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Effect of piston profile on performance and emission characteristics of a GDI engine with split injection strategy – A CFD study

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Abstract. Gasoline direct injection (GDI) engines have gained popularity in the recent times because of lower fuel consumption and exhaust emissions. But in these engines, the mixture preparation plays an important role which affects combustion, performance and emission characteristics. The mixture preparation in turn depends mainly upon combustion chamber geometry. Therefore, in this study, an attempt has been made to understand the effect of piston profile on the performance and emission characteristics in a GDI engine. The analysis is carried out on a four-stroke wall guided GDI engine using computational fluid dynamics (CFD). The spray breakup model used is validated with the available experimental results from the literature to the extent possible. The analysis is carried out for four piston profiles viz., offset pentroof with offset bowl (OPOB), flat piston with offset bowl (FPOB), offset pentroof with offset scoop (OPOS) and inclined piston with offset bowl (IPOB) fitted in an engine equipped with a six-hole injector with the split injection ratio of 30:70. All the CFD simulations are carried out at the engine speed of 2000 rev/min., with the overall equivalence ratio of about 0.65±0.05. The performance and emission characteristics of the engine are compared while using the above piston profiles. It is found that, the OPOB piston is preferred compared to that of the other pistons because it has better in-cylinder flow, IMEP and lower HC emissions compared to that of other pistons.

1. Introduction

Developing engines with higher fuel economy and lower exhaust emissions have been a key research area in recent times [1]. With poor part-load fuel efficiency and higher emissions, port fuel injection (PFI) engines are facing difficulty in fulfilling the emission norms. On the other hand, GDI engines, because of lean mode of operation and mixture stratification, are capable of achieving better fuel economy and lower emissions. In GDI engines, fuel is injected directly into the engine cylinder which helps increasing volumetric efficiency and compression ratio. In a wall guided GDI engine, piston is profiled in such a way that the fuel-air mixture is directed towards the spark plug to form a combustible mixture at the time of ignition. Piston with a bowl gives better air-fuel mixing, higher tumble ratio and turbulent kinetic energy compared to that of the flat piston [1]. It is also beneficial in maintaining in-cylinder turbulent intensity and promotes fuel evaporation and reduces soot emissions [2]. Also, piston with a smoother top surface accelerates combustion and increases in-cylinder temperature and pressure [3]. Stoichiometric mixture around the spark plug results in higher heat release and NO_x emissions, compared to that of the lean mixture [4]. With split injection ratio of

30:70, mixture is slightly homogenous which results in better ignitable mixture around the spark plug which will lead to a better combustion [5].

In the present study, the effect of piston top profile on the performance and emission characteristics of a four-valve, wall guided GDI engine is studied. The engine is operated in the split injection mode with the split injection ratio of 30:70. Four piston top profiles are considered for the analysis.

2. CFD Methodology

2.1. Engine Geometry and Meshing

The CFD analysis is carried out on a four-stroke, four-valve, wall guided GDI engine whose specifications are taken from Costa et al., [6] and are shown in Table 1. Figure 1 shows the computational model of the engine. Table 2 gives the details of fuel injection parameters. The grid embedding for this study is done as per details given in *Krishna and* Mallikarjuna [7], which are not shown here.

2.2. Piston Profile

In this study, four piston profiles are considered viz., offset piston with offset bowl (OPOB), flat piston with offset bowl (FPOB), offset piston with offset scoop (OPOS) and inclined piston with offset bowl (IPOB). Figure 2 shows various piston profiles used.

