

MULTI-DIMENSIONAL MODELING OF THE EFFECTS OF SPLIT INJECTION SCHEME ON COMBUSTION AND EMISSIONS OF DIRECT-INJECTION DIESEL ENGINES AT FULL LOAD STATE

*S. Jafarmadar**

*Faculty of Engineering, University of Urmia
P.O. Box 57561-15311, Urmia, Iran
s.jafarmadar@mail.urmia.ac.ir*

A. Zehni

*Faculty of Mechanical Engineering, University of Tabriz
P.O. Box 57561-15311, Tabriz, Iran
alborz.zehni@gmail.com*

*Corresponding Author

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Abstract One of the important problems in reducing pollutant emission from diesel engines is trade-off between soot and NOx. Split injection is one of the most powerful tools that decrease soot and NOx emissions simultaneously. At the present work, the effect of split injection on the combustion process and emissions of a direct-injection diesel engine under full-load conditions is investigated by the commercial CFD code AVL-FIRE. The study of injection timing and split injection parameters, including the delay dwell and the fuel quantity injected between injection pulses is carried out. Three different split injection schemes, in which 10-20-25 % of total fuel is injected in the second pulse, have been considered. The results show that 25 % of total fuel injected in the second pulse, reduces the total soot and NOx emissions effectively in DI diesel engines. In addition, the optimum delay dwell between the pulses is about 25°CA. The predicted values of combustion process, emission and delay dwell by this CFD model show a good agreement with the corresponding data of multi-zone phenomenological combustion model in the literature.

Keywords Split injection, Combustion, Emission, NOx, Soot

چکیده یکی از مهم ترین مسائل در کاهش آلاینده های موتورهای دیزلی، تقابل آلاینده های اکسید نیتروژن و دوده است. تزریق چندگانه یکی از قوی ترین ابزارها برای کاهش همزمان این دو آلاینده است. در این تحقیق با نرم افزار تجاری دینامیک سیالات محاسباتی فایر، تأثیر تزریق چندگانه بر فرآیند احتراق و آلاینده های موتور دیزلی تزریق مستقیم تحت بار کامل بررسی شده است. زمان تزریق و پارامترهای تزریق چندگانه که شامل زمان تأخیر و مقدار سوخت تزریق شده در خلال پالس های تزریق، مطالعه شده است. سه الگوی متفاوت تزریق که به ترتیب ۱۰، ۲۰ و ۲۵ درصد کل سوخت در پالس دوم تزریق می باشد در نظر گرفته شده است. نتایج نشان می دهند که ۲۵ درصد کل سوخت تزریق شده در پالس دوم آلاینده های اکسید نیتروژن و دوده به طور چشمگیر و همزمان در موتور دیزلی تزریق مستقیم کاهش می یابد. به علاوه، زمان تأخیر بهینه در خلال پالس های تزریق ۲۵ درجه میل لنگ است. نتایج پیشگویی شده احتراق، آلاینده های و تأخیر پالس های تزریق با مدل دینامیک سیالات محاسباتی استفاده شده، توافق خوبی را با نتایج حاصل از مدل های شبه چندبعدی موجود در ادبیات فن نشان می دهد.

1. INTRODUCTION

Stringent exhaust emission standards require the simultaneous reduction of soot and NOx. However

it seems to be very difficult to reduce NOx emission without increasing soot emission. The reason is that there always is a contradiction between NOx and soot emissions whenever the injection timing is

retarded or advanced. Split injection has been shown to be a powerful tool to simultaneously reduce soot and NO_x emissions when the injection timing is optimized. However, an optimum injection scheme of split injection for DI diesel engines has been always under investigation.

Li, et al [1] developed a multi-zone phenomenological combustion model, to study split injection parameters, including delay dwell between injection pulses, and the fuel quantity injected in the second pulse. The results predicted by modeling showed that the optimum delay dwell among injection pulses for reducing soot and NO_x emissions was about 25°CA.

Tow, et al [2] studied the effects of single, double and triple injection schemes on NO_x and soot emissions of a heavy duty DI diesel engine. The results showed that double and triple injection strategies, reduce NO_x without increasing soot while injection timing is retarded.

Bakenhus, et al [3] confirmed the results of Tow, et al [2] by means of combustion visualization techniques.

With developing the numerical methods and computers, it is interested to develop computational models, to help understanding the detailed effects of injection parameters upon combustion characteristics and emission, therefore CFD can greatly assist about this subject.

Patterson, et al [4] used the KIVA-II to study the effect of injection timing, injection pressure and split injection on emissions for a fast rising injection profile. The results show that when compared to the single injection case, an additional high temperature region is observed between the separate spray clouds with split injection. This promotes local NO_x formation while enhancing soot oxidation.

Bianchi, et al [5] investigated the capability of split injection in reducing NO_x and soot emissions of HSDI Diesel engines by CFD code KIVA-III. Computational results indicate that split injection is very effective in reducing NO_x, while soot reduction is related to a better use of the oxygen available in the combustion chamber.

Tatschi, et al [6] studied the impact of combustion variations such as EGR, SOI and split injection on a turbo-charged DI diesel engine by the CFD code FIRE. The predicted values by the modeling, showed a good agreement with the experimental data.

