On the Effect of GDI Injector Configuration on Charge Preparation

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Abstract

A Gasoline Direct Injection (GDI) engine typically operates on multiple fuel-preparation modes. In general, at higher loads a homogeneous mixture is favoured whereas a stratified mixture is preferred at part and low load conditions. This is usually achieved by altering the injection timing with respect to load and speed. In this paper the effect of injector configuration on the mixing process has been studied systematically. Two different injector configurations are considered, one with a central-hole injection and other with a 6-hole injection. The objective is to investigate the effect of initial fuel distribution inside the engine cylinder on charge preparation at the onset of ignition. This study also aims to explore a better solution for mixing in GDI engines by optimizing the GDI injector for both stratified and homogeneous mode of operations. An engine with a pentroof combustion chamber with centrally mounted injector and upright straight intake port and flat piston is selected. The computation begins from the start of the induction process and continued till the point of ignition. The dynamics of the mixing process is studied by grouping the in-cylinder charge in different bins in terms of the equivalence ratio. The temporal variation of the fraction of the mixture in different bins is studied as a function of time to understand the dynamics of the mixing process. Results from the parametric study indicate the possibility of switching the modes of mixing with respect to the operating conditions.

Introduction

GDI engine technology is a promising alternative to port fuel injection because of the potential gains in fuel economy, power density and transient response. However, to realize the key benefits from this technology like high fuel efficiency at part-load conditions it is important to enable the engine to run without throttling of the air supply. This requirement has generally divided the operation of a GDI engine in two modes namely the homogeneous mode and the stratified mode. For a given engine the region for the homogeneous and stratified mode of operations can be precisely located in the torque-speed parameter space [1]. The overall mixture quality for the homogeneous mode is stoichiometric whereas that for the stratified mode is extremely lean. Under lean conditions the only way to assure stable operation of the engine is to operate in a stratified mode by ensuring a near-stoichiometric mixture around the spark plug during the onset of ignition. The challenge is to switch the mode from homogeneous to stratified using the same injection system. This is generally attempted by varying the injection timing, the injector and the piston geometry. Different design configurations have been introduced from the spray, air and engine cylinder point of views, which can be classified as wall-guided combustion system [2,3], air-guided combustion system [4] and spray guided combustion system [5]. Injector design has also evolved over a period of time starting from the injectors similar to Diesel engines [6,7] and then to hollow-cone swirl injectors [7,8]. The recent trend [9] is again to go for multi-hole diesel-like injectors where unlike the swirl injector the spray does not collapse at high surrounding pressure in case of late injection during operation in the stratified mode. In this work, straight-hole injectors are investigated in a realistic GDI engine geometry. Two particular configurations are studied. In the first case, six holes are placed symmetrically around the centerline of the injector and in the second case a single hole is placed along the centerline of the injector. The cross-sectional areas of the single hole in the second case is kept same as the sum of the cross sectional area of all the six holes in the first case. The suction and the compression process till the onset of ignition are simulated in STAR-CD to study the differences in mixing process and the charge quality at the ignition point for these two different injectors. The objective is to study the effect of the initial distribution of the spray (as obtained in the six hole symmetrically distributed and the single hole injectors) on the mixing and mixture quality at the onset of ignition.

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Methodology

The analysis is carried out using a commercial code STAR-CD [10] which can solve the three – dimensional compressible Navier – Stokes equations with moving boundaries. RNG k-ε turbulence model is used for the simulation. A Lagrangian-Eulerian approach is used for the modeling of the spray. The primary and secondary breakup is modeled using the Reitz-Diwakar model [11] and the spray impingement and film formation was modeled using the Bai model [12]. The engine specifications as used for the simulation are given in Table 1 and the injector details are given in Table 2. Since the analysis is carried out till the point of ignition, exhaust ports are not modeled. Meshing is done in ES-ICE [13]. The final engine model and the meshed geometry are shown in Fig. 1. The cylinder domain was initialized with a temperature of 500 K and a pressure of 1.1 bar and for the ports with 350 K and 1 bar respectively. The cylinder wall, piston crown and combustion dome are maintained at temperatures of 550 K, 600 K and 600 K respectively. The most critical component of the simulation is the spray model because the constants used in the model are not universal and need to be fine-tuned with respect to the injectors and operating conditions. The constants C_{b1} (empirical coefficient of Weber number for Bag break-up model) and C_{s2} (empirical coefficient in time scale for stripping break-up model) in the Reitz-Diwakar model are fine tuned with respect to the experimental data of Mitroglou et al. [9]. The injectors used in this reference [9] are similar type of injectors and the values of the constants are fine-tuned for an injection pressure of 120 bar and for a chamber pressure of 1 bar. These values are found to be $C_{b1} = 6$, $C_{s2} = 2$. For these values of the constants a good match was found for the droplet size and droplet velocity at different stations from the injector tip. The same injection pressure of 120 bar is used in the simulation to minimize the error from the spray model. Standard models in STAR-CD are used to calculate the heat and mass transfer between the gas phase and the spray. Isooctane was used as a surrogate of gasoline in the simulation as in the experiments with respect to which the model was validated.

