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Effect of spark plug and fuel injector location on mixture stratification in a GDI engine - A CFD analysis

O P Saw[1] and J M Mallikarjuna[2]
[1] MS Research Scholar, Indian institute of technology Madras, Chennai, India
[2] Associate Professor, Indian institute of technology Madras, Chennai, India

E-mail: opsawiitm@gmail.com[1], jmmallik@iitm.ac.in[2]

Abstract: The mixture preparation in gasoline direct injection (GDI) engines operating at stratified condition plays an important role in deciding the combustion, performance and emission characteristics of the engine. In a wall-guided GDI engine, with a late fuel injection strategy, piston top surface is designed in such a way that the injected fuel is directed towards the spark plug to form a combustible mixture at the time of ignition. In addition, in these engines, location of spark-plug and fuel injector, fuel injection pressure and timing are also important to create a combustible mixture near the spark plug. Therefore, understanding the mixture formation under the influence of the location of spark plug and fuel injector is very essential for the optimization of the engine parameters. In this study, an attempt is made to understand the effect of spark plug and fuel injector location on the mixture preparation in a four-stroke, four-valve and wall-guided GDI engine operating under a stratified condition by using computational fluid dynamics (CFD) analysis. All the CFD simulations are carried out at an engine speed of 2000 rev/min., and compression ratio of 10.6, at an overall equivalence ratio (ER) of about 0.65. The fuel injection and spark timings are maintained at 605 and 710 CADs respectively. Finally, it is concluded that, combination of central spark plug and side fuel injector results in better combustion and performance.

1. Introduction

The GDI engine technology has received considerable attention over the last few years as a way to improve fuel efficiency along with meeting future emission norms in spark ignition (SI) engines [1]. In a wall guided GDI engine, fuel is injected directly into the combustion chamber at a high pressure [2]. In these engines, air-fuel mixing in a short duration poses a major challenge. In order to assist mixing of air with fuel, piston top profile is shaped to enhance swirl and tumble flows [3]. However, intake port angle, fuel injector orientation, fuel spray foot print etc., play major role in the mixture formation [4-5].

However, the mixture stratification mainly depends on the engine operating conditions viz., intake port design, combustion chamber geometry, engine speed and compression ratio, position of fuel injector etc. Whereas, the combustion characteristics of the engine depend upon in-cylinder flows, location of spark-plug, mixture stratification inside the combustion chamber etc. Among these, the location of fuel injector and spark plug together impact largely on fuel-air mixing, combustion, performance and emission characteristics.

Previously Church et al., [2] conducted experiments, using a pulsed laser and high-speed imager for particle tracking to quantify the effect of change in intake port geometry on in-cylinder motions in a modular engine with replaceable intake port blocks. They used three different intake ports and found

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that the in-cylinder flows exhibited large-scale motions up to 90° before TDC, but they disappeared at TDC. The smaller intake port angles had higher tumble ratios at BDC and at 90° before TDC. While, the larger port angles had significantly lower tumble ratios at BDC.

Costa et al., [7] studied mixture preparation and combustion in a GDI engine under stoichiometric and lean conditions. Experiments were carried out in an optically accessed turbocharged GDI engine at the engine speed of 1500 rev/min. The engine performance and emissions were studied for the fuel injection pressures of 15 and 6 MPa under the stoichiometric and lean conditions respectively. Their results showed that the stoichiometric mixture resulted in higher cumulative heat release compared to that of the lean stratified mixture. They reported that the higher injection pressure resulted in improved combustion in the case of lean stratified mixture. The NO emissions were higher in the case of stoichiometric mixture, compared to that of lean mixture. Soot emission trends were opposite to that of NO emissions. They also reported that increasing fuel injection pressure and advancing the ignition timing improved power output and lowered the cyclic variation. Also, CO and soot emissions reduced, but increased NO emissions.

Li et al., [8] investigated the effect of split fuel injection with various dwells and mass ratios on stratified charge formation in a GDI engine. The experiments were performed using a swirl type injector in a container filled with the nitrogen gas. The laser absorption and scattering (LAS) technique was used to measure the concentration of liquid and vapour phase sprays. The laser induced fluorescence-particle image velocimetry (LIF-PIV) technique was used for analysing the spray and ambient air flows. The tests were carried out for ambient pressures of 1 and 0.6 MPa, for two ambient temperatures of 500 and 300 K. The fuel injector pressures of 5 and 4.6 MPa; and total fuel quantity of 9.4 and 14 mg were used. They reported that an optimum combustion in a GDI engine could be achieved by using two fuel injection pulses. They found that the penetration of liquid fuel decreased with increased dwell period or percentage of fuel mass in the first pulse, but with a marginal reduction in the penetration of the vapour fuel.