Shayler, et al [7] used the KIVA-III to investigate the influence of split main ratio and delay dwell on NO_x and soot emissions. Numerical conclusions show that when delay dwell is small, reducing ratio spreads the heat release variation, lowering soot but raising NO_x production. When delay dwell is large, the second injection has very little influence on soot production and oxidation associated with the first injection.

Chryssakis, et al [8] studied the effect of multiple injections on combustion and emissions of a DI diesel engine by using experimental flame visualization techniques and the multi-dimensional code KIVA-III. The results show that employing a post-injection combined with a pilot injection results in reduced soot formation from diffusion combustion and enhances the soot oxidation process during the expansion stroke, resulting in decreased smoke emissions, while the NO_x concentration is maintained in low levels.

At the present work the effect of split injection on combustion and pollution of a DI diesel engine has been investigated by the commercial CFD code AVL-FIRE. Study of injection timing and split injection parameters, including the delay dwell and the fuel quantity injected between injection pulses is carried out. The engine tests show maximum emissions of NO_x and soot at the engine speed of 1400 rpm and full load state [9]. Hence, the calculations are done for the mentioned above conditions. Three different split injection schemes in which 10-20-25 % of total fuel is injected in the second pulse have been considered.

2. PROBLEM STATEMENT

The specification and operating conditions of the diesel engine are described in Table 1.

The computational mesh was created using AVL ESE Diesel Tool [10]. Details of the computational mesh used are given in Figure 1. The computation used a 90 degree sector mesh (the diesel injector has four Nozzle holes) with 25 nodes in the radial direction, 20 nodes in the azimuthal direction and 5 nodes in the squish region at top dead center. The ground of the bowl has been meshed with two continuous layers for a proper calculation of the heat transfer through

TABLE 1. Engine Specifications [1].

Engine: OM-355 Diesel	Piston Shape: Cylindrical Bore
Make and Model: Mercedes-Benz	No. of Nozzles/Injector: 4
Type: Heavy Duty D.I Diesel Engine	Nozzle Opening Pressure: 195 (bar)
Cylinders: 6, in-Line-Vertical	IVC: 61°C A After BDC
Bore Stroke: 128 (mm) 150 (mm)	EVO: 60°C A Before BDC
Capacity: 11.58 (lit)	Max. Power: 240 (HP) at 2200 (rpm)
Compression Ratio: 16.1:1	Max. Torque: 824 N m at 1400 (rpm)

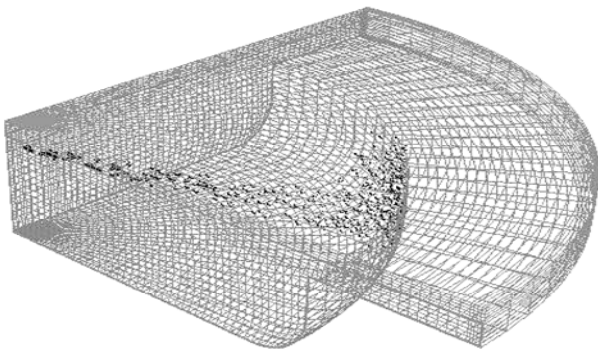


Figure 1. Computational mesh with diesel spray drops at 350°C, single injection case.

the piston wall. The final mesh consists of a hexahedral dominated mesh. Number of cells in the mesh was about 64,000 and 36,000 at BDC and TDC, respectively. The present resolution was found to give adequately grid independent results.

The governing equations for unsteady, compressible, turbulent flow and thermal fields were solved from IVC to EVO by the commercial CFD code AVL-FIRE [11]. The k-ε model was used for taking the turbulence field into account. Details about the in-cylinder flow in a DI engine with the similar specifications to OM-355 diesel engine have been studied in the reference [12].

2.1. Spray and Combustion Models The standard WAVE model, described in [13] was used for the primary and secondary atomization modeling of the resulting droplets. In this model the growth of an initial perturbation on a liquid surface is linked to its wave length and to other physical and dynamic parameters of the injected fuel and the domain fluid. Drop parcels are injected with characteristic size equal to the nozzle exit diameter (blob injection). The injection rate profile is rectangular type and consists of four injection schemes, i.e. single injection and three split injection cases.

The Dukowicz model was applied for treating the heat-up and evaporation of the droplets, which is described in [11,14]. 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.

A stochastic dispersion model was employed to take the effect of interaction between the particles and the turbulent eddies into account by adding a fluctuating velocity to the mean gas velocity [11]. This model assumes that the fluctuating velocity has a randomly Gaussian distribution. The spray-wall interaction model used in the simulations was based on the spray-wall impingement model described in [15]. This model assumes that a droplet, which hits the wall is affected by rebound or reflection based on the Weber number.

The Shell auto-ignition model was used for modeling of the autoignition [11,16,17]. 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 autoignition such as initiation, propagation, branching and termination were presented by generalized reactions, described in [11,16,17].

The Eddy Break-up model (EBU) based on the turbulent mixing was used for modeling of the combustion in the bowl [11]. 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. In other words, chemistry occurs fast and the combustion is mixing controlled.

2.2. Emission Models The Zeldovitch mechanism [11] was used for prediction of NO_x formation. The soot formation rate [11] is described as a model which is based on the difference between soot formation and soot oxidation.