The calculation was started at TDC in the suction stroke and continued till 20 degree before TDC during the compression stroke. The main objective of this study is to compare the 6-hole and the central-hole injector for different start of injection at different overall air-fuel ratios. Table 3 shows the details of the cases that are simulated.

Results and Discussion

The dynamics of the mixing process and the state of the charge at the onset of ignition is studied for the different cases as explained in the last section. It is important for this purpose to asses the mixing process for the different cases. The approach adopted in this paper to characterize the charge quality is to classify the entire charge into three bins namely the rich mixture (phi>1.1), near-stoichiometric mixture (0.9<phi<1.1) and the lean mixture (phi<0.9) bins. The variation of the mass percentage of the charge in each of these bins is studied. Figure 2 and Figure 3 show the variation of the mass percentage of the charge in three bins for case-4 and case-3 respectively. As can be seen from these figures, initially the entire mixture is lean and gradually gets converted to rich and stoichiometric mixtures. For case-4 the quantity of rich mixture also starts reducing and the entire mixture moves towards the near-stoichiometric conditions. This is because the overall mixture is stoichiometric (phi =1). Case-4 shows a high percentage of near-stoichiometric mixture at the onset of ignition, which is desirable for the homogeneous mode.

Looking at the flow structure it is clear that the straight intake port results in a tumble as shown in Fig. 4, which in turn helps in mixing. The tumble has a considerable effect on the spray structure as could be seen in Fig. 5. The spray plume aligned with the tumble shows more penetration and deflection along the tumble direction.

To compare the results of different cases in detail only certain relevant parameters at the onset of ignition are selected. These are the mass percentage of the near-stoichiometric charge at the onset of ignition, the percentage mass of fuel vapour originating from the fuel film formed due to spray impingement and the mass of liquid fuel left (in the form of droplet and film). Figure 6 shows the variation of mass percentage of near-stoichiometric mixture at the ignition point with respect to start of injection (SOI). As expected the amount of stoichiometric charge increases with early injection for both 6-hole and central-hole injector. However, the amount is more for the six-hole than for the central hole for early injection timings. Interestingly, although the amount increases constantly and almost linearly for the central hole but for the 6-hole there is a flat region in between. The reason for this will be clear from Fig. 7, which shows the variation of the percentage of vapour originating from the film. This clearly shows that for a late injection the impingement is very less for the 6-hole as compared to the central-hole. As the SOI is advanced to 200^{O} from 180^{O} the impingement increases substantially for the six-hole. This is because of the fact that the spray is injected into a low temperature environment for an early injection. Hence, although the early injection allows for more residence time the increase in the impingement and subsequent film formation reduces the rate of evaporation of the fuel. For the central hole the impingement does not increase substantially with early injection and hence the amount of near-stoichiometric charge increases constantly with early injection. Figure 8 shows the percentage of liquid fuel left at the start of ignition. It is seen that for the six-hole injector no liquid fuel is left either in the form of droplet or film whereas for the central injection even for an injection as early as 300^{O} before TDC some amount of liquid fuel is left back as film. Figure 9 shows the variation of near stoichiometric charge at the onset of ignition with SOI for an overall equivalence ratio of 0.35. For this lean condition it is important to ensure that some stoichiometric mixture is left at the start of ignition. It is seen from Fig. 9 that the central-hole injector ensures higher amount of charge at stoichiometric condition for a late injection of 80^{O} before TDC. However, it was also observed that around 6-7% of the injected fuel was at a liquid state (as film and droplet) at the onset of ignition for 80^{O} SOI with the central injection. The amount of the charge at a near-stoichiometric condition was also quite low. Hence, it is important that this charge remains at a location near the spark plug and is not distributed all around inside the combustion chamber. Figure 10 shows the distribution of the charge in the engine cylinder in the central plane. It is visible from this that near-stoichiometric charge is available on one side of the combustion chamber at the point of ignition. In general, it can be concluded from the results that the six-hole injector with an early injection is more appropriate for the homogeneous mode of operation whereas the central-hole injection is relatively better for the stratified mode with late injection. Thus different injector configurations and injection parameters are needed based on the operating conditions. However, there is a problem with liquid fuel remaining at the onset of ignition for the late injection with the central-hole injector.