Figure 1. Computational model of the engine

Table 1. Detailed specifications of the engine

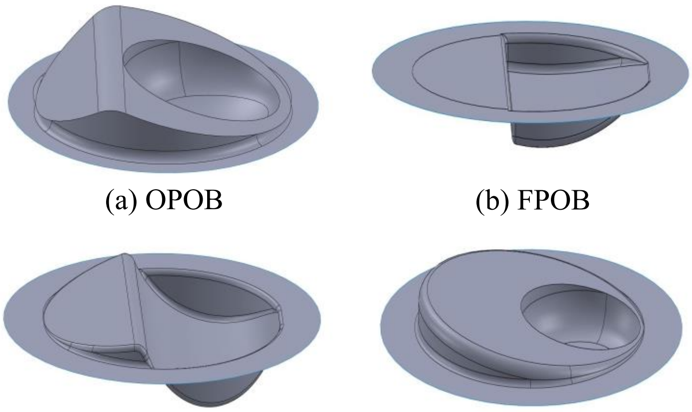
Parameters	Specification
Displacement volume	398.5 cm3
Stroke	81.3 mm
Bore	79 mm
Connecting rod	143 mm
Compression ratio	10.6:1
Speed (rpm)	2000
Number of valves	4
Exhaust valve opening	27º before BDC
Exhaust valve closure	0º after TDC
Inlet valve closure	36º after BDC
Inlet valve opening	3º before TDC

(*360 CAD is suction at TDC)

Table 2. Details of fuel injection parameters

No. of holes	Six
Fuel	Gasoline
Start of Pilot Fuel Injection	210^0 BTDC
Start of Main Fuel Injection	90^0 BTDC
Pilot fuel injection duration	6.9 CAD
Main fuel injection duration	16.2 CAD
Fuel injection pressure	120 bar
Spray cone angle	15 degree
Mass of fuel injected per	22 mg

*CAD- Crank Angle Degrees



(a) OPOB (b) FPOB

(c) OPOS (d) IPOB

Figure 2. Various piston profiles used in the study

2.3. CFD Modeling

The governing equations used, in the study, to generalize the in-cylinder flows are given below [8].
Conservation of mass:

$$\frac{\partial \rho}{\partial t}+\frac{\partial(\rho u_j)}{\partial x_j}=s_m \tag{1}$$

General transport equation:

$$\frac{\partial(\rho\varphi)}{\partial t}+div(\rho\varphi u)=div(\Gamma grad\,\varphi)+S_\varphi \tag{2}$$

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.
The CFD simulation is carried out using the CONVERGE. Mass, momentum and energy are calculated at each node of the grid using unsteady Navier-Stokes equations which is implicitly discretized based on finite volume method over the Cartesian grid [8]. RNG k-ε turbulence model is used to analyze the turbulence characteristics [9]. The KH-RT model is used to capture fuel spray break-up [7]. The pressure implicit for splitting of operator (PISO) algorithm is used to solve the pressure-velocity coupling [10]. The O'Rourke and Amsden model is used for wall heat transfer calculations [11]. For the analysis of droplet evaporation, the FROSSLING model is used [12]. For the analysis of film splash and fuel turbulent dispersion, the O'Rourke model is used [12]. The droplet

collision has been analyzed by the NTC collision model [12] and initial droplet distribution is analyzed using the blob injection model [12]. The inlet and exhaust boundaries are maintained at atmospheric conditions [11,14]. In order to obtain the initial conditions in various regions of the computational domain, full-cycle simulations are carried out successively four times. The flow characteristics, at the end of the fourth cycle, are used as the initial conditions for all the simulations. The constants used for the KH-RT spray model and k-ε turbulence model are taken from the literature [13]. In this study, all the CFD simulations are carried out from the inlet valve opening (IVO) to the exhaust valve opening (EVO) at the engine speed of 2000 rev/min., with the overall equivalence ratio of 0.65±0.05. The time step varies between 10^{-8} and 10^{-5} seconds depending on the grid CFL criteria. The pilot injection is maintained at 510 crank angle degree (CAD), the main injection is maintained at 630 CAD and the spark timing is maintained at 710 CAD. The convergence tolerance for momentum and pressure are taken as 10^{-5} and 10^{-8} respectively. Whereas, for the density, the energy and the species, the convergence tolerance is taken as 10^{-4}.

2.4. CFD Model Validation

The CFD models have already been validated by Krishna et al., [12] in their work. Validation was done based on the experimental results obtained from Costa et al., [6] on the same engine. Comparison of in-cylinder pressure with crank angles obtained from both the CFD and experimental results are shown in Figure 3. The comparison of ER distribution at the time of ignition for both CFD and experimental results are shown in Figure 4. From Figures 3 and 4, it is evident that; the two results are in good agreement. Therefore, the selected CFD models are used for further analysis.