3. RESULTS AND DISCUSSION

Figure 2 shows the cylinder pressure and the rate of heat release for the single injection case. As can be seen from HRR curve, the peak of the heat release rate occurs at 358°CA (2°CA before TDC). The premixed combustion occurs with a steep slope and it can be one of the major sources of NO_x formation. The good agreement of predicted in-cylinder pressure with the experimental data [9] can be observed.

Figures 3 and 4 imply that the predicted total in-cylinder NO_x and soot emissions for the single injection case, agree well with the engine-out measurements [9].

Figure 5 shows the trade-off between NO_x and soot emissions at EVO when the injection timing is varied. As indicated, the general trends of reduction in NO_x and increase in soot when injection timing is retarded can be observed and it is independent on injection strategy. The reason is that it causes the time residence and ignitions delay to be shorter, resulting in a less intense premixed burn and soot formation increases; in addition, the less temperature in different parts of combustion chamber keeps the soot oxidation low but decreases

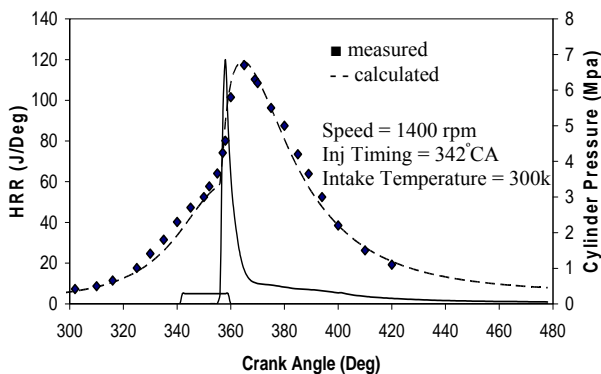


Figure 2. HRR and comparison of calculated and measured [9] in-cylinder pressure, single injection case.

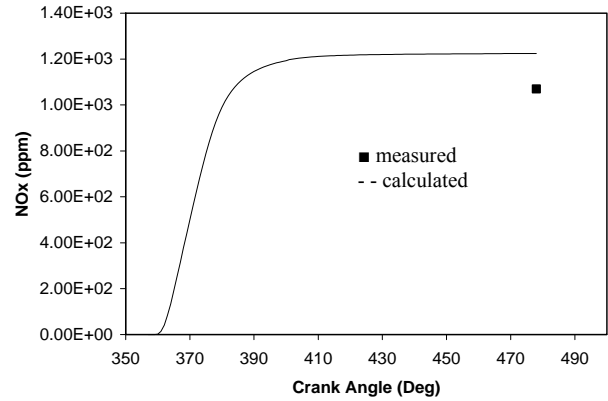


Figure 3. Comparison of calculated and measured [9] NO_x emission, single injection case.

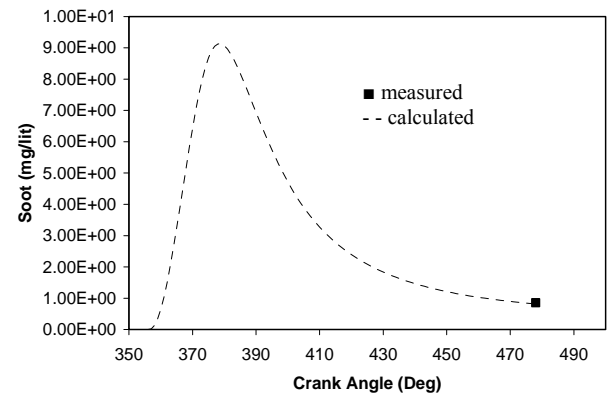


Figure 4. Comparison of calculated and measured [9] soot emission, single injection case.

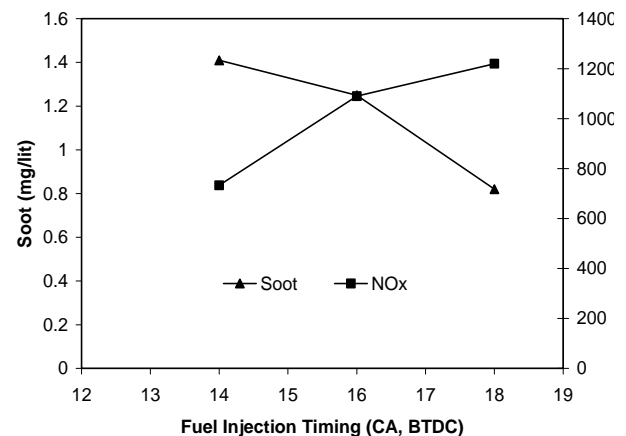


Figure 5. The effect of injection timing on NO_x and Soot trade-off, single injection case.

the formation of thermal NOx.

To simulate the split injection, the original single injection profiles are divided into two injection pulses without altering the injection profile and magnitude. In order to obtain the optimum dwell time between the injections, three different schemes including 10 %, 20 % and 25 % of the total fuel injected in the second pulse are considered.

