTION DIESEL ENGINES

by

Hassan KHATAMNEZHAD [a*], ***Shahram KHALIL ARYA*** [a], ***Samad JAFARMADAR*** [a], ***and Arash NEMATI*** [b]

[a] Department of Mechanical Engineering, Urmia University, Urmia, Iran
[b] Department of Mechanical Engineering, Miyaneh Branch, Islamic Azad University, Miyaneh, Iran

Original scientific paper
UDC: 621.43.05:614.72:662.75
DOI: 10.2298/TSCI100317019K

In this paper, reduced temperature combustion has been investigated at high load condition of a direct injection Diesel engine. A 3-D computational fluid dynamics model for flow field, spray, air-fuel mixture formation, combustion, and emissions formation processes have been used to carry out the computations. The combined effect of exhaust gas recirculation temperature and exhaust gas recirculation rate was analyzed to choose with consideration of engine performance. Then, the influence of different injection rates and split injection was explored at a reduced temperature combustion condition by the use of exhaust gas recirculation. The results represent sensitiveness of various injection schemes on the combustion process and emission formation at reduced temperature condition in direct injection Diesel engines.

Key words: *Diesel engine, combustion, emission, exhaust gas recirculation, split injection*

Introduction

Direct injection (DI) Diesel engines have proved to be an efficient option in heavy-duty applications like transportation or power generation. However, due to the natural conditions of high pressure and temperature in the combustion process, Diesel engines emit considerable amounts of pollutants, especially nitrogen oxides (NO_x) and soot [1]. International regulations ratified in recent years have imposed more stringent limits on pollutant emissions in internal combustion engines. To comply with these regulations with the common rail injection system which is widely used in recently developed engines, several new fuel injection strategies on conventional diesel combustion have been investigated in DI Diesel engines.

Variable injection rates are a possible way to meet increasingly restrictive emissions requirements for DI diesel engines. Jafarmadar *et al*. [2] investigated the effect of injection mode parameters on combustion and emissions of a DI diesel engine via FIRE computational

* Corresponding author; e-mail: khatamnezhad@yahoo.com

fluid dynamics (CFD) code. The results indicated that the use of different injection rate curves affects the combustion related properties and can be used as an alternative way to control combustion and emission in the Diesel engines. The ramp injection mode has shown to reduce the NO_x mass fraction due to decreased in-cylinder temperature, also the soot mass fraction decreases considerably at this case. Sayint *et al*. [3] explored the effects of fuel injection timing on the exhaust emission characteristics of a DI Diesel engine fueled with canola oil methyl ester-diesel fuel blends. The results indicated the similar trends for diesel fuel and canola oil methyl ester-diesel blends on the exhaust emissions for different injection timings. The advanced injection timings reduce soot emission and emission boosted for all test conditions.

In order to further reduction of both NO_x and soot emissions, new diesel combustion concepts should be developed in conjunction with suitable injection strategies.

One concept of new diesel combustion is homogenous charge compression ignition (HCCI) based on the simultaneous ignition of a highly diluted premixed air-fuel mixture throughout the combustion chamber [4-6]. Close to homogenous conditions are obtained by very early fuel injection. This concept corresponds to combustion in an area with an equivalence ratio leaner than 1 and a temperature lower than 2200 K. HCCI combustion results in minimal soot and NO_x emissions with only a slight decrease in fuel efficiency.

Partial premixed charge compression ignition (PCCI) is a further possibility for low emission combustion [7, 8]. This concept uses partial premixing of the fuel to reduce the non-premixed part of the combustion. In this concept, low NO_x and soot emissions achieved by injecting fuel early in the compression stroke in order to form a premixed lean mixture over a long mixing period. Also, increase the amount of fuel injected beyond a certain level is resulted in knock that this is limiting the operating range of PCCI combustion.

Recently, high equivalence ratio combustion, namely low temperature combustion (LTC) concept, based on extensive use of high cooled exhaust gas recirculation (EGR) rates has been investigated by Sasaki *et al*. [9] and Akihama *et al*. [10] in a direct injection Diesel engine at very low load condition. They indicated a smoke-less and NO_x-less rich combustion when using EGR above a critical point due to a lower combustion temperature below the minimum soot formation temperature (*i. e*., about 1600 K). Bianchi *et al*. [11] carried out a numerical study to explore low temperature combustion condition in a Diesel engine at medium load condition. They reported that EGR cooling reduces the soot formation but cannot almost eliminate it because portions of rich mixture are at high temperature. This high temperature is due the increase heat release from combustion and the reduced dilution effect in compared to low load engine condition. Yun *et al*. [12] demonstrated an optimization of combustion in the low-temperature diesel combustion regime and high EGR fraction. A genetic algorithm scheme was applied to optimize combustion in a diesel engine in order to reduce NO_x, soot, and brake specific fuel consumption (BSFC), simultaneously. Start of injection (SOI) timing, intake boost pressure level, cooled (EGR) rate, and fuel-injection pressure were used for optimization. Alriksson *et al*. [13] investigated possibilities for extending the range of engine loads using low temperature combustion in conjunction with high levels of EGR to reduce soot and NO_x emissions. Their results indicate low soot emissions could not be achieved with high levels of EGR at 50% engine load and in order to achieve lower soot emission, the charge air pressures should be tuned for higher engine load. Beatrice *et al*. [14] represented the LTC concept is a better candidate to reduce both the soot and NO_x forming conditions because it allows easier auto-ignition control and it can be applied to conventional Diesel engines with minimal design modifications. However, the differences in combustion between this concept and conventional diesel combustion must be investigated to determine their effects on spray combustion characteristics as well as emissions. Wakisaka *et al*. [15] presented a detailed study of the effects of increased boost pressure, high EGR level and high injection pressure on exhaust emissions of a light duty Diesel engine. It was indicated that a combination of increased boost pressure, high EGR level, and high injection pressure reduced and soot emissions, significantly.

Therefore, referring to the literature review, the low temperature combustion concept by using high cold EGR rates is a better candidate to highly reduce of both soot and NO_x emissions at low engine load as well as easier auto-ignition and knock occurrence control in compared to HCCI and PCCI concept. Also, it can be applied to conventional Diesel engines with minimal design modifications. However, the differences in chemistry and combustion between this concept and conventional diesel combustion must be investigated to determine their effects on spray combustion characteristics as well as emissions.

The present study has two major objectives. The first is to investigate the effect of various EGR temperatures and EGR rates on mixture formation by discussion on spray characteristics as well as combustion process and emission formation in a DI Diesel engine at high engine load condition. The cooled EGR rate is changed from low level to level corresponding to reduced combustion temperature. As mentioned above, cooled EGR rate can be made NO_x-less condition although soot emission do not show reduction because more portion of rich mixture at high temperature exists on high load condition. Therefore, it is important to consider the mixture formation because operation at real LTC condition is impossible.

The second objective is to examine the effects of advanced injection timing and injection pressure in reduced temperature combustion condition in order to reduce soot engine-out emission using better air-fuel mixing before the start combustion. The results are obtained by simulation of reduced temperature combustion condition with 3-D CFD procedures based on FIRE code.

Model description

The commercial CFD software FIRE was used to perform the numerical simulation of combustion and emission formation in a Diesel engine. The engine specification and operating condition that is used in this work are shown in tab. 1.

Table 1. Engine specifications

Engine type	Caterpillar 3406 DI diesel engine
Engine speed	1600 rpm
Bore × stroke	137.19 × 165.1 mm
Displacement	2.44 liters
Engine load	75%
Compression ratio	15:1
Injector type	Common-rail, electronic control
Injection pressure	90 MPa
Number of nozzle holes	6
Nozzle hole diameter	0.259 mm

Figure 1 shows the 60° sector computational mesh of combustion chamber in 3-D at top dead centre (TDC). Since a 6-hole nozzle is used, only a 60° sector has been modeled. This takes advantage of the symmetry of the chamber geometric set-up, which significantly reduces computational runtime. Number of cells in the mesh is 19,385 cells at TDC. This fine mesh size will be able to provide good spatial resolution for the distribution of most variables within the combustion chamber. Calcula-

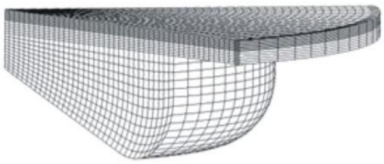

Figure 1. Outline of the computational mesh at TDC

tions are carried out on the closed system from inlent valve closing (IVC) at –147 °CA aTDC to exhaust valve opening (EVO) at 134 °CA aTDC.