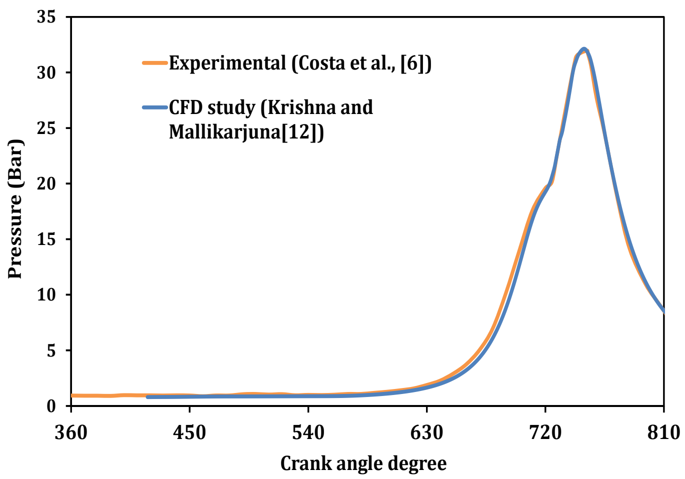
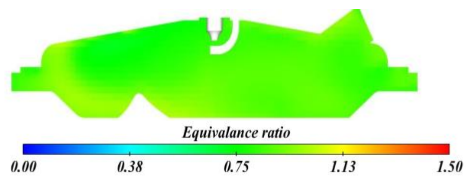
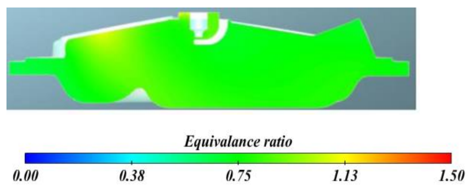


Figure 3. Comparison of in-cylinder pressure with crank angle degree



(a) From CFD analysis of Krishna and Mallikarjuna [6]



(b) From CFD analysis of Costa et al., [16]

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

3. Results and Discussions

3.1. Effect of Piston Profile on In-cylinder flow

Figure 5 shows the comparison of in-cylinder flows for various piston profiles at various CADs. From figure 5, it is seen that, at 640 CAD, the piston profiles considered have minimum impact on in-the cylinder flows. Whereas, at 675 CAD, for the OPOB and the FPOB pistons, a strong vortex occurs near the spark plug which leads to better mixture stratification compared to that of the other two pistons. At the end of the compression stroke (700 CAD), all piston profiles show vortices in the combustion chamber. But, for the OPOB piston, the intensity of the vortex is observed to be higher than that of the other pistons which in turn help enhanced combustion and performance.