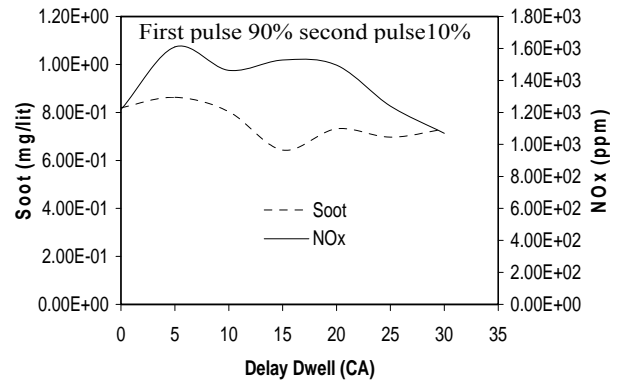
Figure 6 shows the effect of delay dwell between injection pulses on soot and NOx emissions for the three split injection cases. For all the cases, the injection timing of the first injection pulse is fixed at 342°CA.

The variation trend of curves in Figure 6 is very similar to the numerical results obtained by Li, et al [1]. As can be seen, the optimum delay dwell between the injection pulses for reducing soot with low NOx emissions is about 25°CA. The results of reference [1], which has used a phenomenological combustion model, confirm this conclusion.

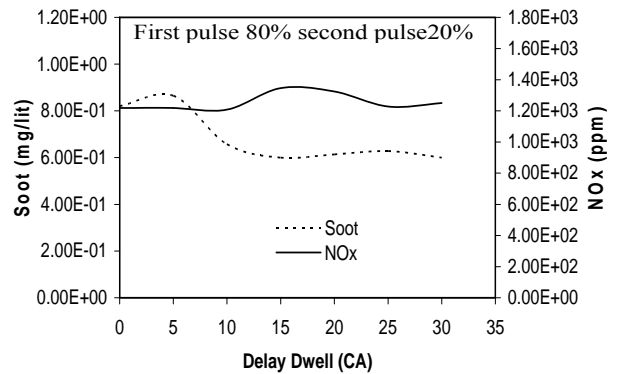
Table 2 compares Exhaust NOx and soot emissions for the single injection and optimum split injection cases for the optimum delay dwell. As shown, for the 75 % (25) 25 % case, NOx and soot emissions are lower than the other cases. It is due to the fact that the premixed combustion which is the main source of the NOx formation is relatively low. Higher amounts of the second injection pulse into the lean and hot combustion zones cause the newly injected fuel to burn rapidly and efficiently at high temperatures, resulting in high soot oxidation rates. In addition, the heat released by the second injection pulse is not sufficient to increase the NOx emissions. Figures 7 and 8 confirm the explanations.

Figure 7 shows the cylinder pressure and heat release rates for the three split injection cases in the optimum delay dwell i.e. 25°CA. As shown, split injection reduces the amount of premixed burn compared to the single injection case in Figure 2. The second peak which appears in heat release rate curves of split injection cases indicates that a rapid diffusion burn is realized at the late combustion stage and it affects the in-cylinder pressure and temperature. The calculated results of the cylinder pressure and HRR for the optimum split injection cases show very good similarity with the results of the reference [1].

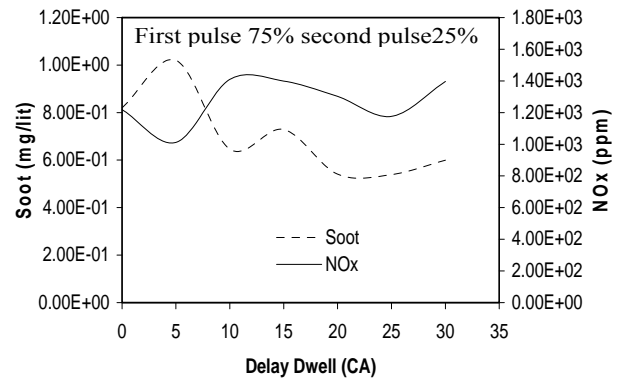
Figure 8 indicates the cylinder temperature for



(a)



(b)



(c)

Figure 6. The variation of soot and NOx at different delay dwells, split injection cases.

the single injection and optimum split injection cases. For the split injection cases, two peaks due to the first and second injection pulses in contrast with the one peak in the single injection case can

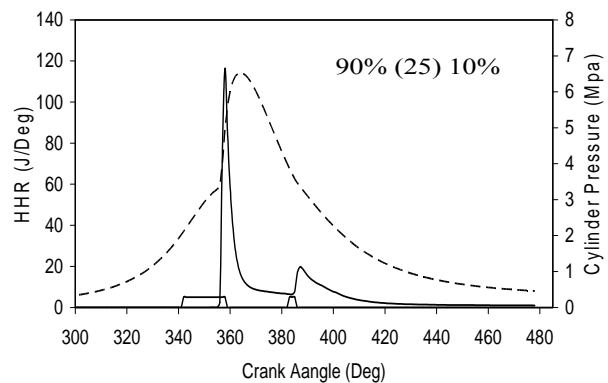
TABLE 2. Comparison of NO_x and Soot Emissions Among the Single Injection and Optimum Split Injection Cases for the Optimum Delay Dwell (25°CA).