At the applied code, the compressible, turbulent, three dimensional transient conservation equations are solved for reacting multi-component gas mixtures with the flow dynamics of an evaporating liquid spray by Amsden *et al*. [16]. The turbulent flows within the combustion chamber are simulated using the random number generation (RNG) k-ε turbulence model which is presented by Han *et al*. [17], modified for variable-density engine flows.

Spray and combustion models

The spray module is based on a statistical method referred to as the discrete droplet method (DDM). This operates by solving ordinary differential equations for the trajectory, momentum, heat and mass transfer of single droplets, each being a member of a group of identical non-interacting droplets termed a parcel. Thus one member of the group represents the behavior of the complete parcel.

The Kelvin-Helmholtz Rayleigh-Taylor (KH-RT) model was selected to represent spray breakup [18]. In this model Kelvin-Helmholtz surface waves and Rayleigh-Taylor disturbances should be in continuous competition of breaking up the droplets.

The Dukowicz model [19] was applied for treating the heat-up and evaporation of the droplets. This model assumes a uniform droplet temperature. In addition, the rate of droplet temperature change is determined by the heat balance, which states that the heat convection from the gas to the droplet either heats up the droplet or supplies heat for vaporization.

The Shell auto-ignition model was used for modeling of the auto-ignition. In this generic mechanism, 6 generic species for hydrocarbon fuel, oxidizer, total radical pool, branching agent, intermediate species and products were involved. In addition the important stages of auto-ignition such as initiation, propagation, branching, and termination were presented by generalized reactions, described in [20].

Combustion process is modeled by eddy break-up model [21]. In the eddy break-up model, the rate of consumption of fuel is specified as a function of local flow properties. The mixing-controlled rate of reaction is expressed in terms of the turbulence time scale k-ε, where k is the turbulent kinetic energy and ε is the rate of dissipation of k. With s as the stoichiometry coefficient and C'_R and C_R as model constants, a transport equation for the mass fraction of fuel is solved, where the:

$$S_{fu} = -\rho \frac{\varepsilon}{k} \min.\left(C_R M_{fu}, C_R \frac{M_{ox}}{s}, C'_R \frac{M_{pr}}{1+s} \right) \tag{1}$$

The first two terms of the “minimum value of” operator determine whether fuel or oxygen is present in limiting quantity, and the third term is a reaction probability which ensures that the flame is not spread in the absence of hot products.

Emissions models

NO_x formation model is derived by systematic reduction of multi-step chemistry, which is based on the partial equilibrium assumption of the considered elementary reactions using the extended Zeldovich mechanism [22] describing the thermal nitrous oxide formation. The reaction mechanism can be expressed in terms of the extended Zeldovich mechanism:

$$N_2 + O \underset{K_2}{\overset{K_1}{\leftrightarrow}} NO + N \tag{2}$$

$$N + O_2 \underset{K_4}{\overset{K_3}{\leftrightarrow}} NO + O \tag{3}$$

$$N + OH \underset{K_6}{\overset{K_5}{\leftrightarrow}} NO + O \tag{4}$$

The assumption of partial equilibrium provides satisfactory results for the formation of NO_x at considerably high temperature.

The overall soot formation rate is modeled as the difference between soot formation and soot oxidation. Soot formation is based on Hiroyasu model and the soot oxidation rate is adopted from Kennedy *et al*. [23]:

$$\frac{dM_s}{dt} = \frac{dM_{sf}}{dt} - \frac{dM_{so}}{dt} \tag{5}$$

where

$$\frac{dM_{sf}}{dt} = k_f . M_{fv}, \quad k_f = A_f P^{0.5} \exp\left(-\frac{E_f}{RT}\right) \tag{6}$$

$$\frac{dM_{so}}{dt} = \alpha A\left(\frac{\varepsilon}{k}\right) M_s, \quad \alpha = \min.\left(1, \frac{M_{O_2}}{M_s \Phi_s + M_f \Phi_f}\right) \tag{7}$$

All above equations are taken in account simultaneously to predict spray distribution and combustion progress in the turbulent flow field, wall impingement and diesel combustion rate using two stage pressure correction algorithms.

Numerical model

The numerical method used in this study is a segregated solution algorithm with a finite volume-based technique. The segregated solution is chosen, due to the advantage over the alternative method of strong coupling between the velocities and pressure. This can help to avoid convergence problems and oscillations in pressure and velocity fields. This technique consists of an integration of the governing equations of mass, momentum, species, energy, and turbulence on the individual cells within the computational domain to construct algebraic equations for each unknown dependent variable. The pressure and velocity are coupled using the SIMPLE (semi-implicit method for pressure linked equations) algorithm which uses a guess-and-correct procedure for the calculation of pressure on the staggered grid arrangement. It is more economical and stable compared to the other algorithms. The upwind scheme is employed for the discretization of the model equations as it is always bounded and provides stability for the pressure correction equation.

Results and discussion

In the first part of this section, the combined effect of EGR temperature and EGR rate to achieve reduced temperature combustion is investigated. In the subsequent sections, parametric study is conducted to explore the influence of the injection rate and split injection on combustion and emissions at this combustion condition.

Model validity

Before using the 3-D CFD model to examine the effect of EGR on combustion process and emissions, it is necessary to validate its predictive ability. For this reason we have used experimental data for the single cylinder test engine mentioned above. Parameters tuned are: spray tip penetration, in-cylinder pressure, NO_x and soot exhaust emissions.

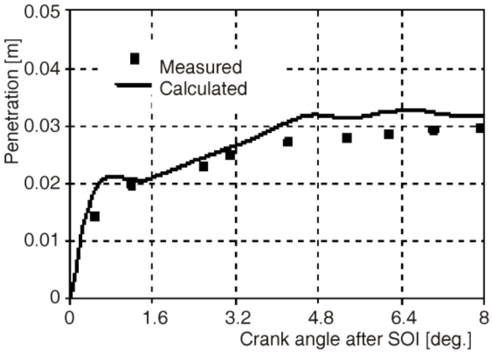

Figure 2. Comparison of calculated and measured spray tip penetration with 8.5 ºCA bTDC and 75% load

Figure 2 shows predicted spray penetration for caterpillar Diesel engine in comparison with experimental data which has been investigated by Ricart *et al*. [24]. Experimental data for spray penetration exist until about 8 °CA after SOI as ignition occurred and the optical measurement system did not contain a narrow pass filter to block out the flame luminosity. There is a good agreement between calculated and measured results and as can be seen, the numerical model tends to over predict the measured spray tip penetration. The accuracy of calculated spray penetration illustrates the ability of this model in prediction of spray and mixture formation.

Figures 3 and 4 indicate the comparison of simulated and experimental in-cylinder pressures against to crank angle for the caterpillar Diesel engine with SOI at 6 ºCA bTDC and 9 ºCA bTDC in 75% engine load condition, respectively. The good agreement of predicted in-cylinder pressure with the experimental data which has been investigated by Chan *et al.* [25] can be observed. It is due to time step and computational grid independency of obtained results. Also, the simulated in-cylinder pressure in compared to experimental value has been illustrated in fig. 3(b) and fig. 3(d) for SOI at 9 ºCA bTDC and 6% EGR rate and 10% EGR rate in 75% engine load condition, respectively.