Conclusions

This study gives certain important information regarding the effect of different nozzle configurations for different SOI in a GDI Engine. It is seen from the results that the fuel impingement generally increases with an early SOI. This is more significant for the 6-hole injector. For a homogeneous mode of operation the six-hole injector with an early start of injection gives a better homogeneity of the mixture. For low overall equivalence ratio a central injection seems to be preferable with late injection timing to ensure a stratified charge with a near-stoichiometric region around the spark gap.

Nomenclature

ϕ equivalence ratio
k turbulent kinetic energy
ε dissipation of turbulent kinetic energy

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Table 1. Engine Specifications

Parameter	Value
Engine type	4- stroke 4-valve GDI
Bore (mm)	89
Stroke (mm)	86.6
Compression ratio	10.5
Swept Volume (cm^3)	538.5
Engine speed (rpm)	2000
Piston	Flat

Intake Ports	Upright Straight
Air Fuel ratio	14.9 for homogeneous mode and 42.7 for stratified mode.
Spark plug	Centrally located
Injector	Centrally located

Table 2. Injector Specifications

Parameter	6-hole	Central hole
Fuel Injection pressure	120 bar	120 bar
Hole diameter	140 micron	343 micron
Nozzle L/D	2.1	2.1
Coefficient of Discharge, Cd	0.7	0.7
Cone angle (for each spray plume)	16^{O}	16^{O}
Angle between spray plume axis and central axis	40^{O}	-

Table 3. Cases simulated

Case	Injector	SOI (degree BTDC)	Overall equivalence ratio
1	6-hole	80	1.0
2	6-hole	120	1.0
3	6-hole	200	1.0
4	6-hole	300	1.0
5	central-hole	120	1.0
6	central-hole	200	1.0
7	central-hole	300	1.0
8	6-hole	80	0.35
9	6-hole	120	0.35
10	6-hole	180	0.35
11	central-hole	80	0.35
12	central-hole	120	0.35

Figure 1. The geometry and mesh of the computational domain.

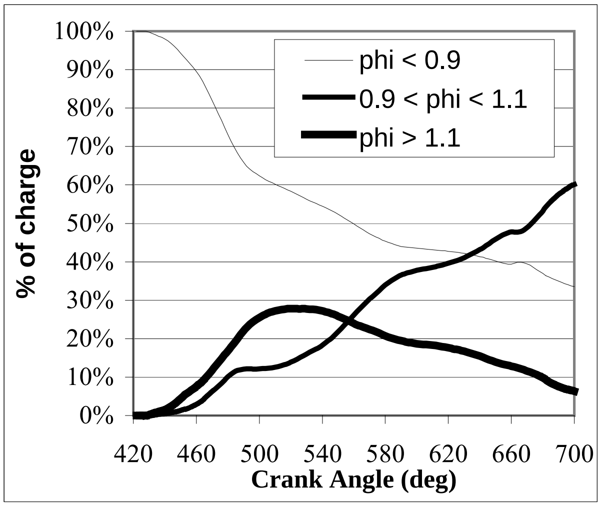

Figure 2. Mass percentage of Charge for 300^{O} SOI – 6hole (Case-4).

