Numerical investigation of injection parameters and piston bowl geometries on emission and thermal performance of DI diesel engine

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Abstract

Tightening noose on engine emission norms compelled manufacturers globally to design engines with low emission specially NO_x and soot without compromising their performance. Amongst various parameters, shape of piston bowls, injection pressure and nozzle diameter are known to have significant influence over the thermal performance and emission emanating from the engine. This paper investigates the combined effect of fuel injection parameters such as pressure at which fuel is injected and the injection nozzle size along with shape of piston bowl on engine emission and performance. Numerical simulation is carried out using one cylinder naturally aspirated diesel engine using AVL FIRE commercial code. Three geometries of piston bowls with different tumble and swirl characteristics are considered while maintaining the volume of piston bowl, compression ratio, engine speed and fuel injected mass constant along with equal number of variations for injection nozzle size and pressures for this analysis. The investigation corroborates that high swirl and large turbulence kinetic energy (TKE) are crucial for better combustion. TKE and equivalence ratio also increased as the injection pressure increases during the injection period, hence, enhances combustion and reduces soot formation. Increase in nozzle diameter produces higher TKE and equivalence ratio, while CO and soot emission are found to be decreasing and NO_x formation to be increasing. Further, optimization is carried out for twenty-seven cases created by combining fuel injection parameters and piston bowl geometries. The case *D2H1P1* ($H_1 = 0.2$ mm, $P_1 = 200$ bar) found to be an optimum case because of its lowest emission level with slightly better performance.

Keywords AVL FIRE · Injection pressure · Nozzle size · Piston bowl shape

1 Introduction

In the wake of stringent emission regulations and rapid depletion of fossil fuel reserves, production of fuel efficient compression ignition engines with low emission has become extremely challenging task for engine manufacturers around the world today. The governing bodies are increasingly succumbing to pressure from the threats of environmentalists and on the other hand, growing demand for higher fuel efficiency is driven by greater industrialization and dwindling natural energy resources. With enforcement of the Euro VI standards, achieving NO_x

levels lower than 0.3 g/kWh has compelled researchers to accelerate efforts for greater technological advancements and evolve comprehensive optimization design techniques. Further, greater legislations on CO_2 emission may not be ruled out in the near future. The handling of all such mutually competing issues complicates the engine design and necessitates thorough understanding of diesel engines and their mechanism specially the process of fuel combustion. The fuel combustion inside the combustion chamber is a fusion of various intricate processes, and it mainly hinges on formation of adequate fuel and air mixture. The formation of mixture essentially involves

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various issues such as fuel–air interaction, fuel atomization, evaporation of fuel droplets, collision with cylinder walls and other surfaces, rupture of fuel droplets with the boundaries and their speed, energy and mass exchange. In order to reduce exhaust emission and achieve better engine performance necessitates comprehension of all intricate details of piston bowl geometry, dynamics of fuel–air mixture and how these parameters affect the combustion process. Höök and Tang [1] presented a review of fossil fuels depletion and climate change; fossil fuels were the major source of CO₂ emission which resulted in global temperature rise. The problem of fossil fuel depletion and the problem of anthropogenic climate change were closely linked with each other and both the issues should be addressed simultaneously.

Gafoor and Gupta [2] performed numerical study on the effects of piston bowl shape and swirl ratio on exhaust emanating from diesel engines and concluded that reduction in piston bowl diameter led to increase NO_x formation and reduced soot during combustion because of high combustion rate. Jafarmadar et al. [3] examined the effects of combustion chamber geometries on the emission formation and on the performance of engine. It was deduced that the higher bowl depth created stronger squish pressure and increased power output of the engine. Whereas a higher radius of the piston bowl produced higher swirl leads to soot reduction. Corcione et al. [4] noted that re-entrant combustion chamber with elevated TKE and air velocity during fuel injection yielded the excellent results and lowers the smoke emission. Montajiri et al. [5] investigated the spray behaviour over a re-entrant piston cavity of the round lip and a simple cylindrical shape. It was concluded that for DI diesel engine having re-entrant piston cavity of the round lip resulted enhancement of spread of fuel spray and its volume, and subsequently, better fuel–air mixture is formed. Khan et al. [6] deliberated the influence of piston bowl shape and spray pattern on mixing, combustion and emission characteristics of a diesel engine and found that the toroidal combustion chamber generated a strong squish, turbulence and thus resulted in upgraded mixture of air and fuel. It was observed that air motion and spray angle significantly influenced the process of fuel–air mixture preparation and consequently the combustion process and emission characteristics. Fuel sprayed up to an angle 165° was appropriate, however fuel sprayed at 160° showed the excellent results. Fuel spray greater than 165° showed detrimental effects to the engine as the spray directed mostly in the squish region. Risi et al. [7] employed an optimization methodology for different geometries of the combustion chamber and spray injection angle. For lower NO_x emission, narrow and deep combustion chamber with shallow re-entrance was recommended in which the spray should be directed towards

