DECREASING THE EMISSIONS OF A PARTIALLY PREMIXED GASOLINE FUELED COMPRESSION IGNITION ENGINE BY MEANS OF INJECTION CHARACTERISTICS AND EGR

by

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This paper is presented in order to elucidate some numerical investigations related to a partially premixed gasoline fuelled engine by means of three dimensional CFD code. Comparing with the diesel fuel, gasoline has lower soot emission because of its higher ignition delay. The application of double injection strategy reduces the maximum heat release rate and leads to the reduction of NO_x emission. For validation of the model, the results for the mean in-cylinder pressure, H.R.R., NO_x and soot emissions are compared with the corresponding experimental data and show good levels of agreement. The effects of injection characteristics such as, injection duration, spray angle, nozzle hole diameter, injected fuel temperature and EGR rate on combustion process and emission formation are investigated yielding the determination of the optimal point thereafter. The results indicated that optimization of injection characteristics leads to simultaneous reduction of NO_x and soot emissions with negligible change in IMEP.

Key words: *CI engine, Gasoline fuel, Emission reduction, Combustion, Injection characteristics, EGR*

1. Introduction

The conventional compression ignition (CI) engines have high thermal efficiency and produce lower amount of nitrogen oxides (NO_x) and carbon monoxide (CO) comparing to the Spark Ignition (SI) engines. These advantages make the CI engines as appropriate power generator in automotive industry [1].

Lots of new methods have been researched and performed for the reduction of emission problems. One of the recent methods in engine emissions reduction is utilization of the low temperature combustion (LTC) method [2-4]. Different types of low temperature combustion have been explored, such as homogeneous charge compression ignition (HCCI), premixed charge compression ignition (PCCI), high equivalence ratio combustion based on extensive use of cold EGR rates and partially premixed combustion (PPC) engines.

Homogenous charge compression ignition (HCCI) method is based on an instantaneous ignition of a highly diluted premixed air-fuel mixture through the combustion chamber [5-7]. Such combustion reduces soot and NO_x emissions due to the lower local equivalence ratio and lower combustion temperature respectively [7]. However, the control of combustion in these engines is very difficult and some serious problems occur at low engine speeds.

Premixed Charge Compression Ignition (PCCI) method is more proper method in low emission combustion procedure [8, 9]. This method applies very advanced injection timing and produces lower amounts of NO_x and soot emissions because of providing a better vaporized air-fuel mixture. The mixture conditions approaches to the homogenous charge at low temperature combustion condition. At this method, the control of combustion process and reduction of pollutant emissions can be improved, but increase of the fuel consumption results in knock occurrence, so the operation ranges of the PCCI engines is limited.

Application of the cold EGR is another way to achieve the low temperature combustion condition [9]. It is a very effective procedure in NO_x reduction by decreasing the oxygen concentration in the intake system and chemical effects of added CO_2 and H_2O [10]. Also, at part load conditions, the low combustion temperature in the fuel-rich zones (wherein normally soot produces) contributes to greater suppression of soot formation [11]. But higher engine loads leads to higher soot formation. In these loads portions of rich mixtures are at high temperatures due to the increase heat release from combustion of higher fuel amount.

The PPC strategy is able to combine low smoke and NO_x emissions while having a combustion controllability that is higher than the Homogenous Charge Compression Ignition (HCCI). The use of an early small injection (pilot injection) with a large amount injection (main injection) at later crank angle in PPC engines is able to reduce the maximum value of heat release by spreading out the heat release through split injection [12]. In addition, reduction of heat release rate in this strategy is suitable for higher engine load conditions.

Fuels with lower cetane number (i.e., gasoline) have higher ignition delay and stability against auto-ignition. Because of a better air-fuel mixing before the start of ignition, soot emission reduces and also NO_x formation decreases due to the retarded combustion phase at lower temperature conditions at optimized injection timing.

Several studies confirmed a possibility of reaching low soot and NO_x emissions using a fuel with low cetane number. The effect of different cetane numbers (CN) to retard the first ignition timing on multiple stage diesel combustion has been investigated by Hashizume et al. [13] via numerical modeling and experiments. They demonstrated that in this kind of combustion, first stage ignition is using a fuel with 19 CN in compared to fuel with 62 CN. Also the fuel consumption improved for the lower cetane number fuel due to the degree of constant volume combustion at first stage combustion is increased largely. Shimazaki et al. [14] have shown that a fuel with low cetane number (CN=19) accompanied with a narrow injection angle and shallow dish combustion chamber, improve fuel-air mixing and enable low soot and NOx combustion at higher ignition delay. Kalghatgi et al. [15, 16] investigated effect of fuel auto-ignition quality on engine ignition timings and emissions experimentally for four different fuels with different CN and volatility, including conventional gasoline. Their results indicate that there is significantly higher soot with diesel compared to the gasoline fuel due to lower ignition delay at the same condition. Also for a given IMEP, PPC strategy reduces the maximum heat release rate and enables heat release to occur with low cyclic variation compared to a single injection strategy. In addition, the engine could be run at high loads using gasoline fuel with injection timing near TDC due to much larger ignition delay with lower soot and NO_x emissions compared to diesel fuel. More recently, an experimental study of partially premixed combustion with gasoline fuel was performed using a heavy duty comparison ignition engine by Hanson et al. [17]. They use pilot and main injections in their experiment.

At this work, the effect of injection characteristics such as injection duration, spray angle, nozzle hole diameter, injected fuel temperature and EGR rate have been investigated on combustion and emissions in a partially premixed gasoline-fuelled compression ignition engine.