Costa et al., [9] analysed mixture formation and early flame development in a GDI engine through numerical simulations and ultraviolet digital imaging. They investigated the effect of injection and ignition timing on flame propagation development. The tests were carried out at full-throttle operation, compression ratio (CR) of 10.5 and at the engine speed of 2000 rev/min., in a four-valve, single-cylinder wall-guided GDI engine. The fuel injection pressure of 100 bar and equivalence ratio of 1.1±1% were used. The flame growth was detected by an intensified CCD camera and the AVL-FIRE was used to perform simulations. They reported that the early fuel injection resulted in the formation of homogeneous mixture which led to stable combustion. They also noted that the stratified mixture increased cycle-by-cycle variations.

From the above discussion, it is understood that, the mixture formation, in a GDI engine, is influenced significantly by many engine parameters and conditions. Also, in a wall guided GDI engine, it mainly depends upon piston shape, engine speed, fuel injection strategy, etc. However, studies on the evaluation of the effects of fuel injector and spark plug location on mixture formation are rarely reported in the literature. Therefore, the present CFD analysis investigates the effect of fuel injector and spark plug location on mixture formation in a wall guided GDI engine.

2. CFD Methodology

2.1. Engine geometry and meshing

The CFD analysis has been done on a single-cylinder, four-stroke, four-valve, wall guided GDI engine which has a pentroof cylinder head. The spark plug is considered to be placed at the centre of cylinder head and the fuel injector is placed between the two intake ports as shown in Figures 2(a) and (b). The piston has a pentroof shape with an offset bowl as shown in Figure 2 (c). The engine specifications are given in the Table 1 [9]. A single fuel injection system has been used and the details about fuel injection events are given in Table 2 [9].

Table 1. Engine specifications

Parameter	Specifications
Displaced volume	398.5 cm^3
Stroke	81.3 mm
Bore	79 mm
Connecting rod	143 mm
Compression ratio	10.6 : 1
Speed (rpm)	2000
Charge induction system	Naturally aspirated
Spark timing	10^0 before TDC
Number of valves	4
Exhaust valve opening	27° before BDC
Exhaust valve closure	0° after TDC
Inlet valve closure	36° after BDC
Inlet valve opening	3° before TDC
Intake valve lift	6.5 mm
Exhaust valve lift	6.5 mm

(*360 CAD is suction at TDC)

Table 2. Fuel injection system parameters

Parameter	Detail
No. of holes	Six
Fuel	Gasoline (C_8H_{18})
Start of fuel injection	115 before TDC
Fuel injection duration	23.1 CAD
Fuel injection pressure	120 bar
Spray cone angle	15 degree
Mass of fuel injected per cycle	22 mg

2.2. Cases considered in this study

In the present study, four cases have been studied with different location of the spark plug (SP) and fuel injector (FI). Figure 1 shows the top views of the engine, for all the cases.

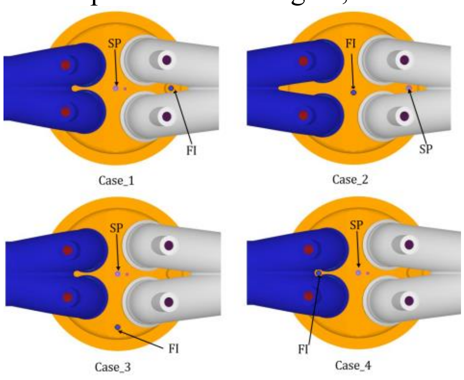

Figure 1. SP and FI location in all four cases considered

IOP Conf. Series: Materials Science and Engineering **243** (2017) 012025 doi:10.1088/1757-899X/243/1/012025

2.3. CFD Modelling and Boundary conditions

In the present study, the CFD analysis has been carried out using the CONVERGE and the post processing of the 3D output is done by using EnSight 10.1. Here, the SAGE model is used for the analysis of combustion, this model is based on the detailed chemical kinetics with modifications of Givler et al., [10]. This model considers 152 reactions along with 48 species. For modelling the ignition event, a high temperature source between the spark plug electrodes is introduced at the time of ignition. Modeling of wall heat transfer is done by the O'Rourke and Amsden models [11]. The droplet collision and initial droplet distribution has been analyzed by the NTC collision model [12] and the blob injection model [12] respectively. Film splash and fuel turbulent dispersion is analysed by the O'Rourke model [13]. The FROSSLING model is used for the analysis of droplet evaporation [14]. The turbulence is analysed using the RNG k-ε model [15]. Whereas, the pressure-velocity coupling is done by using the PISO algorithm [16]. The KH-RT model is used to study the spray breakup [17]. The calculation of embedded cell size, shown in figure 2 (e), is done by equation 1.