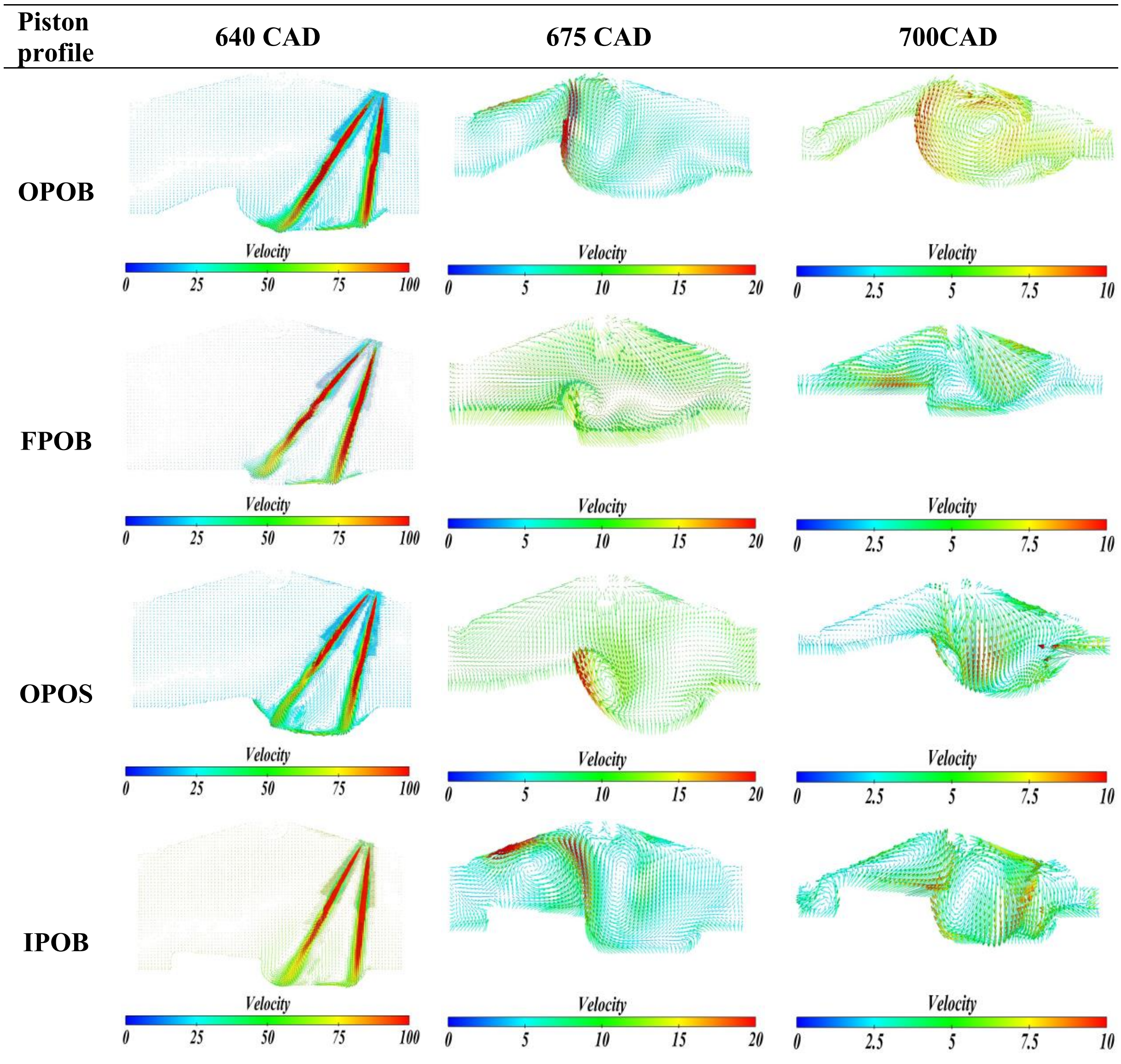


Figure 5. Comparison of in-cylinder flows for various piston profiles at various CAD

3.2. Effect of Piston Profile on Equivalence Ratio(ER) Distribution

Figure 6 shows the ER distribution for various piston profiles. From figure 6, it is seen that, the ER near the spark plug location for OPOB, FPOB, OPOS and IPOB piston profiles are about 1.12, 0.85, 0.6 and 1.45 respectively. For OPOB and FPOB, mixture is almost stoichiometric near the spark plug location because of better in-cylinder flow. Whereas for OPOS the mixture is very lean and for IPOB, the mixture is very rich near the spark plug location.

3.3. Effect of Piston Profile on In-cylinder Pressure

Figure 7 shows the comparison of in-cylinder pressure for various piston profiles. From figure 7, it is seen that, for the OPOB piston profile, the peak in-cylinder pressure is about 48 bar which occurs at about 740 CAD. Whereas, for the FPOB, OPOS and IPOB pistons, the peak in-cylinder pressures occur at about 740 CAD, which are lower by about 27, 38 and 27% compared to that of the OPOB piston. Since, the ER is nearer to stoichiometric around the spark plug in the case of OPOB piston, the in-cylinder pressure is higher compared to the other pistons considered.