2. Model Description

At the present model, a compressible, turbulent and three dimensional transient conservation equations are solved utilizing the AVL Fire v8.31 CFD code. The turbulent flows within the combustion chamber are simulated using the RNG k-ε turbulence model which is presented by Han and Reitz [18], modified for variable-density engine flows.

The Kelvin-Helmholtz, Rayleigh-Taylor (KH-RT) model has been selected to represent the spray breakup [19]. In this model Kelvin-Helmholtz (KH) surface waves and Rayleigh-Taylor (RT) disturbances should be in continuous competition of breaking up the droplets. The utilized spray-wall interaction model in the simulations is based on the spray-wall impingement model which represented by O'Rourke and Amsden [20].

The Dukowicz model [21] is applied for treating the heat-up and evaporation of the fuel droplets. This model assumes a uniform droplet temperature. In addition, the rate of droplet temperature change is determined with the heat balance, which means that the heat convection from the gas to the droplet heats it up or supplies heat for vaporization.

Combustion process is modeled by Eddy Breakup model [22]. This model assumes that in premixed turbulent flames, the reactants (fuel and oxygen) are contained in the same eddies and are separated from eddies containing hot combustion products. The rate of dissipation of these eddies determines the rate of combustion according to:

$$\overline{\rho \dot{r}_{fu}} = \frac{C_{fu}}{\tau_R}\bar{\rho}\min\left(\bar{y}_{fu}, \frac{\bar{y}_{ox}}{S}, \frac{C_{pr}.\bar{y}_{pr}}{1+S}\right) \quad (1)$$

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. Above equation includes three constant coefficients (C_{fu}, τ_R, C_{pr}) and C_{fu} varies from 3 to 25 in compression ignition engines. An optimum value was selected according to experimental data.

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 [23] describing the thermal nitrous oxide formation.

The overall soot formation rate is modeled as the difference between soot formation and soot oxidation. Soot formation is based on Hiroyasu model [24] and the soot oxidation rate is adopted from Nagle and Strickland-Constable [25].

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

2.1 Model Validity

The numerical model for simulation of the Caterpillar 3401 heavy duty gasoline fueled engine with the specifications and operation conditions on the Table 1 is carried out using a three-dimensional CFD code.

Table 1- Engine specifications [17]

Engine type	Caterpillar 3401 heavy duty
Engine speed	1300 rpm
Bore × stroke	137.20 × 165.1 mm
Connecting rod length	261.6 mm
Compression ratio	16.1:1
Swirl ratio	0.7
Intake valve close timing	-85 ˚ATDC
Exhaust valve open timing	130 ˚ATDC

The injection system specifications which has been utilized in this engine, is shown in the Table 2.

Table 2- Injection system specifications [17]

Injector type	Caterpillar HEUI
Number of nozzle holes	8
Nozzle hole diameter	0.229 mm
Included spray angle	154 deg
Injection amount	5.3 kg/h
Injection strategy	Double

Considering the symmetry of the combustion chamber, problem is only solved for a 45 degrees sector. Application of symmetrical grid in engine parametrical investigation reduces the computation runtime and has a negligible error in the output data comparing to the full geometry [26]. Figure.1 shows the 45˚ sector of computational mesh for combustion chamber at TDC.

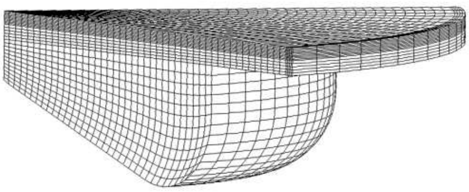


Fig. 1 Computational mesh at TDC

The generated geometry contains 18,385 cells at TDC. This resolution was found to give adequately grid independent results. This fine mesh size will be able to provide good spatial resolution for the distribution of most variables within the combustion chamber. The calculations are carried out on the closed system from Inlet Valve Closure (IVC) to Exhaust Valve Open (EVO).

Figure 2 shows the comparison of the numerical and experimental [17] in-cylinder pressure and heat release rate (H.R.R.) at two main injection timings, -8 ˚CA and -6 ˚CA ATDC. A good agreement of predicted in-cylinder pressure and H.R.R. with the experimental data can be observed.

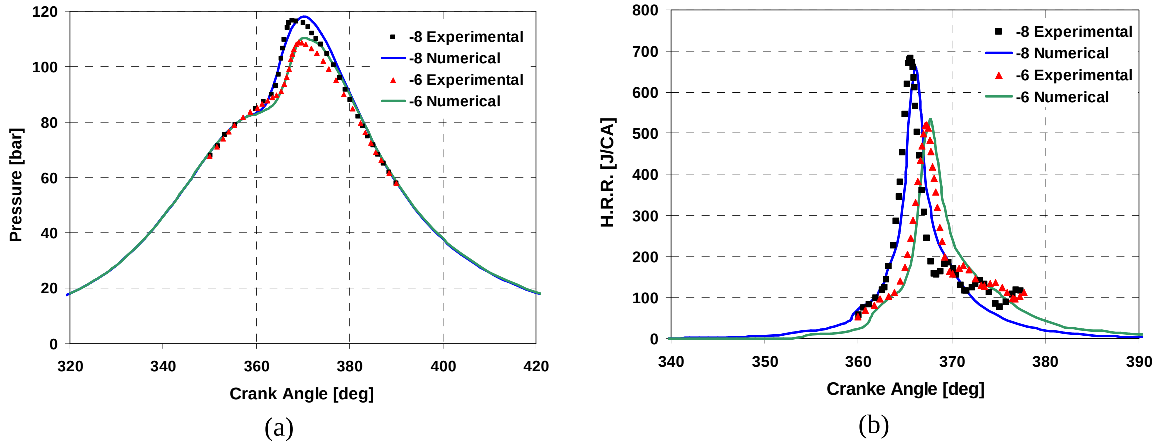


Fig 2. Comparison between numerical and experimental in-cylinder pressure and heat release rate, (a) pressure (b) heat release rate [17]

Figure 3 shows the comparison of the calculated NO_x and soot emissions with the experiments for two-stage injection scheme at four different main injection timings [17].