Case	NO _x (ppm)	Soot (mg/lit)
Single Inj	1220	0.82
Split Inj - 90 % (25) 10 %	1250	0.697
Split Inj - 80 % (25) 20 %	1230	0.614
Split Inj - 75 % (25) 25 %	1180	0.541

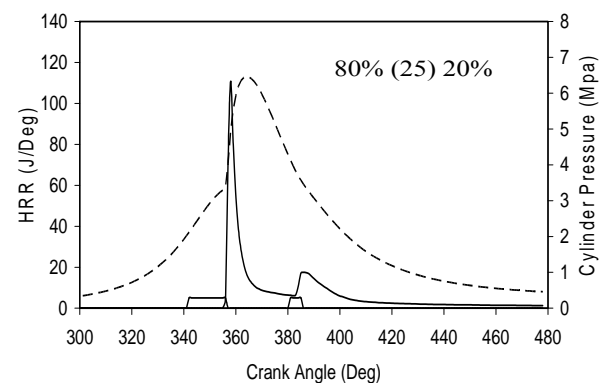
be observed. As shown, for the 75 % (25) 25 % case, the first and second peaks which are related to the premixed combustion and NO_x formation are lower than the other cases. In addition, after the second peak, the cylinder temperature tends to increase more in comparison with the other cases and causes more soot oxidation. Hence, for the 75 % (25) 25 % case, NO_x and soot emissions are lower than the other optimum cases.

Figure 9 shows the isothermal contour plots at different crank angle degrees for the 75 % (25) 25 % case at a cross-section just above the piston bowl. The rapid increase in temperature due to the stoichiometric combustion can be observed at 355°CA. The injection termination and maximum cylinder temperature can be observed at 360°CA. At 370°CA, the fuel injection has been cut off and the cylinder temperature tends to become lower. As described, injection termination and resumption prevents not only fuel rich combustion zones but also causes more complete combustion due to better air utilization. The resumption of the injection can be observed at 380°CA, which causes diffusion combustion and increases the temperature in the cylinder. At 390°CA, the increase in cylinder temperature creates bony shape contours and soot oxidation increases.

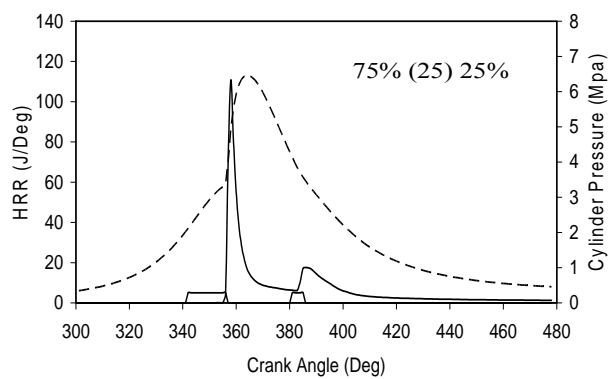
Figure 10 compares the contour plots of NO_x, temperature, equivalence ratio and soot for the single injection and 75 % (25) 25 % cases in a plane through the center of the spray at 380°CA. As explained above, 380°CA corresponds to a time when the second injection pulse has just started for the 75 % (25) 25 % case. For the two described cases, it can be seen that the area in which the equivalence ratio is close to 1 and the temperature is higher than



(a)



(b)



(c)

Figure 7. Cylinder HRR and pressure curves, optimum split injection cases.

2000 K is the NO_x formation area. In addition, the area which the equivalence ratio is higher than 3 and the temperature is approximately between 1600 K and 2000 K is the soot formation area.

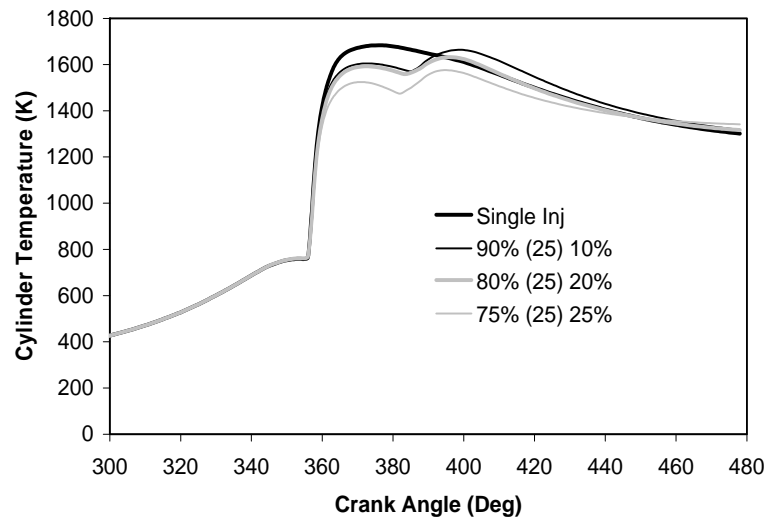


Figure 8. Comparison of cylinder temperature among the single injection and optimum split injection cases.