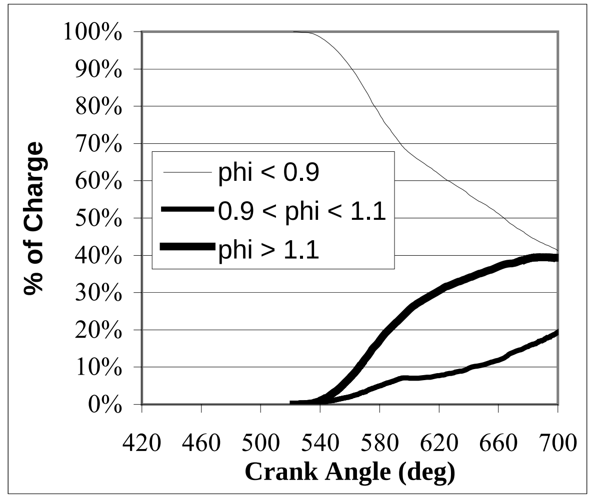

Figure 3. Mass percentage of Charge for 200^{O} SOI – 6 hole (Case-3).

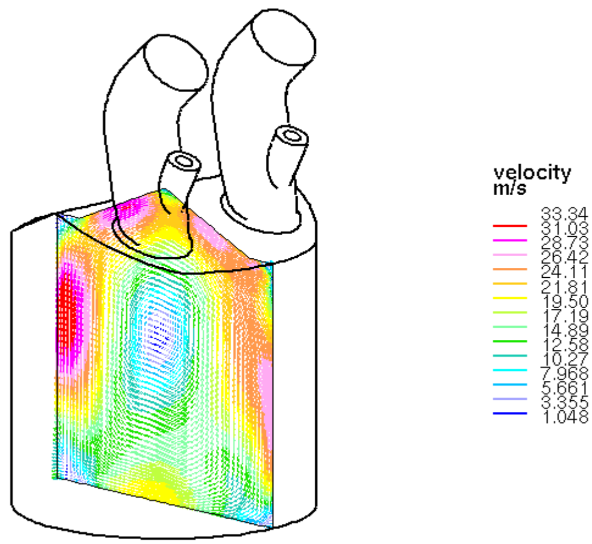

Figure 4. Velocity plot showing tumble.

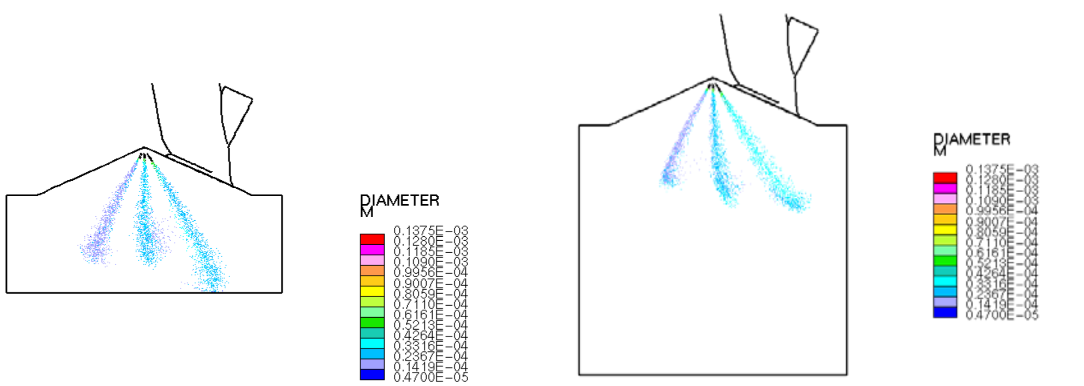

Figure 5. Effect of tumble in mixing for 300^{O} (left, Case-4) and 200^{O} (right, Case-3) SOI (BTDC), 6 hole, $\phi = 1$. Time is 0.67 ms after SOI.

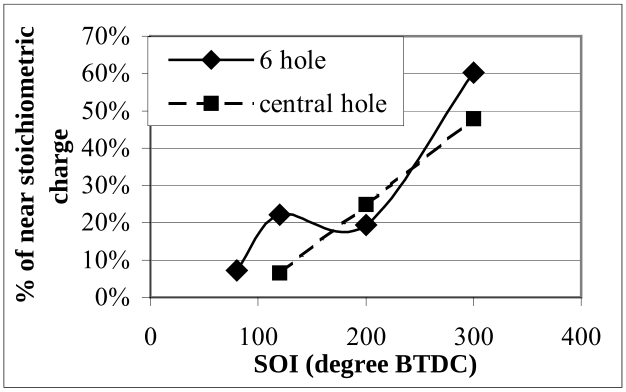

Figure 6. Variation of the percentage (mass) of near-stoichiometric charge (1>phi>0.9) with different start of injection (SOI) at the onset of ignition. The overall equivalence ratio is phi = 1 and the engine speed is 2000 rpm.