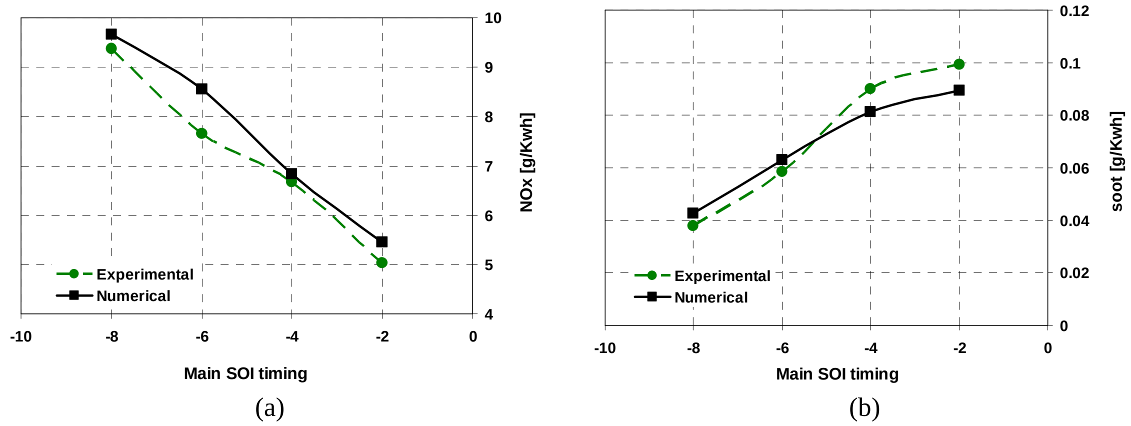


Fig 3. Comparison between numerical and experimental NO_x and soot emissions, (a) NO_x (b) soot [17]

As can be seen, the maximum deviation from the experimental data is less than 4% for in-cylinder pressure, so these agreements validates the model predictions and make it possible for the parametric investigations.

3. Results and Discussion

3.1. The effects of main injection duration

In order to investigate the effects of the main injection duration on the combustion characteristics and emissions formation, the injection duration has been varied from 6 to 18 ˚CA while the start of the main injection was fixed at -8 deg ATDC. The injection velocity is only dependent to the injection duration because the nozzle cross section area and total injected fuel have their constant amounts as the base case. So by decreasing the injection duration, the same amount of fuel should inject to the combustion chamber in shorter time and this leads to the higher injection velocity. Because of the same start of main injection for all cases, decreasing the injection duration led to the increase in peak cylinder pressures as can be seen in Figure 4a. Figure 4b shows the same trend for in-cylinder temperature.

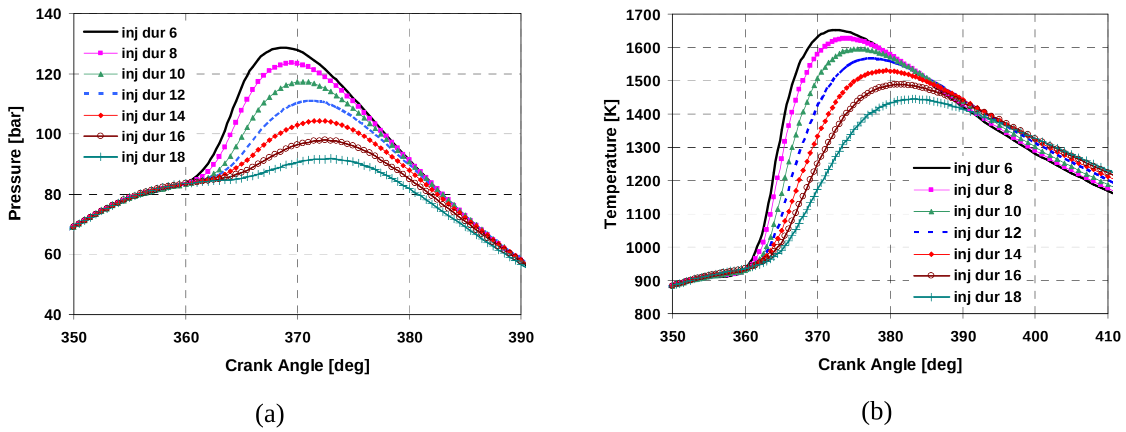


Fig 4. in-cylinder pressure and temperature under various injection durations, (a) pressure (b) temperature

Figure 5 shows the comparison of IMEP and ISFC for various injection durations. As can be seen, longer injection duration has decreased the IMEP and increased the ISFC. It can be said that the injection velocity is one of the major controlling parameters on engine performance. Fuel spray atomization was getting worse with the reduction of injection velocity and resulted in a bigger droplet sizes. So, homogeneity of air-fuel mixture is reduced and the spray impingement is increased, which led to more fuel stratification. Therefore, the combustion is became deficient and maximum in-cylinder pressure and temperature are decreased. The IMEP depends on the in-cylinder pressure and is treated in the same way.