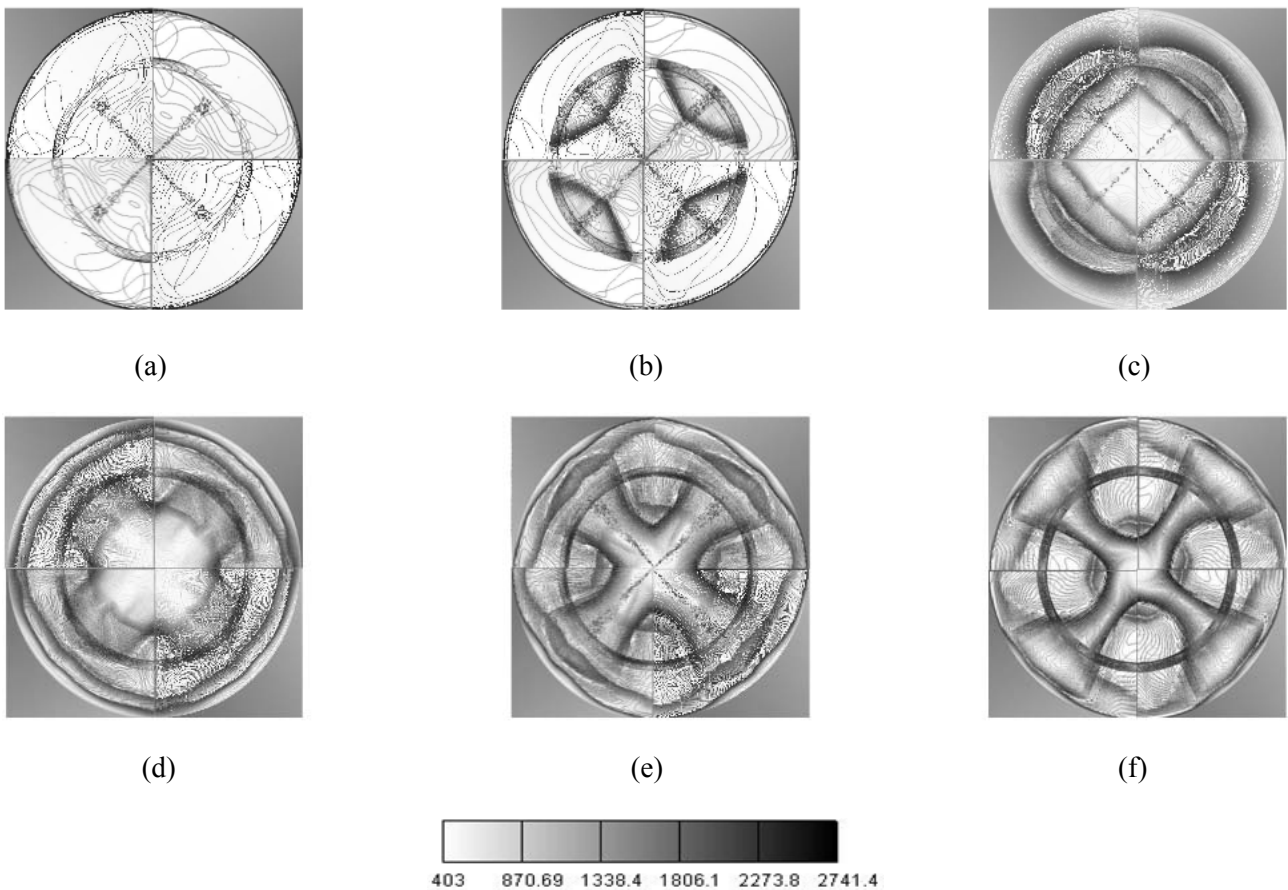
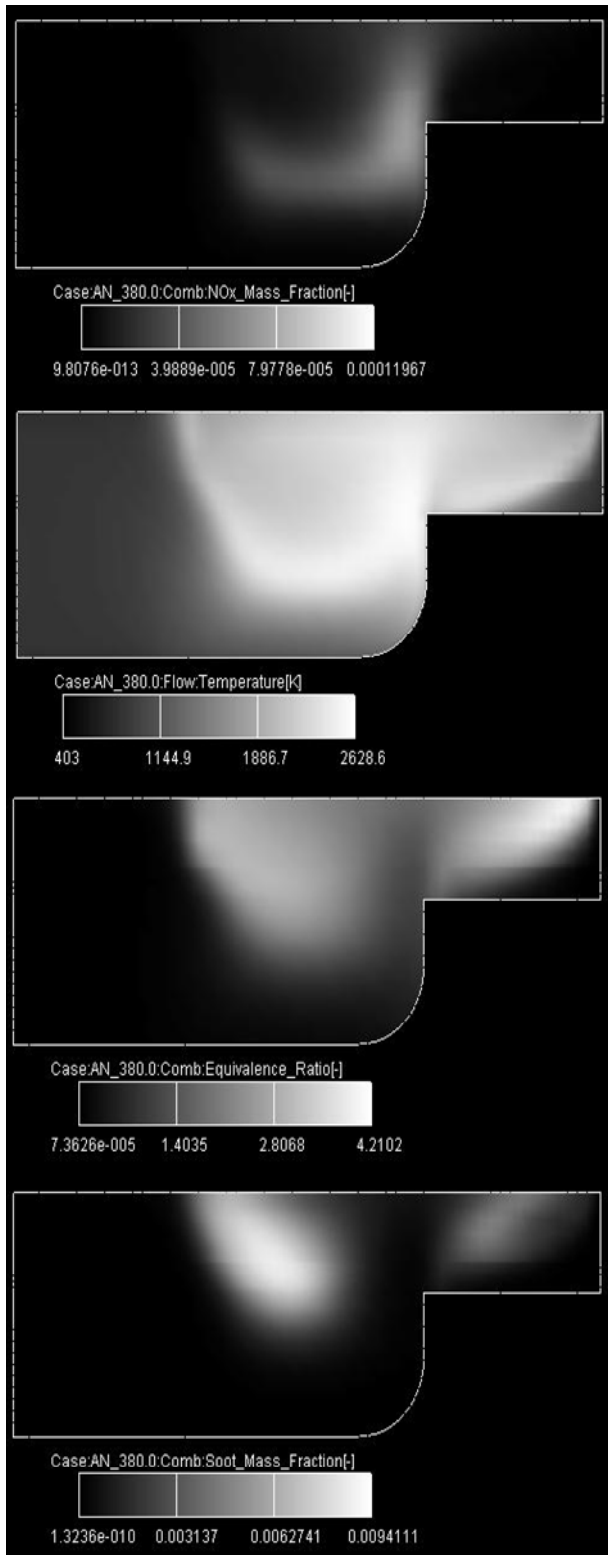