$$\text{Cell size after embedding} = \frac{\text{Cell size before embedding}}{2^{\text{embed scale}}} \tag{1}$$

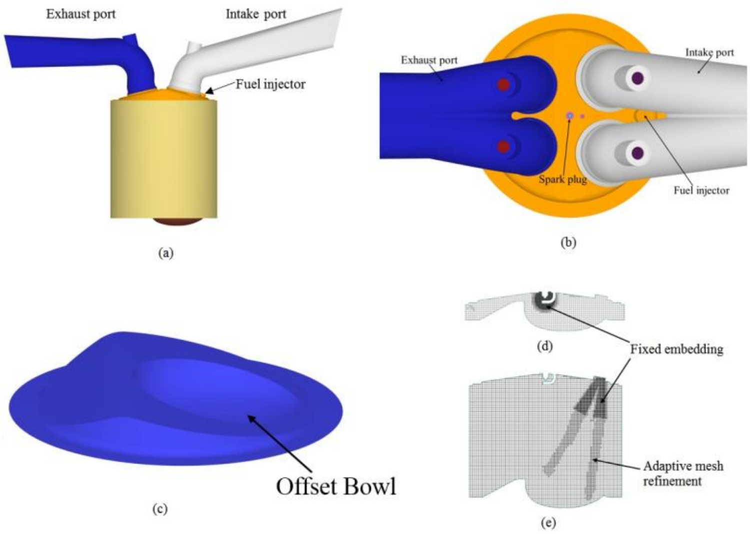

Figure 2. (a) Computational domain of the engine (b) Top view of engine (c) Piston with offset piston (d) Embedding at Spark (e) Adaptive mesh refinement at fuel injection

2.4. Governing equations

To generalize the in-cylinder flows, the following governing equations are used *[20]*:
Conservation of mass:

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\rho u_j)}{\partial x_j} = s_m \tag{2}$$

General transport equations:

$$\frac{\partial(\rho\varphi)}{\partial t} + div(\rho\varphi u) = div(\Gamma grad\,\varphi) + S_\varphi \tag{3}$$

where, ϕ is a general variable of conservative form of all fluid flow equations including equations for scalar quantities such as temperature and pollutant concentration, etc.

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3. Validation of the CFD Models

The validation of the CFD models for the same engine has been already done by Krishna et al., [13] in their previous work. They did the validation by using experimental results available in the literature [9]. Some of the validation results of them are shown here [13]. Figure 3 shows the comparison of in-cylinder pressures with crank angles, obtained from the experimental and CFD results [13]. Figure 4 shows the comparison of the equivalence ratio (ER) distribution between the CFD and experimental results of Costa et al., [9]. From Figures 3 and 4, it can be seen that, the two results are in good agreement. Therefore, these models can be used with confidence for the further analysis.