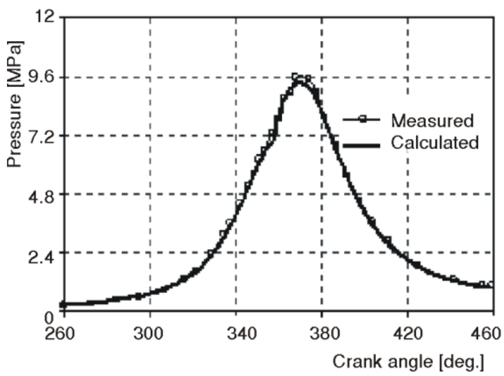

Figure 3. Comparison of calculated and measured in-cylinder pressure with SOI at 6 ºCA bTDC and 0% EGR

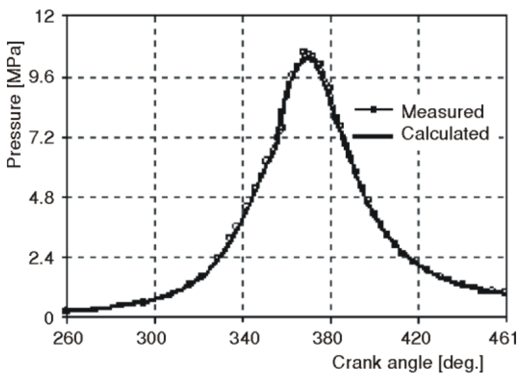

Figure 4. Comparison of calculated and measured in-cylinder pressure with SOI at 9 ºCA bTDC and 0% EGR

Therefore, the CFD and combustion simulation model performed in this study are able to represent the real combustion process inside DI Diesel engine and have the capability to be implemented for further calculation.

Figures 5 to 10 illustrate the total in-cylinder and soot variation with crank angle in compared to experimental values [25] at EVO in different SOI and EGR rates. Also, it can be seen that emission is reduced by retard of SOI as well as the increase of EGR rate. It appears that NO_x formation is highly dependent on temperature. Also, as the inlet combustion air is diluted by exhaust gas recirculation, the overall soot value is increased at EVO. The calculated total in-cylinder and soot values have good agreement to experimental values at EVO that represents the model capability in the assessment of emissions.

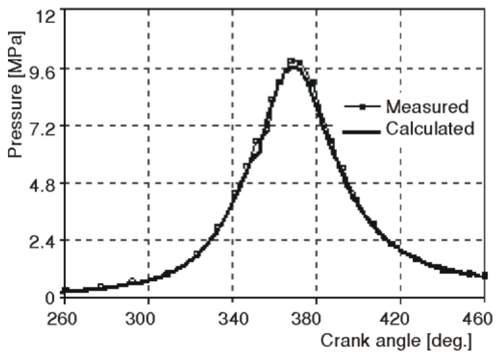

Figure 5. Comparison of calculated and measured in-cylinder pressure with SOI at 9 °CA bTDC and 6% EGR

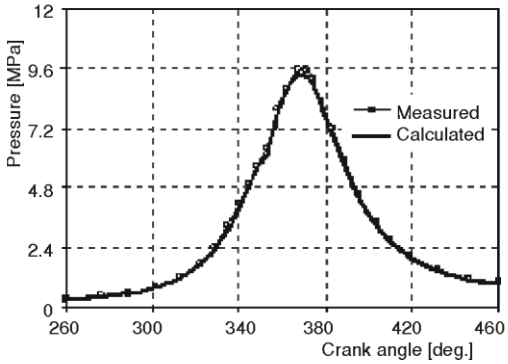

Figure 6. Comparison of calculated and measured in-cylinder pressure with SOI at 9 °CA bTDC and 10% EGR

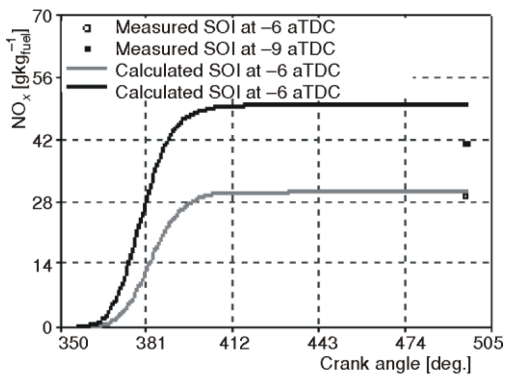

Figure 7. Comparison of calculated and measured NO_x emission value at EVO at 0% EGR

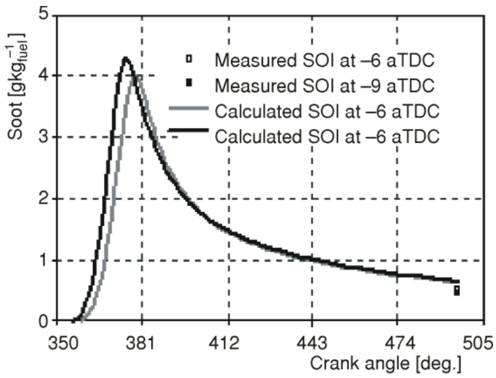

Figure 8. Comparison of calculated and measured soot emission value at EVO at 0% EGR

The good agreement between the measured and computed results for this engine operating condition gives confidence in the model predictions, and suggests that the model may be used to explore new engine concepts about effects of EGR using and injection parameters on the spray characteristics, combustion process and emissions. In this work, single injection case with the SOI timing of 9 °bTDC and with no EGR, which was described in this section, is so-called a baseline case.

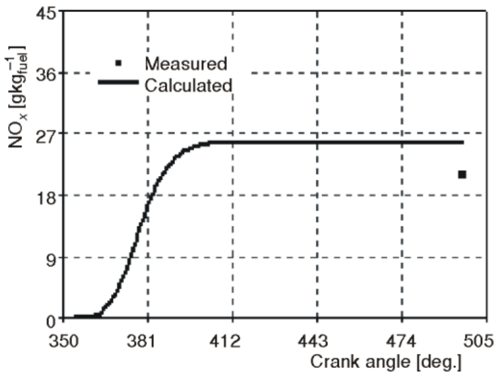

Figure 9. Comparison of calculated and measured NO_x emission value at EVO at 10% EGR

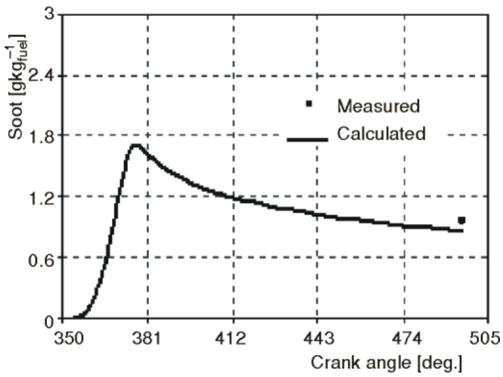

Figure 10. Comparison of calculated and measured soot emission value at EVO at 10% EGR

Effect of EGR temperature

In order to investigate the effect of EGR cooling, a fixed EGR rate of 10% has been considered. Two different cooling temperatures are simulated: (a) hot (388 K) and (b) cold (319 K).

Mean in-cylinder pressure, heat release rate and mean in-cylinder temperature curve against to crank angle have been shown in figs. 11 to 13, respectively. As can be seen from fig. 6, the peak value of premix combustion is reduced by increasing inlet charge temperature in the hot EGR. This is because the increase in inlet charge temperature increases the cylinder gas temperature throughout the combustion cycle. It shortens the ignition delay period which provides less time for the fuel to mix with the oxygen, resulting in the decrease of premixed combustion. When EGR temperature is increased, the peak value of diffusion burning decreased. This is because the reduction in air availability associated with the use of EGR diluents that cause to decrease in the rate of the fuel burnt in the diffusion phase. Results show that, increase in the charge air temperature associated with EGR will proportionally lower the maximum pressure within the cylinder by approximately 7%. The decrease in-cylinder pressure results in a loss of power.