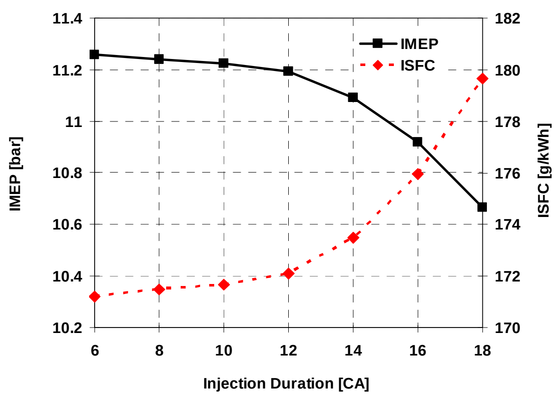


Fig 5. comparison of IMEP and ISFC under various injection durations

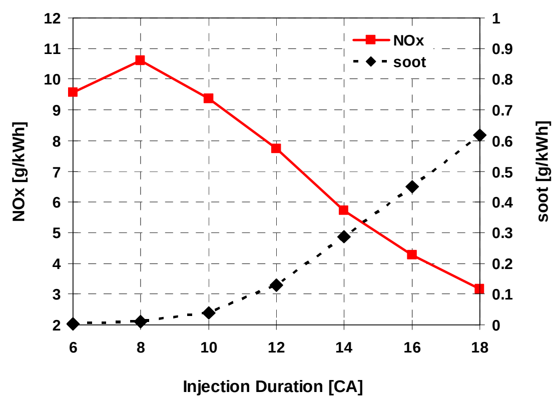


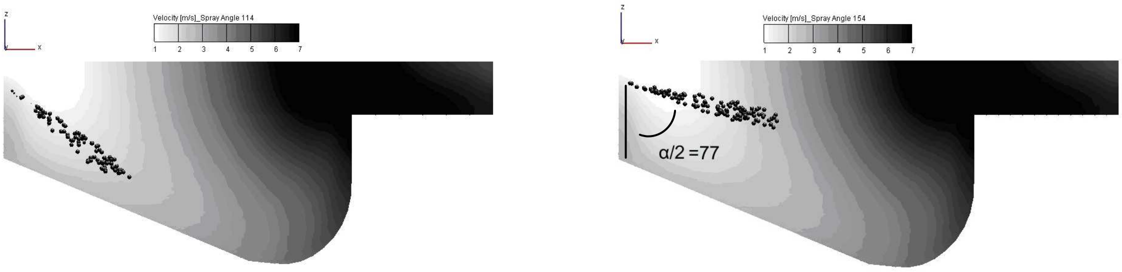
Fig 6. comparison of NO_x and soot under various injection durations

Figure 6 illustrates the NO_x and soot exhaust emissions for various injection durations. The local peak temperatures and the high temperature regions are reduced by increasing the injection duration and led to the reduction of NO_x formation. Soot emission is highly depends on the homogeneity of air-fuel mixture and forms in rich mixture regions. Higher injection duration reduces the injection velocity and leads to deterioration of the sprays atomization, increase of spray impingement and the fuel wall film formation, which leads to the additional fuel stratification. Increasing fuel stratification considerably increases soot emissions, as shown in Figure 6.

Regarding to the figure 6, it can be said that the injection duration is more effective on the soot formation than the NO_x. On the other hand, by reducing the injection duration, NO_x approximately is remained constant and soot is became zero. But further decrease in injection duration leads to higher injection velocities which cause to reduce the injector durability.

3.2. The effects of spray angle

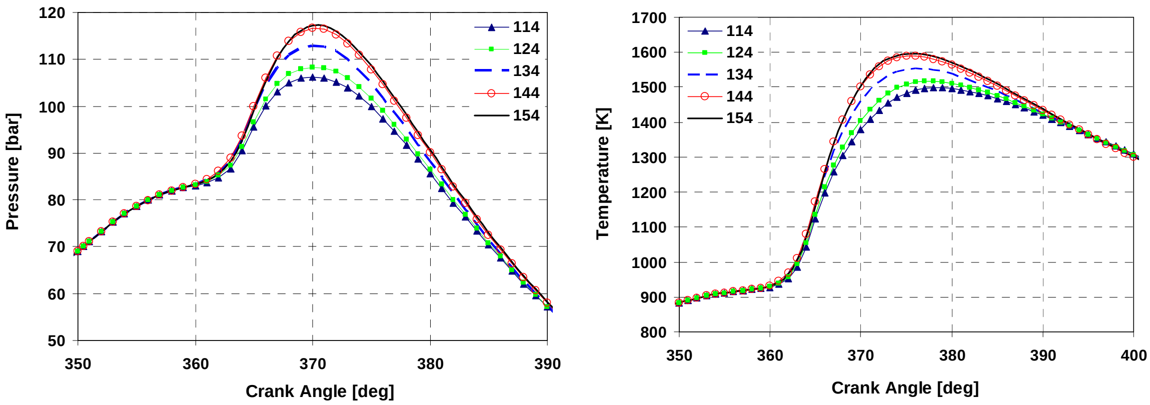
The spray angle was varied from ϴ=114˚ to 154˚ to study its effect on the engine power and emissions. This angle is measured between two spray cones which are projected on a plane. All of the other engine specifications are fixed at the base conditions. Figure 7 shows in-cylinder velocity contours before injection at -10 ˚CA ATDC and spray angle for 114˚ and 154˚.



114˚ 154˚

Fig 7. **in-cylinder velocity contour and spray angle**

As can be seen in the figure 7, with increase of the spray angle, the spray injects to the in-cylinder turbulent region, so squishing of the spray cone increases that leads to the enhanced spray atomization.



(a) (b)

Fig 8. in-cylinder pressure and temperature under various spray angles, (a) pressure (b) temperature

The enhanced atomization of spray leads to evaporation of larger portions of spray and improves the combustion which increases in-cylinder pressure and temperature as can be seen in Figure 8.