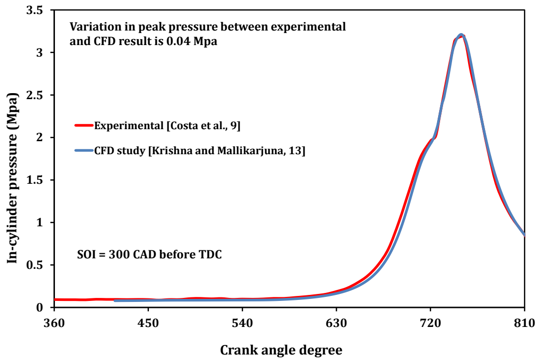


Figure 3. Comparison of in-cylinder pressure with crank angle degree

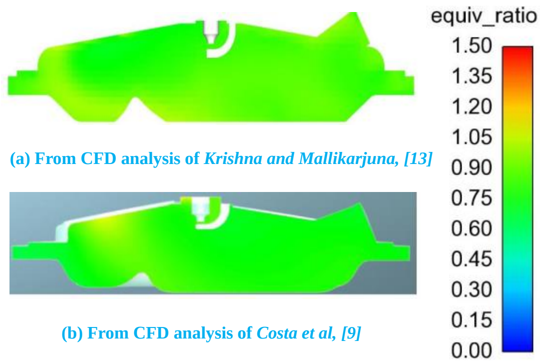


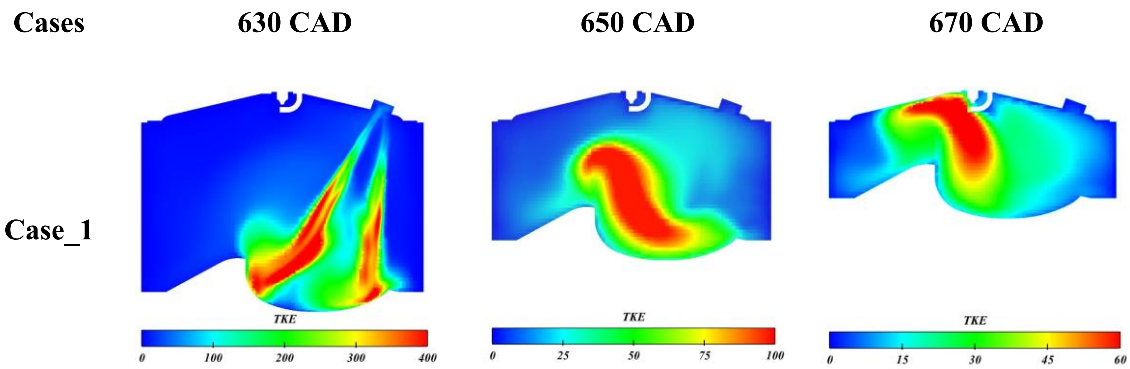
Figure 4. Comparison of ER distribution on central plane at spark timing

4. Result and discussions

In this section, the effect of the spark plug (SP) and the fuel injector (FI) location on various parameters has been discussed.

4.1. Effects of SP and FI location on Turbulent kinetic energy

Figure 5 shows the effect of the SP and FI location on turbulent kinetic energy (TKE) distribution for all the four cases considered at various crank angles. From Figure 5 it is seen that at 630 CAD the TKE is almost similar for cases 1, 2 and 3 whereas for case 4 the TKE is lower than that of the other cases. But at 650 CAD the TKE is maximum for case 1 followed by case 3, 2 and 4, this improves the fuel-air interaction for case 1 and hence better ER distribution. Also similar trend as of 650 CAD is also observed at 670 CAD. From Figure 5 it is observed that case 1 produces highest TKE.