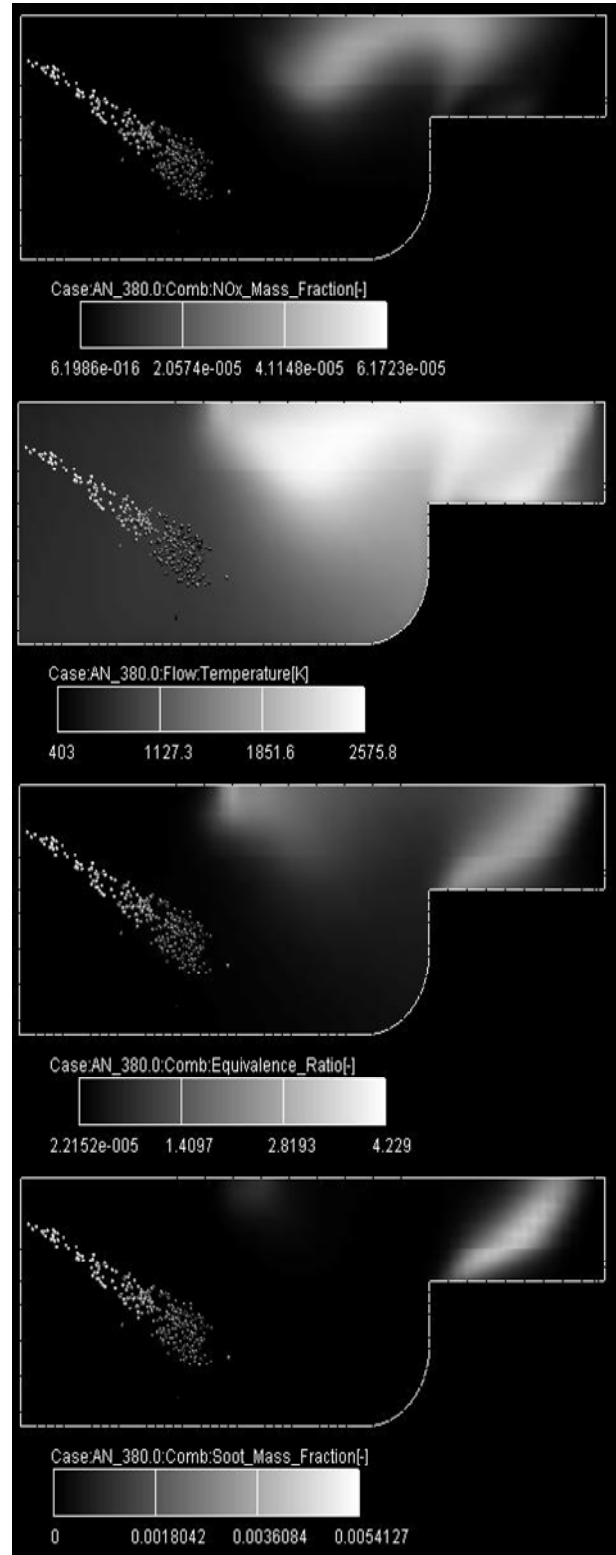