Figures 9 and 10 show the NO_x-soot and IMEP-ISFC trend diagrams respectively. The spray wall impingement and wall-film formation are decreased with the increase of the spray angle because of the improved fuel spray atomization. It can be said that increasing the fuel spray angle reduces the soot emission. NO_x depends on the in-cylinder temperature and increases with increasing the spray angle. As can be seen, the NO_x formation at 154˚ is less than 144˚. It is because of the lower ignition delay in 144˚ spray angle than 154˚ that leads to a stratified mixture when combustion begins and increases the NO_x formation. The indicated results represent that 154˚ spray angle is the best case form the emission point view.

The IMEP is increased and the ISFC is decreased with increasing the spray angle due to the improved combustion and higher in-cylinder pressure.

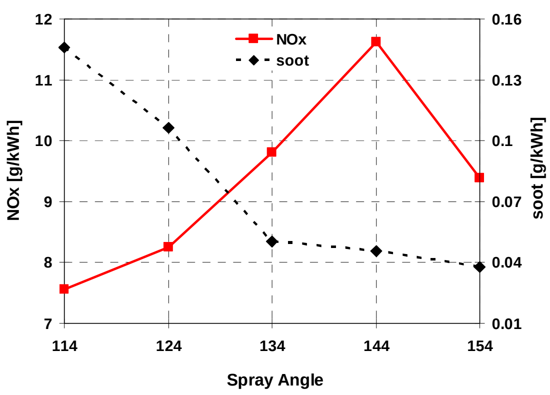


Fig 9. Predicted NO_x and soot emissions for various spray angles

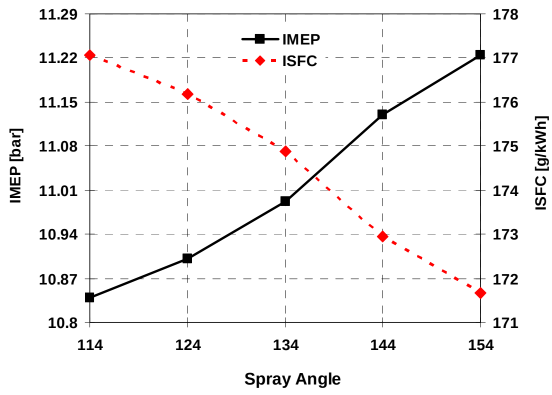


Fig 10. Predicted IMEP and ISFC for various spray angles

3.3. The effects of Nozzle Hole Diameter

In this section, the effects of nozzle hole diameter on emissions and combustion is discussed. The nozzle hole diameter for the base engine is 229 µm as mentioned in Table 2 and it is varied between 199 µm to 259 µm. The injection velocity is contrary depends on the nozzle hole size, because the injection duration and the injected mass are constant. So increasing the nozzle hole size leads to a reduction of injection velocity.

Figure 11 represents in-cylinder pressure and temperature as a function of crank angle for various nozzle hole diameters. In-cylinder pressure and temperature are decreased with increasing the nozzle hole diameter. Increasing the nozzle hole diameter causes to increase the droplet size and decrease the injection velocity. Droplet breakup is highly dependent on the injection velocity and droplet size, and decreased with the increase of the initial droplet size and reduction of the injection velocity. Decreasing the droplet breakup causes to decreasing the evaporation rate and leads to heterogeneity of air-fuel mixture. So combustion performs incompletely and causes to decrease of the in-cylinder pressure and temperature.

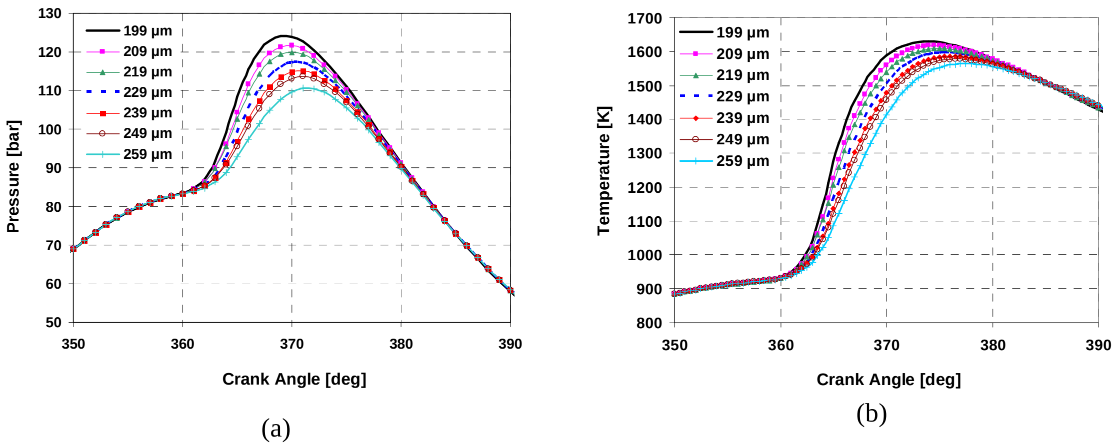


Fig 11. predicted cylinder pressure and temperature for various nozzle hole diameters, (a) pressure (b) temperature

Figure 12 shows the obtained results for NO_x and soot emissions for various nozzle hole diameters. As can be seen in figure 11, increasing the nozzle hole diameter decreases the in-cylinder temperature that leads to reduction of NO_x emission formation. As mentioned before, with increasing the nozzle hole diameter, the droplet size is increased and injection velocity is decreased. So the wall impingement and wall film formation increases that causes to the fuel stratification. All of these parameters have affected on the soot formation and it is increased with the increase of the nozzle hole diameter.

The gasoline spray evaporation deteriorates with increase of the nozzle hole diameter that leads to decreasing the in-cylinder maximum pressure and a slight increase in the gasoline ignition delay. So the IMEP and ISFC are improved. But as can be seen in figure 13 variation of IMEP and ISFC by varying the nozzle hole diameter is negligible.