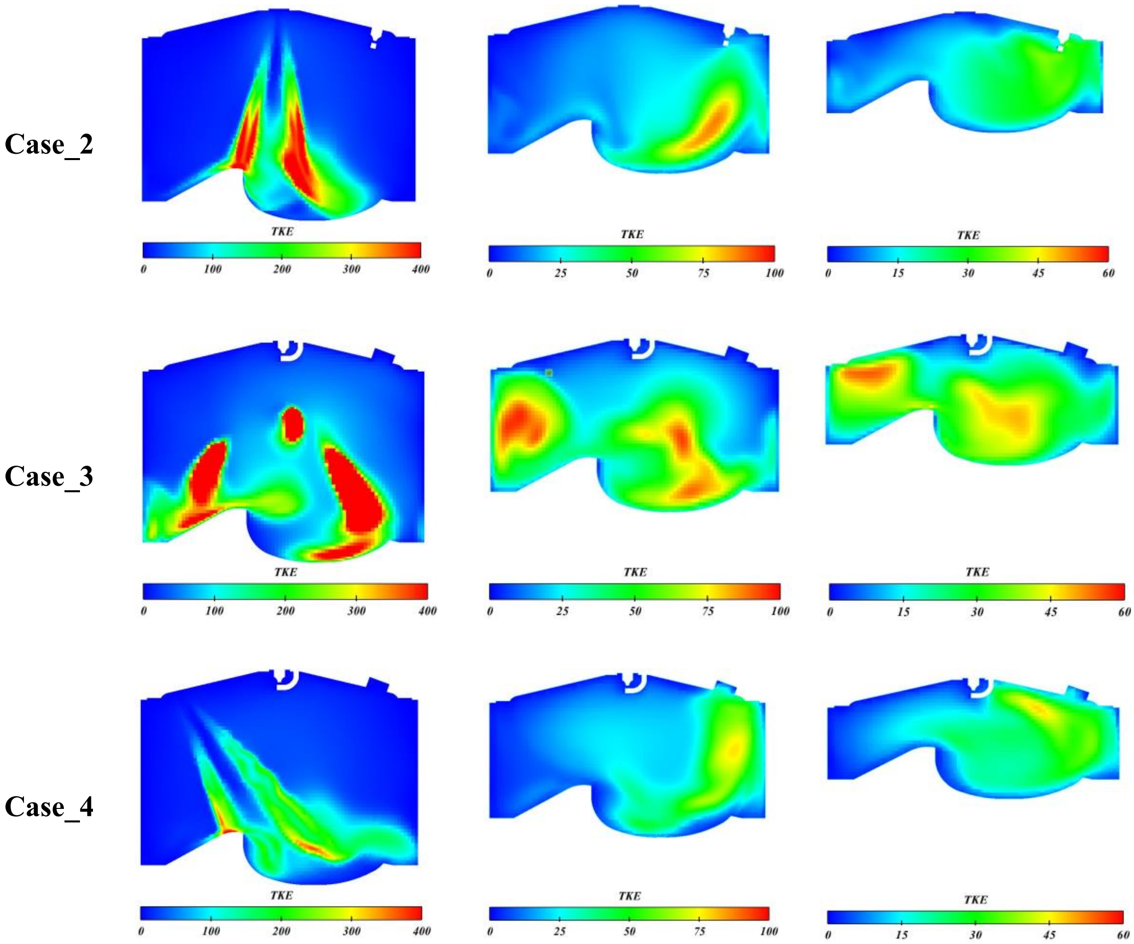


Figure 5. Comparison of TKE on the central plane at various CADs

4.2. Effects of SP and FI location on ER distribution

Figure 6 shows the effect of the SP and FI location on ER distribution for all the four cases considered. The ER around the spark plug, at the time of spark, for the cases 1, 2, 3 and 4 are about 0.99, 1.89, 1.18 and 1.77. Also, it is observed that, the distribution of the mixture is most favourable in the case 1 followed by the case 3, case 4 and case 2. Better ER distribution in the case 1 and case 3 results in better flame propagation which in turn results in better combustion. Whereas, in the case 2 and case 4, the fuel-air mixture is too rich around the spark plug, which leads to quenching of flame and then it becomes difficult for flame propagation. This is because of the higher TKE inside the combustion chamber for case 1 and 3 as discussed in Section 1.5.

4.3. Effects of SP and FI location on Mixture Stratification

In this study, the mixture stratification is characterized by a parameter called "stratification index" (SI) developed by Krishna and Mallikarjuna [13*]*. The SI is calculated based on the ER distribution in the hemispherical zones with spark plug as the center, as shown in Figure 7. In any combustion chamber, the value of SI varies between 0 and 1. The SI value of 0 indicates an ideally homogenous mixture (the ER is constant throughout the combustion chamber) and the SI value of 1 indicates an ideally stratified mixture (the ER varies continuously from higher to lower value away from the spark plug location). A negative value of the SI indicates a mal-distributed mixture (the ER varies continuously from a lower to higher value away from the spark plug location) in the combustion chamber. However, in a GDI engine, the SI value of 0.5 to 1 is desirable and closer to 1 is better.