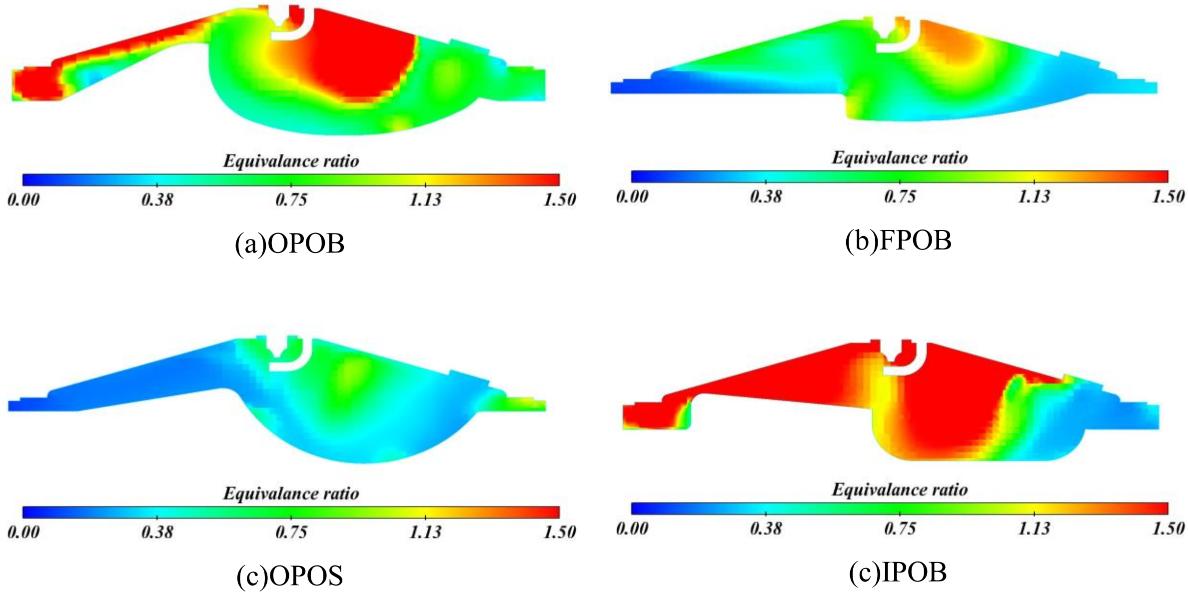


Figure 6. Comparison of ER for various piston profiles

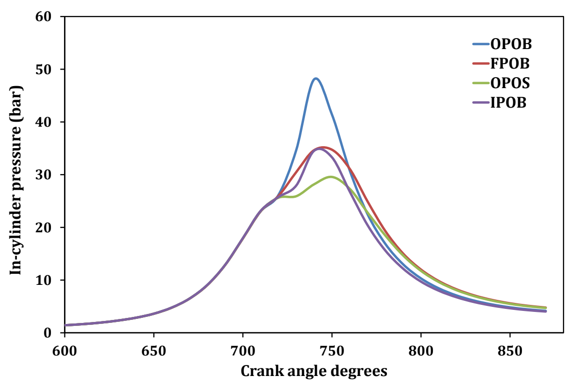


Figure 7. Comparison of in-cylinder pressure for various piston profiles

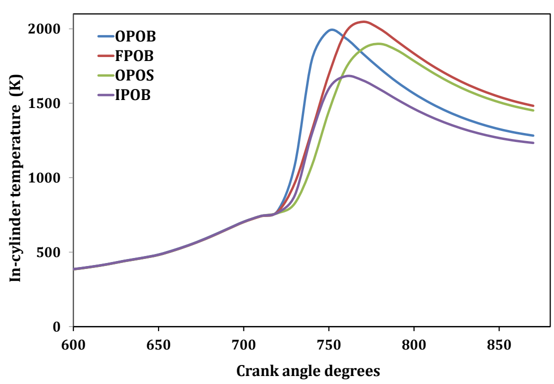


Figure 8. Comparison of in-cylinder temperature for various piston profiles

3.4. Effect of Piston Profile on In-cylinder Temperature

Figure 8 shows the comparison of in-cylinder temperature for various piston profiles. From figure 8, it is seen that, for the OPOB piston, the peak in-cylinder temperature is about 1987 K which occurs at about 750 CAD. Whereas, for the FPOB, the peak in-cylinder temperature is about 3% higher compared to that of OPOB piston. But, for the OPOS and the IPOB, the peak in-cylinder temperatures are about 5 and 15% lower compared to that of the OPOB piston. It means that, the combustion is slower in the FPOB compared to that of the OPOB, as the peak temperature is occurring closer to the TDC in the OPOB. However, because of slower combustion in the FPOB, more fuel is burnt away from the TDC, which gives higher in-cylinder temperature, in the case of the FPOB compared to that of the OPOB piston.

3.5. Effect of Piston Profile on Heat Release Rate

Figure 9 shows the comparison of heat release rate (HRR) for various piston profiles. The HRR is calculated in CONVERGE by using equation 3

$$\frac{dQ}{d\theta}=\frac{\gamma}{\gamma-1}\left(P\frac{dV}{d\theta}\right)+\frac{1}{\gamma-1}\left(V\frac{dP}{d\theta}\right)+\frac{dQ_W}{d\theta} \tag{3}$$

Where, Q is the heat released on combustion of fuel, Q_w is the heat transfer out of the cylinder, P is instantaneous in-cylinder pressure and γ is ratio of specific heat at constant pressure to constant volume.

From figure 9, it is seen that, for the OPOB piston, the peak HRR is about 31 J/CAD. Whereas, for the FPOB, the OPOS and the IPOB pistons, the HRRs are lesser by about 21.2, 28.6 and 8.4% respectively, compared to that of the OPOB piston profile. It is seen that, in the OPOB piston, the in-cylinder flows and tumble ratio are better than that of other pistons. This results in better mixture preparation inside the combustion chamber which leads to faster combustion.