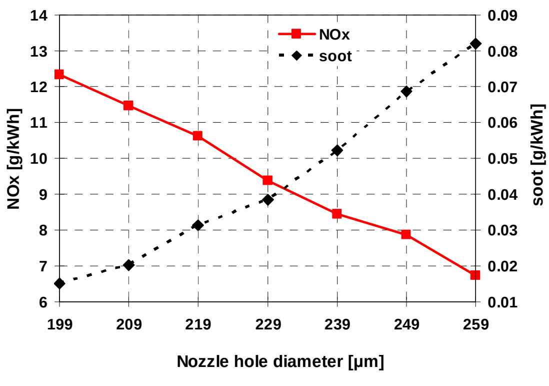


Fig 12. Predicted NO_x and soot emissions for various nozzle hole diameters

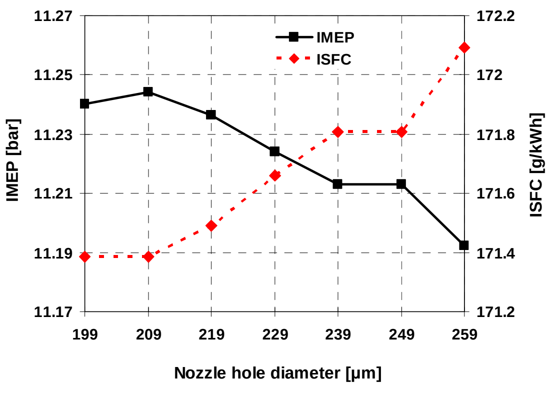


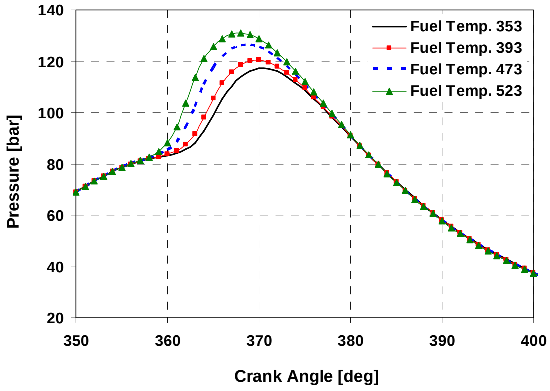
Fig 13. Predicted IMEP and ISFC for various nozzle hole diameters

3.4. The effects of fuel temperature

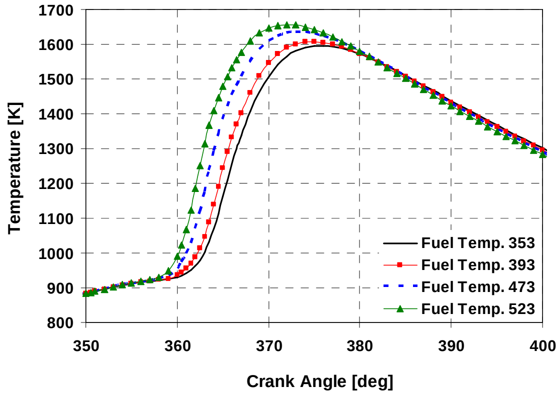
In the case of investigation of the initial fuel temperature, four cases have been mentioned as follows: T_{fuel} =353, 393 ˚K as low temperature and T_{fuel} =473, 523 ˚K as high temperature cases.

Figures 14 to 16 represent respectively the calculated in-cylinder pressure, temperature, NO_x-soot and IMEP-ISFC trend for various fuel initial temperatures.

The effect of increase of the fuel initial temperature emerges as increase of the maximum pressure and shorter ignition delay which yields in higher peak temperatures.



(a)



(b)

Fig 14. Predicted in-cylinder pressure and temperature under various fuel temperatures , (a) pressure (b) temperature

High mean temperature has resulted in increasing of NO_x emission considerably at T_{fuel} =473 and 523 ˚K (Fig.15). Also, at higher spray temperatures, due to increased spray atomization and improved combustion, larger portions of spray evaporate and wall-film thickness will decrease. At such conditions (higher atomization and better air-fuel mixing), exhaust soot mass fraction tends to decrease by about half compared to the maximum initial fuel temperature (T_{fuel} =523K).

IMEP increases and ISFC decreases by increasing the initial fuel temperature due to improved combustion and higher in-cylinder pressure.