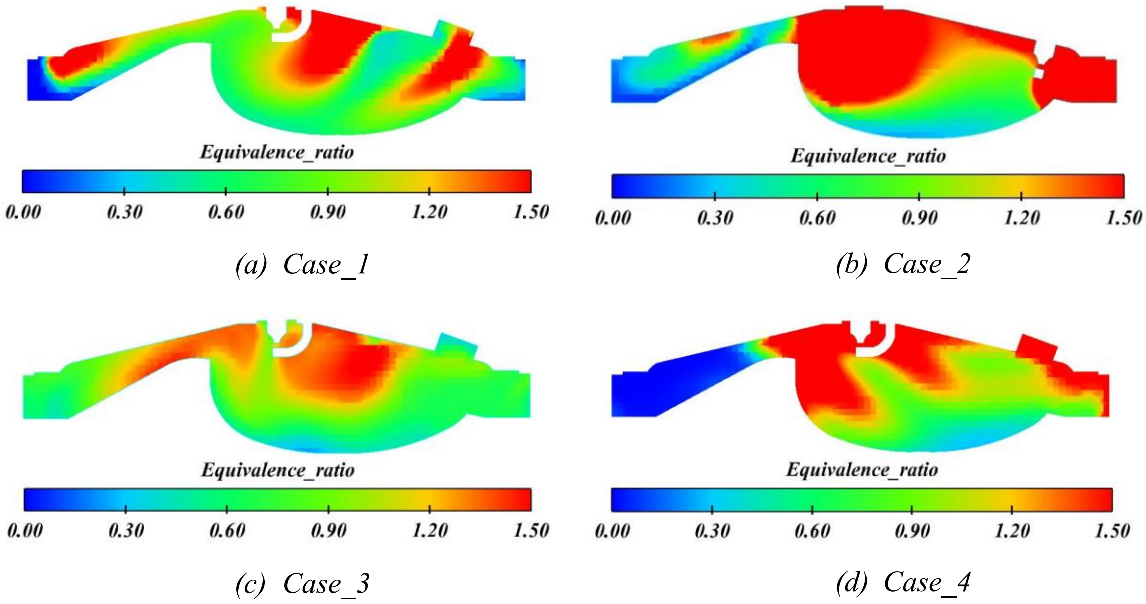


Figure 6. Comparison of ER distribution on the central plane at the spark timing

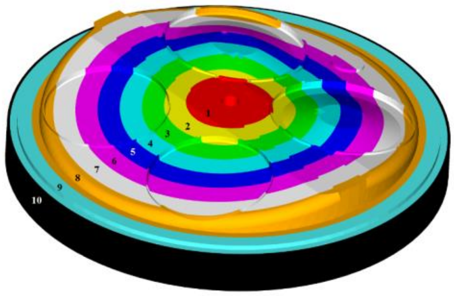


Figure 7. Hemispherical zones in the combustion chamber with spark plug as centre

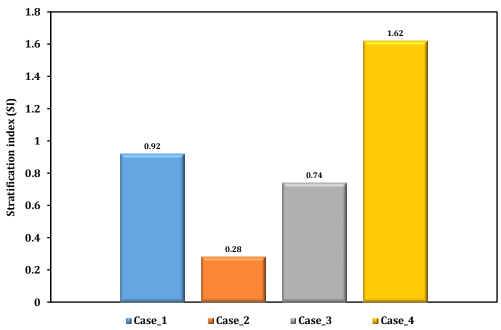


Figure 8. Comparison of SI for various cases of SP and FI location

4.4. Effects of SP and FI Location on Stratification index

Figure 8 shows the comparison of SIs, for various cases considered. From Figure 7, it is seen that, for case 1, 2, 3 and 4, the Sis are about 0.92, 0.28, 0.74 and 1.62. It indicates that, for the case 1, the mixture is almost ideally stratified. Whereas, for the case 2, the stratification is poor. The case 3 also has fairly stratified mixture, but in the case 4, a very poor stratification is seen because of too rich mixture around the spark plug.

4.5. Effects of SP and FI location on In-cylinder pressure and Temperature

Figure 9 shows the comparison of in-cylinder pressures for various SP and FI locations. From Figure 8, it is found that, for the case 1, the peak in-cylinder pressure is about 39 bar, which occurs at about 740 CAD. Whereas, for the cases 2, 3 and 4, the peak in-cylinder pressures are lower by about 32, 24.3 and 27.4% respectively, compared to that of the case 1. However, it is also observed that, after 800 CAD, all the in-cylinder pressure traces follow the same trend.

Figure 10 shows the comparison of in-cylinder temperatures for various cases of SP and FI locations. From Figure 10, it is also found that, for the case 1, the peak in-cylinder temperature is about 2078 K. Whereas, for the cases 2, 3 and 4, the peak in-cylinder temperatures are lower by about 14.3, 13.7 and 22.1% respectively, compared to that of the case 1.

Hence, it is observed that, the in-cylinder pressure and temperature are following the same trend as observed in Figures 6 and 8. The FI location in the case 1 is in such a way that, the interaction between intake air and fuel injected happens in a better way. This in turn results in higher in-cylinder pressure and temperature.

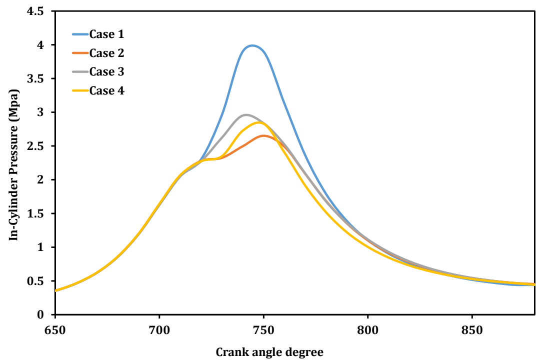


Figure 9. Comparison of in-cylinder pressure for various cases of SP and FI location

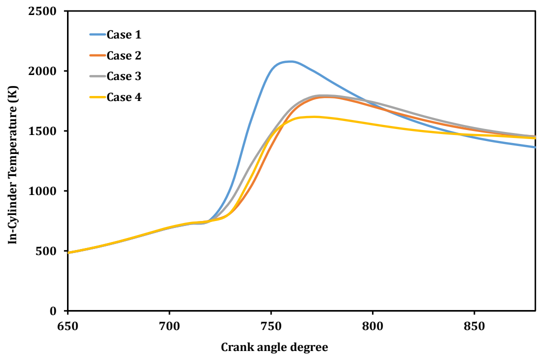


Figure 10. Comparison of in-cylinder temperature for various cases of SP and FI location

4.6. Effects of SP and FI location on Heat release rate and Mass of fuel burnt

Figure 11 shows the comparison of HRR for various cases. From Figure 11, it is seen that, for the case 1, the peak heat release rate is about 34.3 J/CAD. Whereas, for the cases 2, 3 and 4, the peak HRR is lesser by about 38.8, 47.8 and 30.6% respectively, compared to that of the case 1. This is because of better mixture preparation in the case 1 which leads to faster combustion.