Figure 9. Isothermal contour plots of 75 % (25) 25 % case at different crank angle degrees, (a) 345°CA, (b) 355°CA, (c) 360°CA (d) 370°CA, (e) 380°CA and (f) 390°CA.



(a)



(b)

Figure 10. contour plots of NOx, temperature, equivalence ratio, and soot with fuel droplets at 380°CA for the single injection and 75 % (25) 25 % cases, (a): Single injection case and (b): Split injection case-75 % (25) 25 %.

A local soot-NOx trade-off is evident in these contour plots, as the NOx formation and soot formation occur on opposite sides of the high temperature region. It can be seen that for the 75 % (25) 25 % case, NOx and soot mass fractions are lower in comparison with the single injection case. Because of the optimum delay dwell, the second injection pulse, maintains the low NOx and soot emissions until EVO.

4. CONCLUSION

At the present work, the effects of split injection on combustion and pollution of a DI diesel engine have been investigated using multi-dimensional CFD code AVL-FIRE. The study of injection timing and split injection parameters, including the delay dwell between injection pulses and the fuel quantity injected in the second pulse were carried out. Hence, three different split injection schemes, in which 10-20-25 % of total fuel was injected in the second pulse, were considered. Because of the maximum emissions of NOx and soot at the engine speed of 1400 rpm and full load state, the calculations were based on the described conditions. It was concluded that:

- A good agreement among predicted in-cylinder pressure and exhaust NOx and soot emissions with the experimental data can be observed.
- Advancing or retarding the injection timing can not decrease the soot and NOx trade-off by itself. Hence split injection is needed.
- The optimum delay dwell between the injection pulses for reducing soot with low NOx emissions is about 25°CA. The results of phenomenological combustion models in the literature support this conclusion.
- The calculated results of the cylinder pressure and heat release rate for the optimum split injection cases show very good similarity with the numerical results obtained by phenomenological combustion models.
- For the 75 % (25) 25 % case, NOx and soot emissions are lower than the other cases. This is due to the fact that the premixed combustion which is the main source of the NOx formation is relatively low. The more

quantity of the second injection into the lean and hot combustion zones, leads to high soot oxidation rates. In addition, the heat released by the second injection pulse is not sufficient to increase the NOx emissions.

- Contour plots of NOx, temperature, equivalence ratio and soot for the single injection and 75 % (25) 25 % cases at 380°CA show that the area which the equivalence ratio is close to 1 and the temperature is higher than 2000 K is the NOx formation area. In addition, the area which the equivalence ratio is higher than 3 and the temperature is approximately between 1600 K and 2000 K is the Soot formation area.

5. NOMENCLATURE

BDC	Bottom Dead Center
CA	Crank Angle (Deg)
DI	Direct Injection
EGR	Exhaust Gas Recirculation
EVO	Exhaust Valve Opening
HSDI	High Speed Direct Injection
IVC	Inlet Valve Closure
SOI	Start of Injection
TDC	Top Dead Center
HRR	Heat Release Rate

6. REFERENCES

1. Li, J., Chae, J., Lee, S. and Jeong, J.S., "Modeling the Effects of Split Injection Scheme on Soot and NOx Emissions of Direct Injection Diesel Engines by a Phenomenological Combustion Model", *SAE Paper*, No. 962062, (1996).
2. Tow, T.C. and Piperpont, D.A., "Reducing Particulate and NOx Emissions by using Multiple Injections in a Heavy duty DI Diesel Engine", *SAE Paper*, No. 950897, (1995).
3. Bakenhus, M.D. and Reitz, R., "Two-Color Combustion Visualization of Single and Split Injections in a Single-Cylinder Heavy-Duty D.I. Diesel Engine using an Endoscope-Based Imaging System", *SAE Paper*, No. 1999-01-1112, (1999).
4. Patterson, M.A., Kong, S., Hampson, G.J. and Reitz, R.D., "Modeling the Effects of Fuel Injection Characteristics on Diesel Engine Soot and NOx Emissions", *SAE Paper*, No. 940523, (1994).

5. Bianchi, G.M., Peloni, P., Corcione, F.E. and Lupino, F., "Numerical Analysis of Passenger Car HSDI Diesel Engines with the 2nd Generation of Common Rail Injection Systems: The Effect of Multiple Injections on Emissions", *SAE Paper*, No. 2001-01-1068, (2001).
6. Tatschi, R., Gabriel, H.P. and Priesching, P., "Fire-a Generic CFD Platform for DI Diesel Engine Mixture Formation and Combustion Simulation", *User's Group Meeting at the SAE Congress*, Detroit, M.I., (March 4, 2001).
7. Shayler, P.j. and Ng, H.K., "Simulation Studies of the Effect of Fuel Injection Pattern on NOx and Soot Formation in Diesel Engines", *SAE Paper*, No. 2004-01-0116, (2004).
8. Chryssakis, C.A., Assanis, D.N., Kook, S. and Bae, C., "Effect of Multiple Injections on Fuel-Air Mixing and Soot Formation in Diesel Combustion using Direct Flame Visualization and CFD Techniques", *Spring Technical Conference, ASME*, No. ICES2005-1016, Chicago, I.L., U.S.A., (April 5-7, 2005).
9. Pirouzpanah, V. and Kashani, B.O., "Prediction of Major Pollutants Emission in Direct-Injection Dual-Fuel Diesel and Natural Gas Engines", *SAE Paper*, No. 1999-01-0841, (1999).
10. FIRE Engine Simulation Environment user Manual, V. 8.5, (2006).
11. AVL FIRE user Manual, V. 8.5, (2006).
12. Payri, F., Benajes, J., Margot, X. and Gil, A., "CFD Modeling of the in-Cylinder Flow in Direct-Injection Diesel Engines", *Computers and Fluids*, Vol. 33, (2004), 995-1021.
13. Liu, A.B. and Reitz, R.D., "Modeling the Effects of Drop Drag and Break-up on Fuel Sprays", *SAE Paper*, No. 930072, (1993).
14. Dukowicz, J.K., "Quasi-Steady Droplet Change in the Presence Of Convection", *Informal Report Los Alamos Scientific Laboratory*, LA7997-MS.
15. Naber, J.D. and Reitz, R.D., "Modeling Engine Spray/Wall Impingement", *SAE Paper*, No. 880107, (1988).
16. Halstead, M., Kirsch, L. and Quinn, C., "The Auto Ignition of Hydrocarbon Fueled at High Temperatures and Pressures-Fitting of a Mathematical Model", *Combustion Flame*, Vol. 30, (1977), 45-60.
17. Khoshbakhti Saray, R., "Enhancement of Combustion Process in Dual Fuel Engines at Part Loads by using Suitable Techniques", *IJE, Transactions B*, Vol. 22, No. 1, (2009), 77-90.