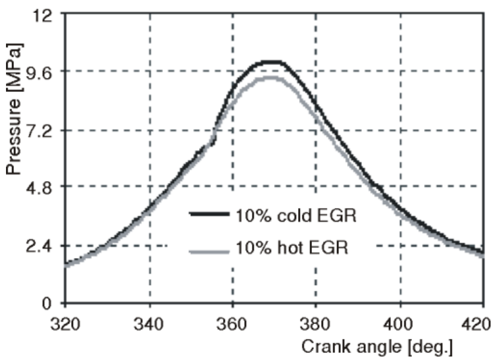

Figure 11. Mean in-cylinder pressure comparing at 10% EGR rate for hot and cold EGR with SOI at 9 °CA bTDC

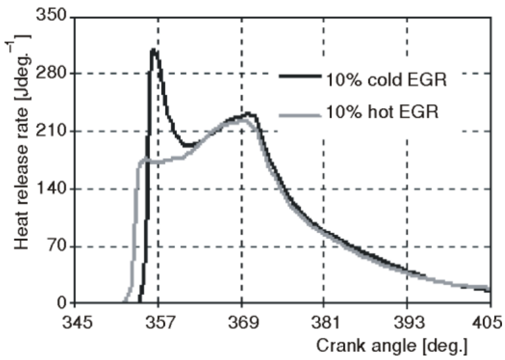

Figure 12. Heat release rate comparing at 10% EGR rate for hot and cold EGR with SOI at 9 °CA bTDC

NO_x, soot, and HC emission results obtained with cold and hot EGR against to crank angle have been illustrated in figs. 14 to 16, respectively. Figure 14 shows that by the use of EGR, NO_x emission is obviously decreased in compared to baseline case (49.8 g/kg$_{fuel}$). It can be seen that the NO_x emissions of the cold EGR is lower than that of the hot EGR case with the same EGR mass fraction. The reduction of emissions with cold EGR is explained to these reasons: a) in the cold EGR case, the flame temperature and the peak cylinder temperatures is decreased due to a low charge temperature at IVC; (b) the mass of EGR and exhaust gas is increased in the engine due to its increased density and more energy is required to heat up these high specific heat product gases that causes a decrease in the in-cylinder temperature. Therefore NO_x emission production is reduced. The peak value for soot emission in the hot EGR cases is higher than that of the cold EGR. This is because fuel pyrolysis in the hot EGR can be enhanced, leading to higher soot production. In addition, lower charge temperature increases the charge air density and oxygen concentration that causes soot oxidation is enhanced. Therefore the overall soot emission is reduced in compared to hot EGR case. Also, the results show that increase in the EGR temperature causes unburned HC to be increased. Increase in EGR temperature associated with decrease in volumetric efficiency. Thus, rate of combustion is reduced and exhaust HC emission is increased.

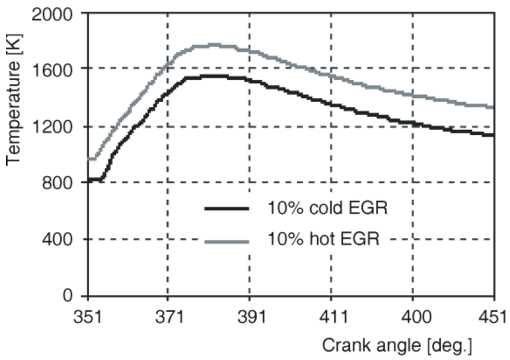

Figure 13. Mean in-cylinder temperature comparing at 10% EGR rate for hot and cold EGR with SOI at 9 ºCA bTDC

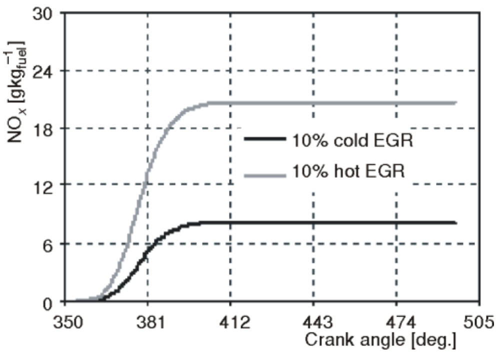

Figure 14. NO_x emission comparing at 10% EGR rate for hot and cold EGR with SOI at 9 ºCA bTDC

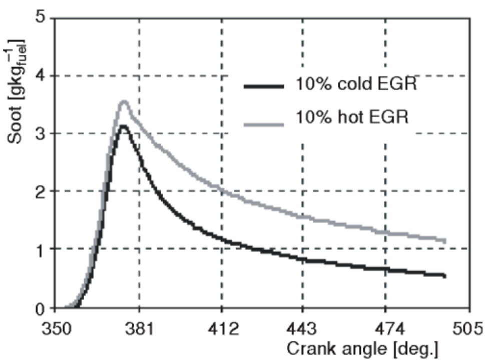

Figure 15. Soot emission comparing at 10% EGR rate for hot and cold EGR with SOI at 9 ºCA bTDC

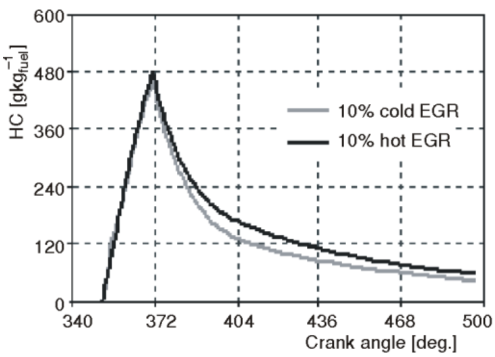

Figure 16. HC emission comparing at 10% EGR rate for hot and cold EGR with SOI at 9 ºCA bTDC

Indicated mean effective pressure (*IMEP*) is a parameter that used for investigating of the engine performance and given by:

$$IMEP = \frac{W_i}{V_1 - V_2}; \quad W_i = \oint P\mathrm{d}V \tag{8}$$

where W_i is the indicated work by piston and V_1 and V_2 are the maximum and minimum cylinder volume, respectively. *ISFC* represents indicated specific fuel consumption which is:

$$ISFC = \frac{\dot{m}_{fuel}}{\dot{W}_i} \tag{9}$$

The obtained gross *IMEP* and *ISFC* in hot and cold EGR condition have been shown in tab. 2. Hot EGR cause to lower *IMEP* in compared to cold EGR condition. As can be seen from tab. 2, the use of hot EGR results in lower *IMEP* in compared to cold EGR. This is because the peak value of mean in-cylinder pressure is decreased at hot EGR and indicated work is decreased, subsequently. Therefore, *ISFC* is decreased by increase in the indicated power.

Table 2. *IMEP* and *ISFC* comparing between 10% hot and cold EGR

	IMEP [bar]	*ISFC* [gkWh^{-1}]
10% cold EGR	13	203
10% hot EGR	12	220

These results indicate that the cold EGR may have been beneficial in term of emission and engine performance. Therefore, cold EGR will be selected for investigation the effect of various EGR rates on combustion process and exhaust emissions in DI Diesel engine in this work.

Effect of EGR rate

In this section at the fixed injection timing and injection pressure and for a given amount of fuel in compared to baseline condition, the effect of various rates of cold EGR is investigated on spray characteristics, combustion process and emissions. EGR temperature at the inlet condition is simulated 319 K equal to inlet temperature in baseline case.