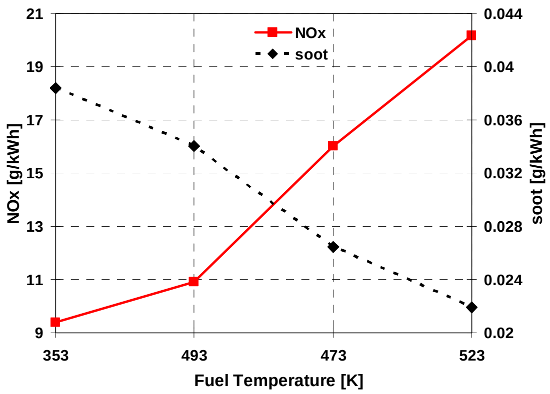


Fig 15. Predicted NO_x and soot emissions for various nozzle hole diameters

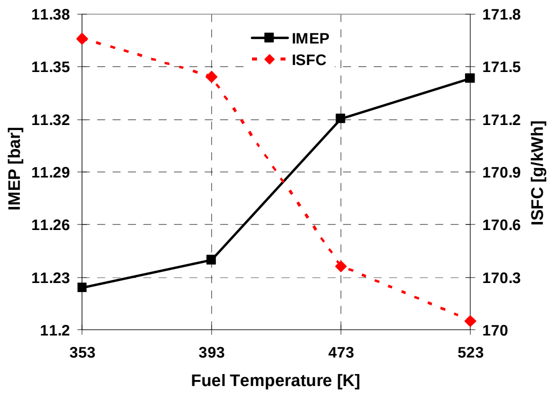
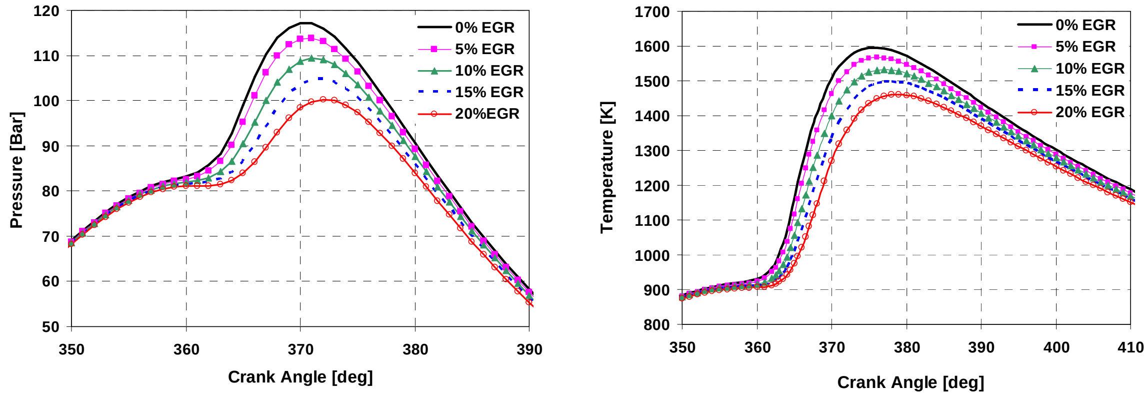


Fig 16. Predicted IMEP and ISFC for various nozzle hole diameters

3.5. The effects of EGR Ratio

The Exhaust gas recirculation (EGR) percent has been increased from 5% to 20%. All of the computation conditions were the same excepting the mixture composition at IVC. The EGR is one of the most cost-effective methods in the NO_x reduction. Figure 17 indicates that with the increase of the EGR ratio, the peak value of in-cylinder pressure and temperature are decreased. Due to utilization of the EGR, the oxygen availability decreases and causes to reduction of the fuel burning rate in the diffusion phase. This would tend to decrease combustion noise and the peak cylinder pressure.



(a) (b)

Fig 17. cylinder pressure and temperature variation under different EGR ratios, (a) pressure (b) temperature

There are three major parameters which affect on NO_x formation: the oxygen concentration, the energy for thermal decomposition of oxygen and nitrogen molecules, and the needed time for reaction of nitrogen and oxygen atoms depending on the engine speed. By increasing the ratio of cold EGR, the oxygen concentration decreases and because of increasing the CO_2 concentration in the inlet air, the energy consumption for CO_2 decomposition increases and less energy remains for thermal decomposition of O_2 and N_2 molecules. Therefore, by increasing the ratios of cold EGR, two effected parameters on NO_x formation are decreased that leads to reduction of NO_x formation, as can be seen in Figure 13. The EGR increases the local equivalence ratio through oxygen-deficient gas entrainment, increasing soot formation rates at low EGR rates [2]. Increasing the EGR is increased the gasoline ignition delay and is decreased the local equivalence ratio. Therefore by increasing the EGR ratio more than 10%, soot emission remains approximately constant as can bee seen in Figure 18.

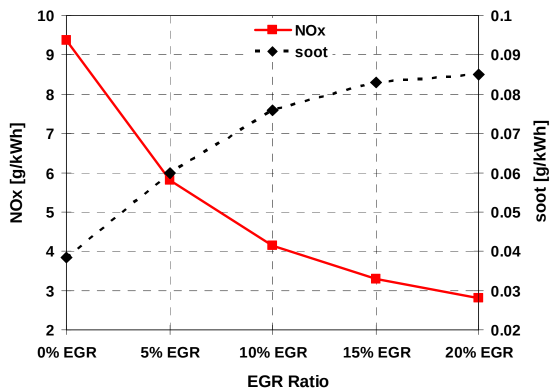


Fig 18. Predicted NO_x and soot emissions for various EGR ratios

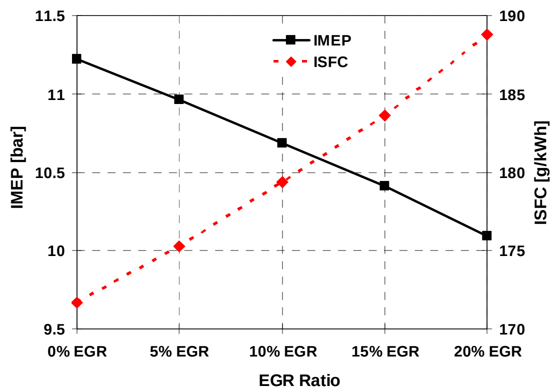


Fig 19. Predicted IMEP and ISFC emissions for various EGR ratios

Figure 19 shows IMEP and ISFC trend and represent that the IMEP is decreased and the ISFC is increased with increasing the cold EGR ratio due to decreasing the peak in-cylinder pressure.

3.6. The optimized point

As mentioned before, the injection duration is very effective on soot formation and can be utilized on soot reduction. NO_x depends on the EGR ratio and used to control the NO_x emission. -10 ˚ATDC main SOI timing, 6 ˚CA injection duration, 154˚ spray angle, 239 µm nozzle hole diameter, 353 ˚K injected fuel temperature and 15% EGR ratio are resulted as optimized injection characteristics and EGR ratio. The 15% of EGR is used because its further increase leads to more reduction in IMEP and power than NO_x. Also further decrease in injection duration leads to instability and decreasing injector durability. Figures 20, 21 and 22 show the comparison of IMEP, NO_x and soot between the optimized and base cases.