Figure 12 shows the variation of the percentage of fuel burnt with crank angle for the various cases. From Figure 12, it is seen that, for the case 1, the center of combustion (COC or CA50) is at about 740 CAD, which indicates a faster and improved combustion. Whereas, for the cases 2, 3 and 4, the COC's are at about 753, 751 and 750 CADs which are quite closer to each other. From Figure 11, it is also seen that, for the case 1, 90% fuel is burned at about 759 CAD. Whereas, for the case 2, 3 and 4, 90% of the fuel is burned at about 791, 790 and 817 CAD respectively. Hence, it is seen that, the faster combustion takes place with the case 1, compared to that of the other cases. This is because of better ER distribution in the combustion chamber for the case 1 as mentioned in Section 4.2. From Figure 12, it is also observed that, the slower combustion takes place with the case 4, because of too rich mixture near the spark plug due to which the spark gets quenched and then it becomes difficult for flame propagation.

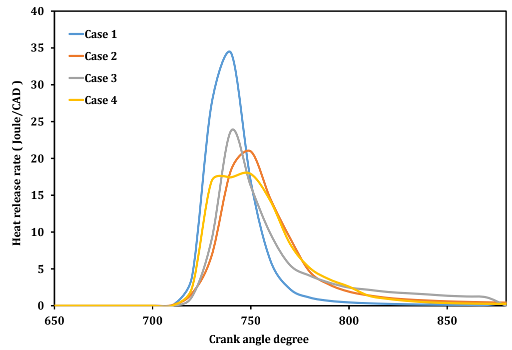


Figure 11. Comparison of heat release rate for various cases of SP and FI location

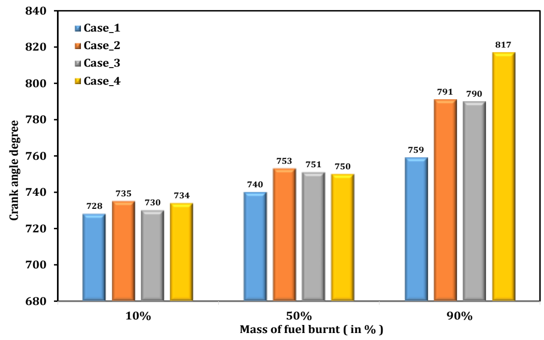


Figure 12. Comparison of mass fuel burnt for various cases of SP and FI location

4.7. Effects of SP and FI location on Indicated thermal efficiency (ITE)

Figure 13 shows the comparison of indicated thermal efficiency for the various cases. From Figure 13, it is seen that, for the case 1, the ITE is about 36.1%. Whereas, for the cases 2, 3 and 4, it is lower by 9.42, 4.8 and 8.2% than that of the case 1. This is because of the higher IMEP and power output, which is obtained by faster combustion in the case 1.

4.8. Effects of SP and FI location on NO_X emission

Figure 14 shows the comparison of NOx emissions at the EVO (873 CAD) for the various cases. From Figure 14, it is seen that, with the faster and enhanced combustion, the NOx emissions increase. Also it is found that, for the case 1, the NOx emissions are about 1 mg/cycle. For the cases 2, 3 and 4, they are lesser by about 63, 52 and 74% respectively, compared to that of the case 1.

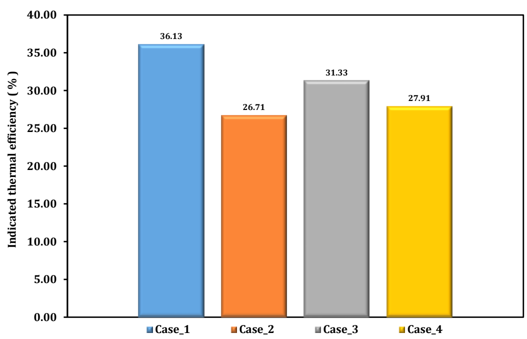


Figure 13. Comparison of indicated thermal efficiency for various cases of SP and FI location