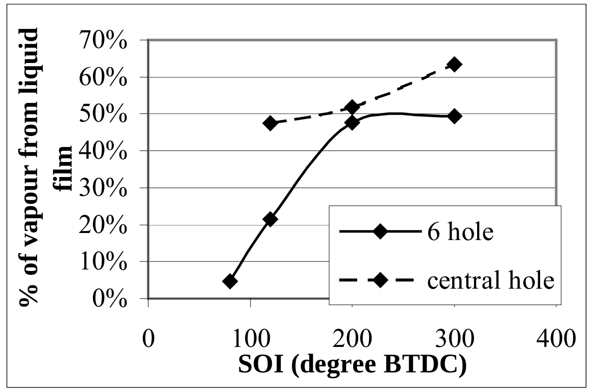

Figure 7. Variation of the percentage (mass) of fuel vapour from liquid film with different start of injection (SOI) at the onset of ignition. The overall equivalence ratio is phi = 1 and the engine speed is 2000 rpm.

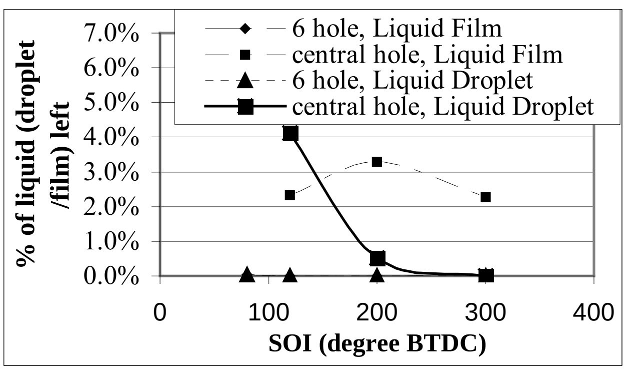

Figure 8. Percentage of liquid droplet and liquid film left at the onset of ignition for different start of injection. The overall equivalence ratio is phi = 1 and the engine speed is 2000 rpm.

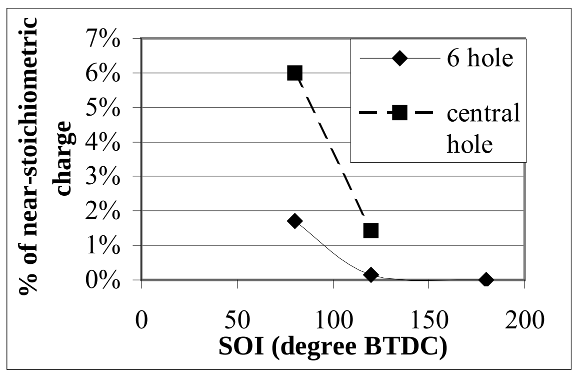

Figure 9. Variation of the percentage (mass) of near-stoichiometric charge (1>phi>0.9) with different start of injection (SOI) at the onset of ignition. The overall equivalence ratio is phi = 0.35 and the engine speed is 2000 rpm.

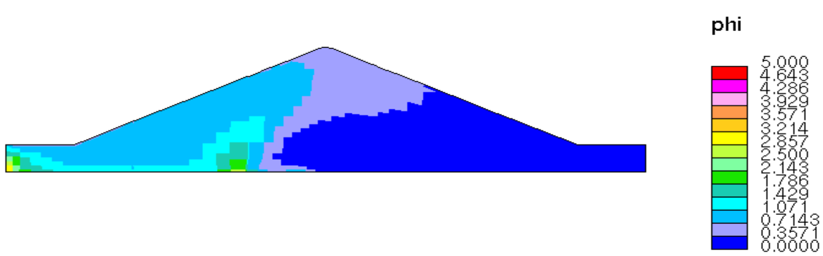

Figure 10. ϕ distribution for 80^{O} SOI Central hole - phi = 0.35