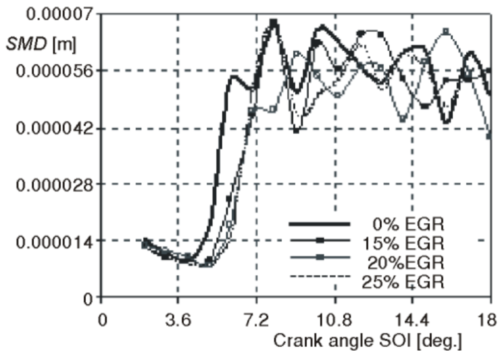

Figure 17. *SMD* distributions for different EGR rates with SOI at 9 ºCA bTDC

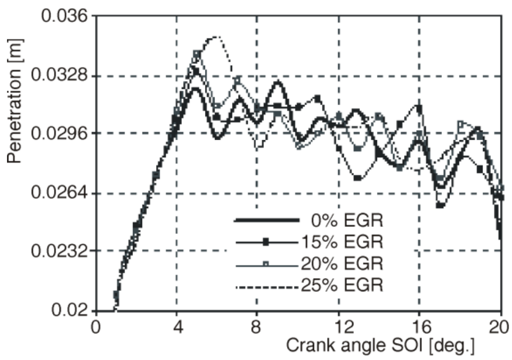

Figure 18. Spray tip penetration for different EGR rates with SOI at 9 ºCA bTDC

SMD distributions in different EGR rates have been shown in fig. 17. *SMD* is one quantity characterizing the average droplet size of a spray and means the average volume of all particles in a cell divided by the average surface area of all particles in the cell. Firstly, *SMD* size is decreased to the lowest value due to break-up of spray in all cases. The increase of *SMD* size after the first decreasing appears for all EGR rates in the combustion chamber. When combustion starts after ignition delay, the smaller droplets were evaporated and extinguished prior to the larger droplets. Therefore larger droplets remained in the fuel jet that led to increase of *SMD* size. Figure 17 indicates the increase of *SMD* size occurs at later crank angle accompanied by the increase of EGR rate. This reveals that the higher EGR rate causes to longer ignition delay.

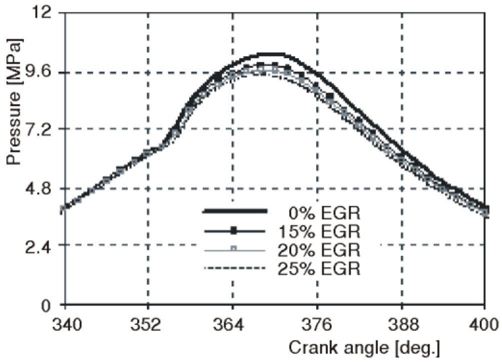

Figure 19. Mean in-cylinder pressure comparing between different EGR rates

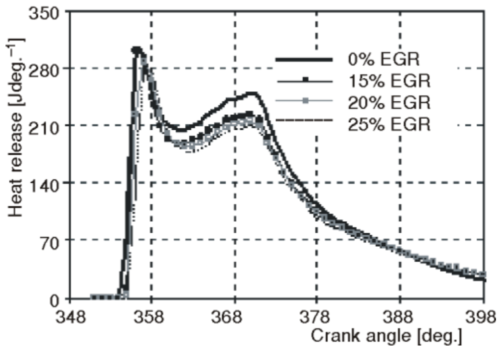

Figure 20. Heat release rate comparing between different EGR rates

The liquid spray penetration length *vs*. crank angle for different EGR rates are shown in fig. 18. The liquid spray penetration profile appeared to follow two different phases; an almost linear phase after start of injection and a rapid transition to a steadier penetration length fluctuated around an average value. Increasing the EGR rate to 25% is clearly seen to increase the rate of penetration. This is the result of reduce of droplets evaporating around the periphery of the penetrating spray in conjunction with the increase of EGR rate and the subsequent decrease of the in-cylinder temperature. Shortly after the break-up period and ignition delay, the liquid penetration profiles fluctuated around a slowly decreasing average value.

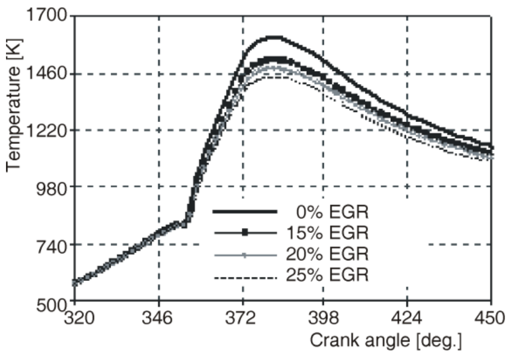

Figure 21. Mean in-cylinder temperature comparing between different EGR rates

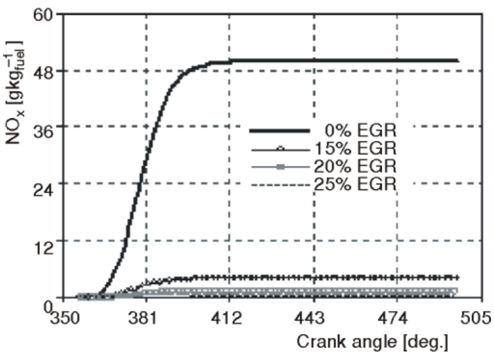

Figure 22. NO_x *vs*. crank angle comparing between different EGR rates

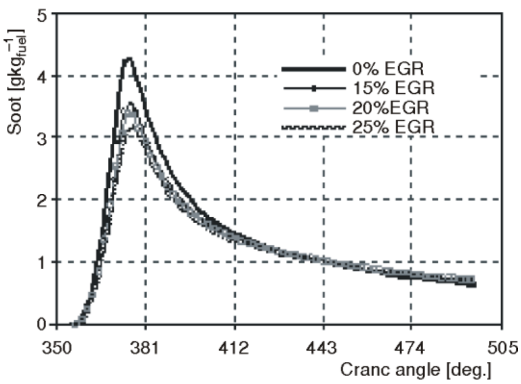

Figure 23. Soot *vs.* crank angle comparing between different EGR rates

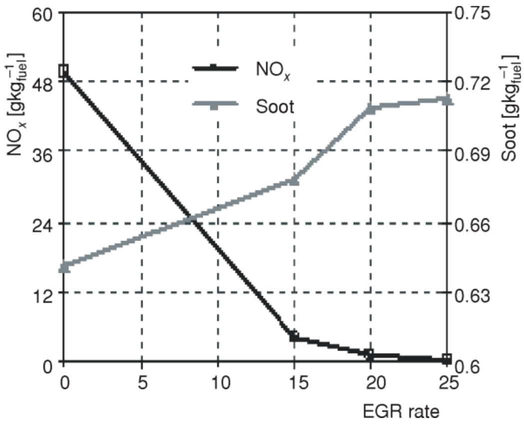

Figure 24. Effect of EGR rate on NO_x and soot emission at EVO for SOI at 9 ºCA bTDC

These fluctuations could be due to the air entrainment in the liquid jet. The momentum exchange between liquid droplets and inert gas leads to breaking away parcels of droplets around the periphery of the leading edge of the spray tip. During the latter phase, it is appear that the penetration profiles of various EGR rates are almost identical.

In figs. 19 and 20, it is given the comparison of cylinder pressure and rate of heat release traces for different EGR rates at 75% engine load condition with SOI at 9 ºCA bTDC. It can be seen that the in-cylinder peak pressure is reduced with increase of EGR rate. This is because the use of high EGR ratio reduces further the availability of oxygen. This lack of oxygen in the cylinder charge reduces the combustion rate leading to retarded combustion and thus to lower peak cylinder pressure values. This is best revealed from the heat release rate curve on various EGR rates.

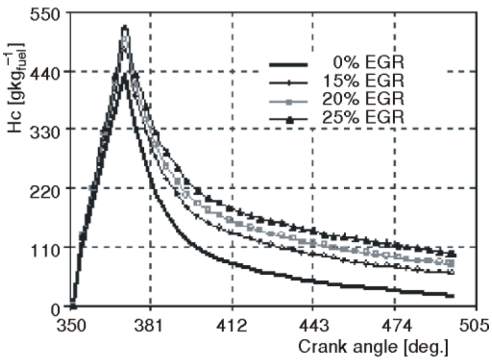

Figure 25. HC *vs.* crank angle comparing between different EGR rates

Figure 21 compares the mean in-cylinder temperature for various EGR rates with baseline case. Operating at 15% EGR, 20% EGR, and 25% EGR rate, the maximum value of mean in-cylinder temperature is reduced down to 1583 K, 1481 K, and 1450 K, respectively, in compared to 1609 K at baseline case. This is due to dilution effect and thermal effect in conjunction with cooled EGR. The introduction of burnt gases into the cylinder replaces a part of the inlet air and causes a reduction of the oxygen concentration. This effect slows down the heat release rate if large amounts of EGR are used and leads to reduction in mean in-cylinder temperature. Moreover, during compression and combustion, the inert burnt gases must be heated up together with the rest of the in-cylinder charge. Because the total heat capacity of the charge is higher with burnt gases due to the higher specific heat capacity values of CO_2 and water vapor, lower end of compression and combustion temperatures are achieved, and heat release rates as well as maximum pressure and temperature are reduced.