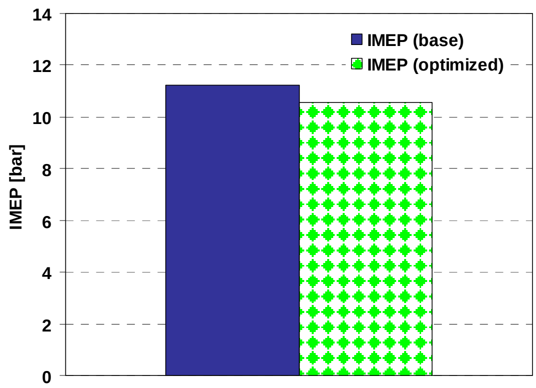


Fig 20. Comparison of IMEP between the optimized and base cases

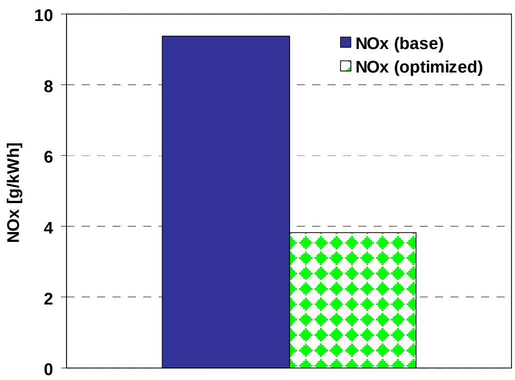


Fig 21. Comparison of NO_x between the optimized and base cases

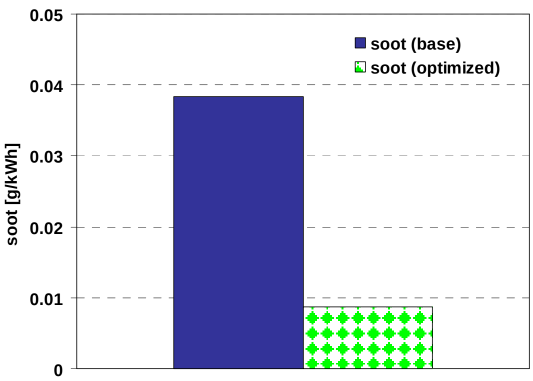


Fig 22. Comparison of soot between the optimized and base cases

As can be seen, with optimization of the injection characteristics for the gasoline fuel and EGR rate, the simultaneous reduction of NO_x and soot emissions can be achieved with a negligible reduction in IMEP.

4. Summery and Conclusion

In the present work, the effects of injection characteristics and EGR ratio have been investigated on combustion and emissions of a partially premixed gasoline fuelled heavy duty compression ignition engine using a three-dimensional CFD code.

The results for in-cylinder pressure, heat release rate, NO_x and soot emissions are compared with the corresponding experimental data at double injection scheme (the pilot injection 30% and main injection 70%) and show a good agreement and the following results were obtained:

- Increasing the injection duration led to reduction of the in-cylinder pressure and temperature due to decrease of the injection velocity and evaporation of gasoline spray. The results showed that soot is highly dependent on the injection duration and with the increase of the injection duration from 6˚ CA to 18˚ CA, it is increased from 0.0037 g/kWh to 0.6173 g/kWh. There was a significant trend in NO_x and soot emissions that by reducing the injection duration NO_x approximately remains constant but soot becomes zero simultaneous with a slight increase in IMEP.
- When the spray angle is increased, the squish of the spray cone is increased and led to the enhanced spray atomization and improved combustion. Therefore in-cylinder pressure and temperature, NO_x and IMEP are increased and soot and ISFC are decreased. The 154˚ spray angle is selected as the best option in order of emission formation.
- When nozzle hole diameter is increased, in-cylinder pressure, in-cylinder temperature and NO_x formation are decreased and soot is increased. It is because of the increase of droplet size with increasing the nozzle hole diameter which leads to the wall impingement and deficient combustion. The variation of nozzle hole diameter had a negligible effect on IMEP and ISFC.
- Results showed that increasing the fuel temperature led to the enhanced spray atomization and better combustion process and increase of the NO_x and decrease of the soot. Also increase of the fuel temperature has a good effect on IMEP and engine power.
- Increase of the EGR rate causes to decrease of NO_x due to the reduction of the oxygen concentration and needed energy for oxygen and nitrogen molecules thermal decomposition. With the increase of the EGR rate, soot is increased at lower EGR rates and then it remains approximately constant. Also, with increasing the EGR rates, in-cylinder pressure is decreased and led to the power loss.
- Considering the performed simulation, an optimized condition is found for gasoline injection and EGR rate which is able to simultaneously reduction of the NO_x and soot emissions against the slight decrease in IMEP.

There is scope for further improvements by using high EGR ratios and optimizing the initial pressure for further reduction of NO_x.

Nomenclature

k	Turbulence kinetic energy [m^2/s^2]
ε	Dissipation rate [m^2/s^2]

T Temperature

p Pressure

Greek letters

τ_R Turbulent mixing time scale

ρ Density [kg/m^3]

Abbreviations

EGR Exhaust gas recirculation

ATDC After top dead center

BTDC Before top dead center

IVC Intake valve close

EVO Exhaust valve open

SOI Start of injection

CA Crank angle

IMEP Indicated Mean Effective Pressure

ISFC Indicated Specific Fuel Consumption

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