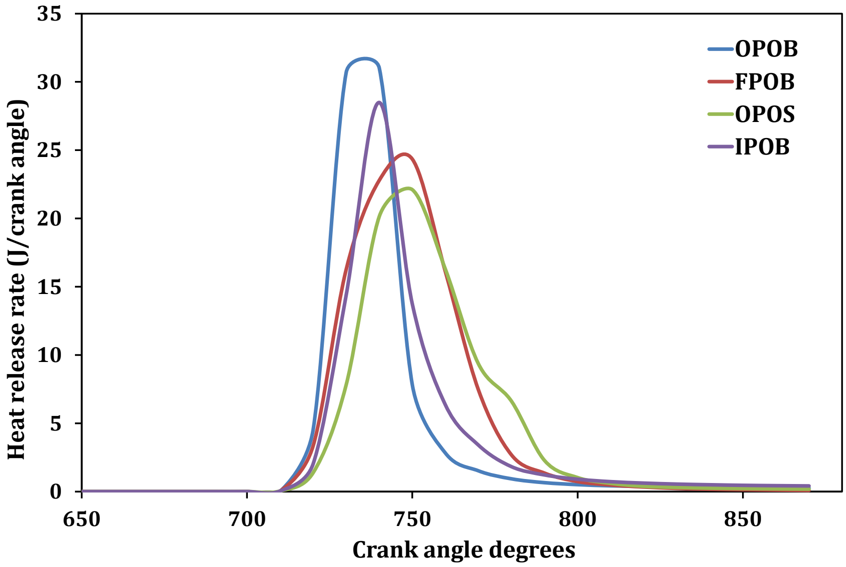


Figure 9 . Comparison of heat release rate for various piston profiles

3.6. Effect of Piston Profile on Percentage Fuel Burnt [PFB]

In this study, the combustion is considered to be completed when 90% of the injected fuel is burnt. Figure 10 shows the comparison of the crank angle position at which 10, 50 and 90% of fuel is burnt. From figure 10, it is seen that, for the OPOB piston, the 90% of the fuel is burned at about 755.7 CAD. However, for the FPOB, the OPOS and the IPOS pistons, the 90% of the fuel is burned at about 769.7, 778.4 and 775.9 CADs respectively. Hence, it is seen that, the fastest combustion takes place with the OPOB piston, compared to that of the other pistons. This is because of better ER distribution in the combustion chamber for the OPOB as mentioned in Section 3.2. In addition, the slowest combustion takes place with the OPOS, because of too lean mixture around the spark plug due to which the flame propagation becomes difficult. For the IPOB piston, the 50% fuel burnt is faster as compared to that of the FPOB piston, which implies that more fuel is available in the case of the IPOB piston at 10-50% PFB phase. However, with the FPOB piston, more fuel is available during 50-90% of the PFB phase. It also resulted in higher HRR with the IPOB piston compared to that of the FPOB piston as seen from figure 9.

3.7. Effect of Piston Profile on IMEP
Figure 11 shows the comparison of IMEP for various piston profiles. From figure 11, it is seen that, for the OPOB piston, the IMEP is about 9.56 bar. With the FPOB, the OPOS and the IPOB pistons, the IMEP is lesser by about 4, 10 and 20% compared to that of the OPOB piston. Higher IMEP, in the case of the OPOB, is mainly because of the better mixture formation at the time of ignition and higher peak in-cylinder pressure.

3.8. Effect of Piston Profile on HC Formation