The effect of different EGR rates on the overall NO_x and soot exhaust emission at EVO can be seen in fig. 14. Soot is increased with increasing EGR rate and decreasing air-fuel ratio accompanied with intake charge dilution with exhaust gas and reduction of in-cylinder oxygen concentration. When the engine load is increased to high load condition, the higher fuel injected leads to higher in-cylinder temperature in compared to low engine load which is due to greater heat release from combustion. Therefore, because of portions of rich mixture are at intermediate temperature (above 1600 K), real low temperature combustion condition with low NO_x and soot emission cannot be reached at high engine load condition. Also, fig. 14 reveals that the use of high EGR rate leads to an obviously decrease in the NO_x emission rate in compared to base engine condition with no EGR. When EGR level is increased beyond 20% EGR, NO_x emission reduced to nearly zero at 25% EGR rate. NO_x is formed at high temperatures due to the high activation energy of the $O + N_2 \rightarrow NO + N$ reaction in the Zeldovich mechanism. Also the exhaust HC emission in the case of 25% EGR would be 4.69 times of the same quantity in the case of 0% EGR due to decrease in combustion rate as well as lower cylinder temperature in expansion stroke.

As can be observed from tab. 3, IMEP is reduced by the increase of EGR rate. This is because the peak value of in-cylinder pressure is decreased. Thus, indicated work and *IMEP* is reduced at higher EGR rates. Consequently, it is worthwhile to say that ISFC is increased.

Table 3. *IMEP* and *ISFC* comparing between different EGR rates

	IMEP [bar]	*ISFC* [gkWh^{-1}]
0% EGR	13.5	195
15% EGR	12.6	208
20% EGR	12.2	215
25% EGR	11.9	221

As an overall result, for fixed injection timing for a given amount of fuel, it is confirmed that the reduced combustion temperature cause to NO_x emission production is decreased to nearly zero. But it increase quantity of soot particles in the emissions owing to combustion temperature is within the soot formation area and large portions of rich mixture exist at high temperature. Also, engine performance parameter is deteriorated. Therefore air-fuel mixture formation has a critical role at this condition, especially for soot emission reduction. In the following section, in order to reduce soot engine-out emissions, different injection rates were examined for better air-fuel mixing before the start of combustion and improvement of mixture quality at 25% EGR rate which produces nearly zero NO_x emission at reduced temperature combustion.

Effect of SPLIT injection in reduced temperature combustion

The goal of this section was to quantify changes in emissions and combustion caused by varying injection schemes. A set of injection schemes was designed and computed which included three single injections and three split injections. The same amount of fuel is injected in all the cases considered. The conditions of the different cases are shown in tab. 4. Figure 26 shows the nomenclature used to describe the split injection strategies investigated.

In fig. 27 is given the comparison of spray tip penetration for different split and single injection schemes at 25% EGR in compared to baseline case. The lower in-cylinder temperature and pressure at advanced timing to TDC causes to increase of spray penetration in combustion chamber at advanced interjection timings. This leads to the increase of air entertainment into the fuel spray because of the longer spray path before impinging piston bowl.

Also, spray further penetrates in combustion chamber in split injection scheme which improves air-fuel mixing. Spray tip penetration of second pulse is lower than that of first pulse in the split injection. This is because combustion of first pulse increases cylinder temperature which improves evaporation of liquid fuel in the second pulse of split injection and reduces liquid spray penetration.

Table 4. Different injection schemes specification

	SOI (aTDC)	EOI (aTDC)	Injection sheme
Baseline (25% EGR)	–9	12	Single inj.
Case 1	–9	12	Single inj.
Case 2	–17	12	Single inj. 25(8)75
Case 3	–17	12	Single inj. 50(8)50
Case 4	–17	12	Single inj. 75(8)25
Case 5	–17	4	Single inj.
Case 6	–17	12	Single inj.

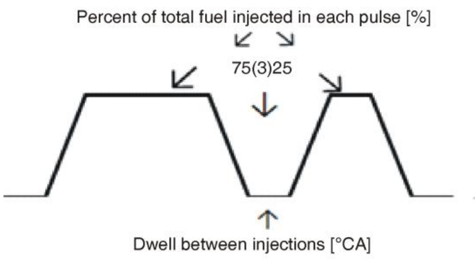

Figure 26. Description of split injection nomenclature (75% of fuels injected in first pulse followed by 3 °CA dwell and 25% of fuel injected in second pulse)

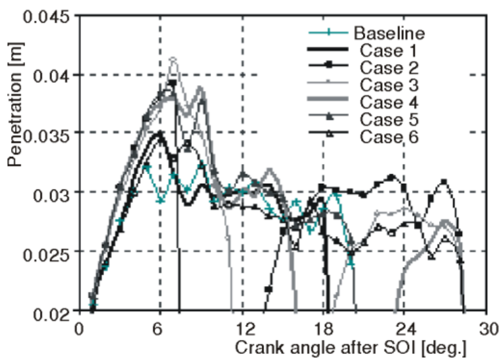

Figure 27. Spray tip penetration for different cases

Mean in-cylinder pressure and heat release rate for different injection rates at 25% EGR rate are illustrated in figs. 28 and 29. As can be seen from fig. 28, in-cylinder pressure gets its maximum increase in case 5. In the case 4 the maximum pressure will be decreased up to 1.7 MPa in compared to case 1. Results show that advancing the injection timing has increased the portion of the premixed combustion and has resulted in high rate of heat release comparing to the case 1 and baseline case. This is due to more ignition delay at advanced injection timing and more fuel injected at this time which leads to more air-fuel mixing and premixed combustion rate. Increasing the premixed portion of combustion also reduces the amount of fuel consumed and prepared for diffusive combustion. Because combustion in the "premixed burn" is approximately stoichiometric, combustion gases will be at their maximum temperature and near peak pressure when the piston is close to TDC. Combustion in split injection cases and single cases are quite different due to their different injection timings, combustion of the fuel in the second pulse is delayed by the injection pause in the former case.

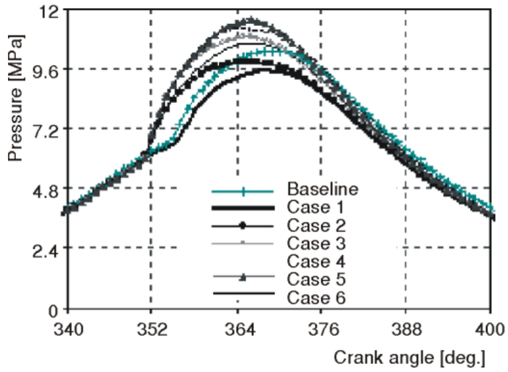

Figure 28. Mean pressure comparing between different cases

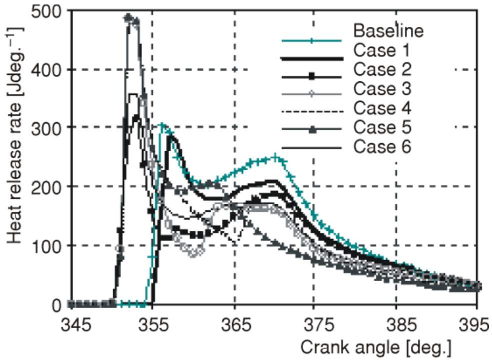

Figure 29. Heat release rate comparing between different cases

Mean in-cylinder temperature at various injection schemes at 25% EGR rate has been shown in fig. 30 compared with the baseline case. As can be seen, advanced injection timing increases the maximum value of in-cylinder temperature. However, high EGR rate causes this maximum value to become less than the baseline case. Case 5 indicates higher cylinder temperature at 25% EGR. It reveals that air-fuel mixing and combustion efficiency of second pulse in hot gases of combustion of main injection (75%) is improved which leads to higher cylinder temperature in expansion stroke.