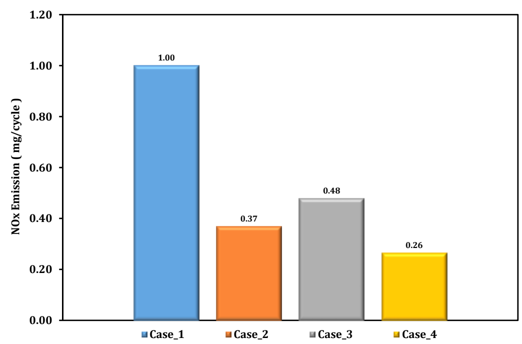


Figure 14. Comparison of NOx emission at EVO for various cases of SP and FI location

4.9. Effects of SP and FI location on Soot and CO emission

Figure 15 shows the comparison of soot emissions at the EVO (at 873 CAD), for various cases considered. From Figure 15, it is seen that, with the faster and enhanced combustion, the soot emissions decrease. Also it is found that, for the case 1, the soot emissions are about 0.017 mg/cycle. For the cases 2, 3 and 4, they are higher by about 123.3, 23.5 and 100% respectively, compared to that of the case 1.

Figure 16 shows the comparison of CO emissions at the EVO, for the various cases considered. From Figure 16, it is seen that, with the faster and enhanced combustion, the CO emissions decrease. This is because, with the better combustion, more carbon will be converted into CO_2. Also, it is found that, for the case 1, the CO emissions are about 1.66 mg/cycle. For the cases 2, 3 and 4, they are higher by about 133.3, 24.7 and 196.3% respectively, compared to that of the case 1. Hence, the case 1 shows the least soot and CO emissions as compared to that of the other cases.

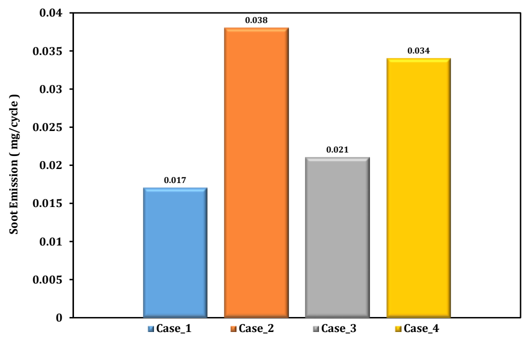


Figure 15. Comparison of soot emission at EVO for various cases of SP and FI location

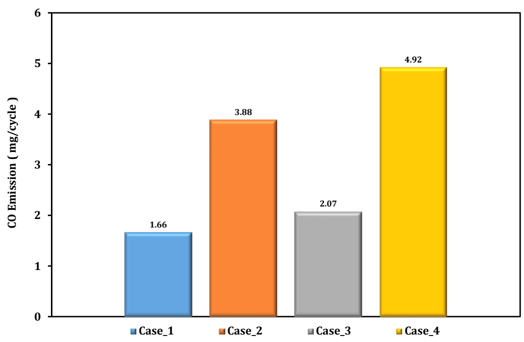


Figure 16. Comparison of CO emission at EVO for various cases of SP and FI location

5. Conclusions

In this study, a CFD simulation study is conducted on a four-stroke four-valve GDI engine to evaluate the effect of SP and FI locations on the mixture stratification, combustion and emission characteristics of the engine. From the analysis of results, the following conclusions are drawn:

- The mixture stratification is found best for the case1 followed by the cases 3, 2 and 4.
- The case 1 shows the higher in-cylinder pressure and temperature, heat release rate and faster burning compared to that of the other cases.
- The case 1 gives the ITE of about 36.1%, whereas for the cases 2, 3 and 4, it is lower by about 9.42, 4.8 and 8.2%.
- The NOx emissions are the highest for the case 1 followed by the cases 3, 2 and 4. The soot and CO emissions are the least for the case 1.

Finally, it is concluded that, the case 1 results in the better mixture stratification, which leads to better combustion and performance. However, the NOx emissions are higher, but the soot and CO emissions are least. Also, in this study, the extent to which location of SP and FI can be changed is limited by the space available in the engine head and other geometrical parameters.

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IOP Conf. Series: Materials Science and Engineering **243** (2017) 012025 doi:10.1088/1757-899X/243/1/012025

Abbreviations

SP	Spark Plug
FI	Fuel Injector
TDC	Top dead center
ER	Equivalence ratio
HRR	Heat release rate
ITE	Indicated thermal efficiency
GDI	Gasoline direct injection
CFD	Computational fluid dynamics
CA50	The crank angle at which 50% of the fuel has burnt
COC	Center of combustion
RNG	Re-normalized group
KH-RT	Kelvin-Helmholtz-Rayleigh-Taylor
PISO	Pressure implicit with splitting of operators
CFL	Courant Fredrick and Lewy Number