Figure 12 shows the comparison of HC formation for various piston profiles. From figure 12, it is seen that, the HC formation, at the EVO (at 873 CAD), is negligible for the OPOB, FPOB and OPOS pistons. This is because of better mixture preparation which resulted in almost complete combustion with these pistons. Even though, for the OPOS piston, there is a very lean mixture around the spark plug, at the time of ignition, due to better in-cylinder motion, most of the mixture is burnt at a later stage of the cycle. Therefore, there are very less HC emissions, at the EVO, with the OPOS piston. However, for the IPOB piston, there is a very rich mixture around the spark plug at the time of ignition because of which the combustion becomes slower. Therefore, with the IPOB piston, the HC emissions, at the EVO, are almost doubled compared to that of the OPOB piston.

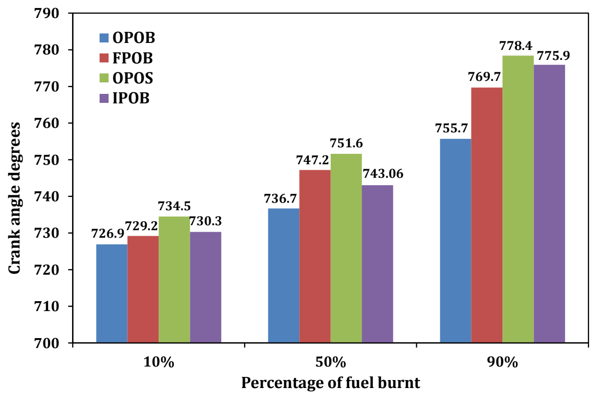


Figure 10. Comparison of percentage of fuel burnt for various piston profiles

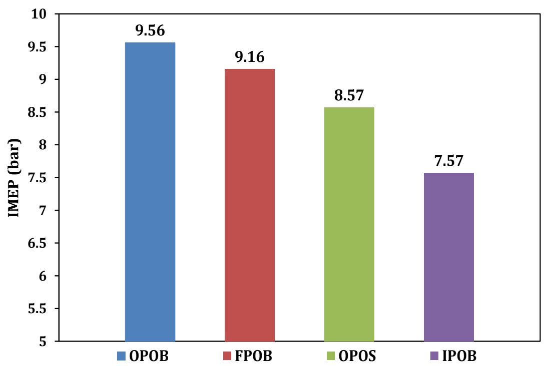


Figure 11. Comparison of IMEP for various piston profiles

3.9. Effect of Piston Profile on NO_X Formation
Figure 13 shows the comparison of NO_X formation for various piston profiles. From figure 13, it is seen that, the NO_X formation at the time of the EVO is the highest for the OPOB piston. Whereas, with the FPOB, OPOS and IPOB pistons, the NO_X emissions are lower by about 15, 50 and 64% compared to that of the OPOB piston. The NO_X emissions, with the OPOB piston, are higher because of better combustion resulting in higher in-cylinder temperature compared to that of the other pistons. Whereas, in the OPOS and IPOB pistons, the mixture at the spark plug location is far from the stoichiometric value. Therefore, there is a poor combustion resulting in lower in-cylinder temperature. Hence, lower NOX emissions are observed in the OPOS and IPOB pistons.