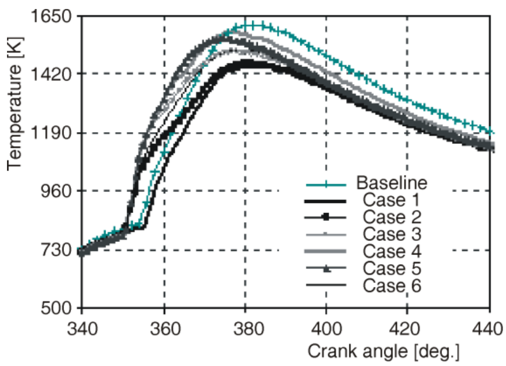

Figure 30. Mean temperature comparing between different cases

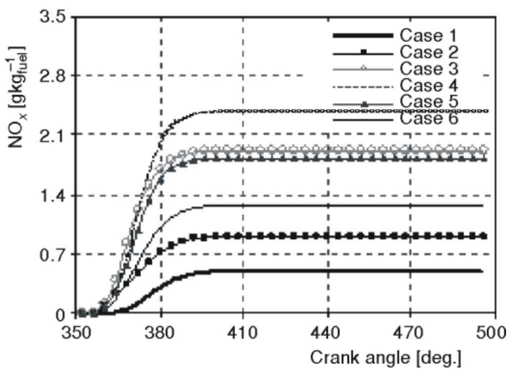

Figure 31. NO_x emission comparing between different cases

Figure 31 indicates the in-cylinder NO_x emission history for the different cases. It is generally believed that NO_x is formed in local high temperature regions present in the cylinder during premixed combustion. With these conditions present, and because the fuel-air charge is not homogeneous, the excess oxygen and nitrogen combines at high rates to form NO_x. Also, NO_x formation rapidly slows down during the later portion of the combustion process, and most of the NO_x which has formed "freezes" as it mixes with cooler cylinder gases and as the

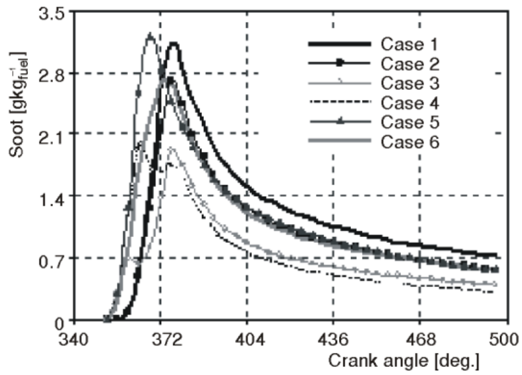

Figure 32. Soot emission comparing between different cases

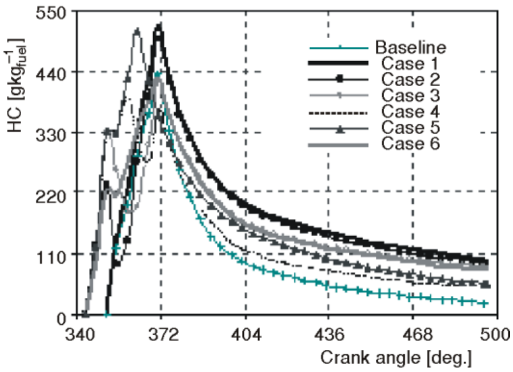

Figure 33. HC emission comparing between different cases

bulk cylinder temperature drops during the expansion stroke. As can be seen, NO_x formation increases at advanced injection timing in compared to case 1. It is because the combustion gases spend more time at high temperatures where NO_x formation rate is enhanced. But it is negligible in compared to NO_x value at EVO in baseline case (49.8 g/kg$_{fuel}$). On the other hand, the injection pause does not affect NO_x formation significantly and combustion of the 25% fuel in the second injection pulse only causes a small increase in NO_x production. However, injection pause delays the major portion of the combustion and reduces the NO_x formation rate in case 2. Figur e 32 presents soot emission history changes with crank angle for the different cases. It is shown that split injections associated with advanced injection timing are effective at reducing soot emission. The quantity of soot particles in the emissions is determined by the balance between the rate of formation and subsequent oxidation. During the second phase of combustion, the "diffusion burn", particulates are formed. Air mixing with the outer edges of the fuel jet sustains the diffusion burn. Fuel in the interior of the spray jet is subjected to high temperatures and pressures but it is starved for oxygen, leading to soot production. Therefore, in-cylinder soot formation rate is increased in the case 6. It's because of higher temperature within the combustion chamber in the case. However, this higher temperature lowers the quantity of soot formed within the cylinder in the rest of expansion stroke. On the other hand, the peak values of in-cylinder soot from the split injections are reduced. Injection pause causes the second pulse of injected fuel enters into a high-temperature region which is left over from the combustion of the first pulse of injection. Soot formation is therefore significantly reduced owing to the injected fuel is rapidly consumed by combustion before a rich soot producing region can accumulate. Also, injection pause promotes the oxygen availability for soot oxidation. Taking the figure in account, it can be seen that longer injection duration in case 6 causes relatively fuel lean and high temperature region in compared to case 1. This leads to lower soot emission at EVO in compared to case 1 and baseline case (0.64 g/kg$_{fuel}$).

Figure 33 shows HC emission comparing between different cases. Baseline case indicates lowest exhaust HC emission at EVO owing to it is operating without EGR associated with high combustion efficiency. Comparing HC emission values at EVO for case 1 to case 6, it is seen that these values increase by advancing injection timing. Case 4 shows lowest HC emission at reduced temperature combustion condition due to better air-fuel mixing and improvement of combustion of second pulse injection. Also, single injection with shorter injection duration near to TDC presents lower HC.

Table 5 studies the effect of different injection schemes on *IMEP* and *ISFC* specifications. As can be seen, case 4 is an optimum case with highest *IMEP* and lowest fuel consumption.

Table 5. *IMEP* and *ISFC* comparing between different cases

	IMEP [bar]	*ISFC* [$gkWh^{-1}$]
Baseline	13.5	195
Case 1	11.9	221
Case 2	12	220
Case 3	12.4	213
Case 4	13.2	200
Case 5	12.8	206
Case 6	12.5	211

In order to further explain for NO_x and soot emissions formation, a general comparison is presented in fig. 14 to verify the interactions of emissions with temperature and local equivalence ratio distributions for crank angles of interest at different cases study. It can be seen that with both oxygen availability (low local equivalence ratio) and high temperature conditions satisfied NO_x formation increases, but high temperature flame leads to a more NO_x formation than the oxygen concentration. As can be seen, the areas with equivalence ratio are close to 1 and the higher temperature than 2200 K are the formation areas. However, high EGR level causes to decreasing in the local high temperature regions and flame temperature due to reduction in oxygen flow rate to the engine and increase of the specific heat of the charge in conjunction with introducing exhaust gas. Therefore, emissions are gradually reduced, especially for the condition of combustion temperature below 2200 K. Also, the areas with high equivalence ratio (higher than 2 to 3) and the temperature approximately between 1600 K and 2000 K in bowl edge and piston surface, were stated the soot formation area. The results indicate in case 4, the extent of the fuel rich regions is reduced since adequate time for mixing has been allowed. Thus, less soot

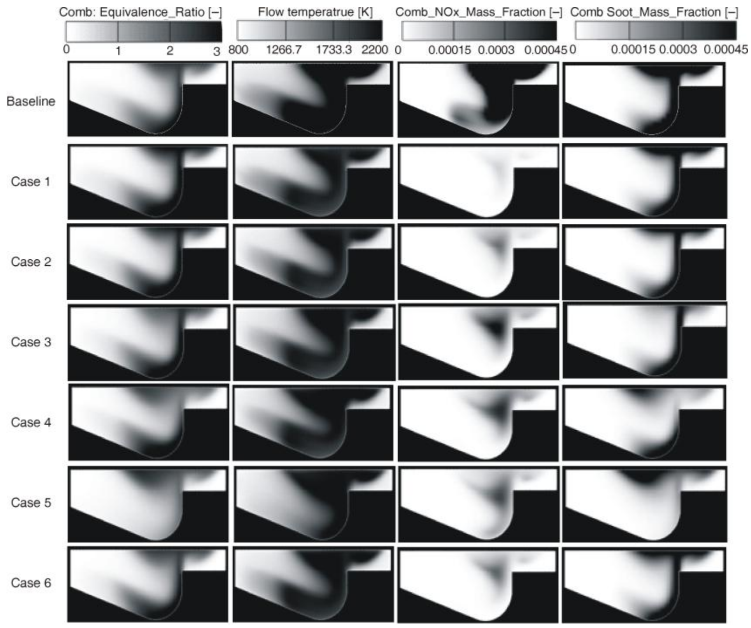

Figure 34. Left to right: contour plots of equivalence ratio (first column), temperature (second column), NO concentration (3rd column), and soot concentration (4th column) at 375 °C

is accumulated in combustion chamber substantially associated with increasing NO_x formation. However, it is negligible in compared to NO_x emission formation in baseline case.

Conclusions

At the present work, after description of reduced temperature combustion condition, the effects of different injection schemes and split injection on spray characteristics, combustion, and emissions have been investigated in reduced temperature combustion regimes by using multi-dimensional CFD code in a DI Diesel engine operating in high engine load condition. Based on the results, 25% cold EGR reduces NO_x emission to nearly zero value at EVO. However, due to soot formation is increased because of portions of rich mixture are at intermediate temperature. Also, *IMEP* is reduced by the increase of EGR rate associated with the decrease of in-cylinder pressure. On the contrary, higher *ISFC* values can be reached with 25% cold EGR in conjunction with the increase of HC formation. In order to improvement of mixture formation at 25% EGR rate to reduce soot emission level, different injection scheme at advanced injection timing explored. It is obvious that split injection with 25% of total fuel injected in the second pulse is able to produce combustion with lower soot level. But, higher premixed combustion causes the increase of NO_x emission which is negligible at 25% EGR level. Also, *IMEP* and *ISFC* are enhanced in this case due to better air-fuel mixing higher cylinder pressure in expansion stroke.

Refrences

[1] Desantes, J. M., *et al*., The Modification of the Fuel Injection Rate in Heavy-Duty Diesel Engines, Part 1: Effects on Engine Performance and Emissions, *J. Applied Thermal Engineering, 24* (2004), 17-18, pp. 2701-2714
[2] Jafarmadar, S., Shafee, S., Barzegar, R., Numerical Investigation of the Effect of Fuel Injection Mode on Spray/Wall Interaction and Emission Formation in a Direct Injection Diesel Engine at Full Load State, *Thermal Science, 14* (2010), 4, pp. 1039-1049
[3] Sayin, C., Gumust, M., Canakci, M., Effect of Fuel Injection Timing on the Emissions of a Direct-Injection (DI) Diesel Engine Fueled with Canola Oil Methyl Ester−Diesel Fuel Blends, *Energy Fuels, 24* (2010), 4, pp. 2675-2682
[4] Ryan, T. W., Callahan, T. J., Homogeneous Charge Compression Ignition of Diesel Fuel, SAE paper 961160, 1996
[5] Dec, J. E., A Computational Study of the Effects of Low Fuel Loading and EGR on Heat Release Rates and Combustion Limits in HCCI Engines, SAE paper 2002-01-1309, 2002
[6] Hosseini, V. Neill, W., Chippior, W., Influence of Engine Speed on HCCI Combustion Characteristics Using Dual-Stage Autoignition Fuel, SAE paper 2009-01-1107, 2009
[7] Lee, C. S., Lee, K. H., Kim, D. S., Experimental and Numerical Study on the Combustion Characteristics of Partially Premixed Charge Compression Ignition Engine with Dual Fuel, *Fuel, 82* (2003), 5, pp. 553-560
[8] Kanda, T., *et al*., PCCI Operation with Early Injection of Conventional Diesel Fuel, SAE paper 2005--01-0378, 2005
[9] Sasaki, S., Ito, T., Iguchi, S., Smoke-Less Rich Combustion by Low Temperature Oxidation in Diesel Engines, 9th *Annual Automobile and Engine Technology Aachen Colloquium*, Aachen, Germany, 2000, p. 767
[10] Akihama, K., *et al*. Mechanism of the Smokeless Rich Diesel Combustion by Reducing Temperature, SAE paper 2001-01-0655, 2001
[11] Bianchi, G. M., *et al*., Numerical Study Towards Smoke-Less and NO_x-Less HSDI Diesel Engine Combustion, SAE paper 2002-01-1115, 2002
[12] Yun, H., Reitz, R. D., Combustion Optimization in the Low-Temperature Diesel Combustion Regime, *Int. J. Engine Res., 6* (2005), 5, pp. 513-524

[13] Alriksson, M., Denbratt, I., Low Temperature Combustion in a Heavy Duty Diesel Engine Using High Levels of EGR, SAE paper 2006-01-0075, 2006
[14] Beatrice, C., *et al*., Critical Aspects on the Control in the Low Temperature Combustion Systems for High Performance DI Diesel Engines, *Oil & Gas Sci.Technology, 62* (2007), 4, pp. 471-482
[15] Wakisaka, Y., *et al*., Emissions Reduction Potential of Extremely High Boost and High EGR Rate for an HSDI Diesel Engine and the Reduction Mechanisms of Exhaust Emissions, SAE paper 2008-01--1189, 2008.
[16] Amsden, A. A., O'Rourke, P. J., Butler, T. D., KIVA ll: A Computer Program for Chemically Reactive Flows with Sprays, Los Alamos National Laboratory, Los Alamos, N. Mex., USA, LA-11560-MS, 1989
[17] Han, Z., Reitz, R. D., Turbulence Modelling of Internal Combustion Engine Using RNG k-ε Models, *Combustion Science Technology, 106* (1995), 4-6, pp. 267-295
[18] Beale, J. C., Reitz, R. D. Modeling Spray Atomization with the Kelvin-Helmholtz/Rayleigh-Taylor Hybrid Model, *Atomization and Sprays, 9* (1999), 6, pp. 623-650
[19] Dukowicz, J. K. Quasi-Steady Droplet Phase Change in the Presence of Convection, Informal Report Los Alamos Scientific Laboratory, Los Alamos, N. Mex., USA, LA-7997-MS, 1979
[20] Halstead, M., Kirsch, L., Quinn, C., The Auto Ignition of Hydrocarbon Fueled at High Temperatures and Pressures-Fitting of a Mathematical Model, *Combustion Flame, 30* (1997), pp. 45-60
[21] Magnussen, B. F., Hjertager, B. H., On Mathematcal Modeling of Turbulent Combustion with Special Emphasis on Soot Formation and Combustion, *Proceedings*, 16th Symposium on Combustion, Cambridge, Mass., USA, 1976, pp. 719-729
[22] Zeldovich, Y. B., Sadovnikov, P. Y., Frank Kamenetskii, D. A., Oxidation of Nitrogen in Combustion, (translation by M. Shelef), Academy of Sciences of USSR, Institute of Chemical Physics, Moscow-Leningrad, USSR, 1947
[23] Kong, S. C., Sun, Y., Reitz, R. D., Modeling Diesel Spray Flame Lift-Off, Sooting Tendency and NO_x Emissions Using Detailed Chemistry with Phenomenological Soot Model, *J. Eng. Gas Turbines Power*, *129* (2007), pp. 245-251
[24] Ricart, L. M., *et al*., In-Cylinder Measurement and Modeling of Liquid Fuel Spray Penetration in a Heavy-Duty Diesel Engine, SAE paper 971591, 1997
[25] Chan, M., Das, S., Reitz, R. D., Modeling Multiple Injection and EGR Effects on Diesel Engine Emissions, SAE paper 972864, 1997

Paper submitted: March 17, 2010
Paper revised: June 17, 2011
Paper accepted: July 25, 2011