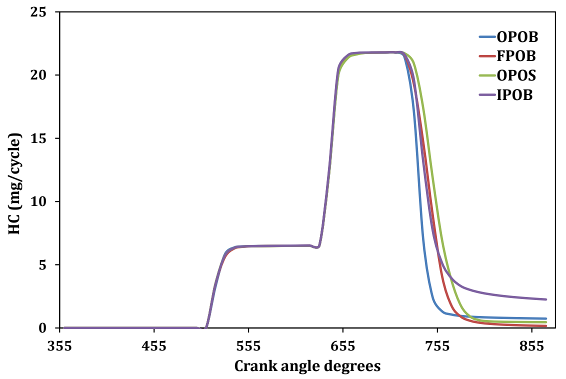


Figure 12. Comparison of HC formation for various piston profiles

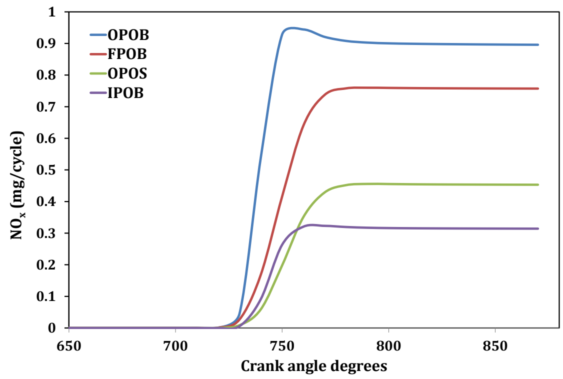


Figure 13. Comparison of NO_X temperature for various piston profiles

4. Conclusion

This study presents the effect of piston profile on the combustion, performance and emission characteristics of a wall-guided GDI engine. Four piston profiles are considered for the analysis. From the results, the following conclusions are drawn:

- Better in-cylinder flows are observed in the OPOB piston compared to that of other pistons.
- In the OPOB and the FPOB pistons, mixture is almost stoichiometric around the spark plug, whereas for the OPOS piston, there is very a lean mixture near the spark plug and for the IPOB piston, there is very a rich mixture near the spark plug at the time of ignition.
- The peak in-cylinder pressure and IMEP are the highest for the OPOB piston compared to that of other pistons.
- The peak in-cylinder temperature is lower by about 3%, in the case of the OPOB piston, than that of the FPOB piston. But, higher than the OPOS and the IPOB pistons by about 5 and 15% respectively.
- The heat release rate is higher for the OPOB piston followed by the IPOB, FPOB and OPOS pistons.
- With the OPOB piston, HC emissions are lower and NO_x emissions are higher compared to that of other pistons.

Finally, it is concluded that, the piston profile plays an important role on the combustion, performance and emission characteristics of a GDI engine. Even though with the OPOB piston, the NO_X emissions are higher, it is preferred, compared to that of others because of better in-cylinder flow characteristics, higher IMEP and lower HC emissions. Also, in the future, it is planned to apply particle image velocimetry (PIV) and planar laser induced florescence (PLIF) techniques using an optical engine for different piston profiles to study the mixture formation and flame propagation under different operating conditions.

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Abbreviations

CFD	Computational Fluid Dynamics
GDI	Gasoline Direct Injection
PFI	Port Fuel Injection
IMEP	Indicated Mean Effective Pressure
PIV	Particle Image Velocimetry
PLIF	Planar Laser-Induced Fluorescence
IVO	Inlet valve Opening
EVO	Exhaust Valve Opening
ER	Equivalence Ratio
CAD	Crank Angle Degree
RNG	Re-Normalized Group
KH-RT	Kelvin-Helmholtz-Rayleigh-Taylor
ISO	Pressure Implicit with Splitting of Operators