bowl entrance. Rakopoulos et al. [8] presented the comparison of the results of quasi-dimensional model and of CFD model to analyse the influence of piston bowl shape and engine rotations on a DI diesel engine and culminated that predictions of both models are identical. Lim et al. [9] conducted 3D CFD simulation to study the influence of piston bowl geometry and spray angle on the particulate matter emanating from diesel engine. It was concluded that the soot emission and wall impingement were reduced with the modification of piston bowl geometry and spray angle. Wei et al. [10] established initial swirl impact on the in-cylinder air flow by observing the effect of various swirl ratios ranging from 0.2 to 3.2. Gan et al. [11] conducted an experimental study on injection timing and initial swirl ratio affecting emission formation and engine performance of a direct injection diesel engine. It conjectured that the rich air–fuel mixture spots in the cylinder can be reduced by increasing initial swirl ratio. Perini et al. [12] carried numerical study of in-cylinder flow on single cylinder diesel engine, and the results showed that combustion simulations were helpful to understand the spray flow field at the time of combustion. Torregrosa et al. [13] analysed combustion noise and emission formation by compression ignition engine fuelled with premixed charge and deduced that the knock generation mainly depended on mixture formation and phase of combustion which could damage the cylinder walls and piston of the engine. Benajes et al. [14] studied the process of NO_x formation in a diesel engine and observed that the formation of NO depends on slow or poor combustion. It was concluded that high amount of EGR mass fraction also increases NO formation in a high speed diesel engine. Jayashankara et al. [15] studied the intake pressure and engine timing effects on emission and performance of diesel engine. It was culminated that supercharging with inter-cooling enhanced the engine performance and augmented NO_x formation. Abedin et al. [16] used vegetable oil and biodiesel in diesel engine and studied its effect on emission formation. It was concluded that the performance was good but NO_x emission increased because of the high temperature of combustion. Ozer [17] reported that the blend of biodiesel and waste cooking oil in diesel engine and examined the effect on emission and performance characteristic. It was concluded that the NO_x formation increased with an increase in the amount of biodiesel in the mixture. Chandran et al. [18] compared results of numerical simulation with the experimental results of DI diesel engine by considering the blend of distilled waste plastic oil and diesel as alternate fuel. This Study revealed that the simulation of the ratio of 70% diesel and 30% distilled waste plastic oil found to have NO_x and SOOT higher by 7% and 16%, respectively, than conventional diesel in simulation while the value of NO_x and SOOT was lesser by

16% and 17%, respectively, than the experimental value of the blended fuel. Kumar et al. [19] used Solketal with Soyabean biodiesel in the proportion 9%, 10%, 12% and 15% by volume to appraise the performance and emission of diesel engine at 50% load condition. Study revealed that CO and total hydro carbon (THC) emission reduced by increasing solketal percentage while the NO_x , CO_2 and BSFC increased as compared to those with pure diesel. Kumara et al. [20] investigated performance and emission of DI diesel engine by adding di-tert butyl peroxide (DTBP) as an additive in fuel where 40% diesel is replaced by hydrogen fuel. Brake thermal efficiency was found to increase as 87.5, 14.68 and 5.89% with 1, 3 and 5% use of additive DTBP as opposed to the engine operating on pure diesel. NO_x and soot emission also decreased with the use of additive DTBP and hydrogen as secondary fuel. Hemanandh et al. [21] prepared a bio-fuel from the fish waste, slaughter waste of chicken and broiler excrete by pyrolysis process. This study revealed that the ratio of bio-fuel and diesel by volume 10:90 when used in diesel engine reduced emission of CO by 30.52%, HC by 25.41% and NO_x by 22.84%. Along with this it also increased CO_2 emission by 30.52% and 23.8% at 25% and 100% load, respectively. Viswanathan et al. [22] used fish oil ethyl ester (FOEE) with diesel in DI diesel engine. This study revealed that the use of FOEE reduces emission of CO, NO and soot and preheating of FOEE by exhaust gas could further reduce emission level. Jaichandar et al. [23] performed experiments on biodiesel fuelled diesel engine and analysed the influence of fuel injection pressure and piston bowl shape. It was concluded that rise in injection pressure resulted in reduction of smoke and unburned hydrocarbon with increased NO_x due to better combustion. Agarwal et al. [24] examined the effects of pressure of fuel injection and timing of fuel injection over spray characteristic and particulate size number distribution using the fuel blend of Karanja biodiesel and pure diesel. The results demonstrated that higher penetration of spray tip and wider spray area were obtained with fuel injected at elevated pressures. Advancement the time of fuel injection resulted in increment of average particulate size. The lowest particulate number was obtained with 10% blend of biodiesel and diesel. Chinnamuthu et al. [25] varied fuel injection nozzle angle in the plane perpendicular to cylinder axis and considered angle of 30°, 60° and 90°. This study showed that angle 60° nozzle yielded better emission and combustion characteristics because of higher turbulence produced by fuel spray. Fayad [26] investigated the effects of fuel injection strategies over the emission level of smoke and NO_x by varying timing of fuel injection at different load and engine speed by keeping design invariant. Experimental result showed that advancement in fuel injection resulted in reduction of NO_x and BSFC. The

similar results were also reported with low speed and high load. High fuel injection pressure reduced smoke exhaust. Kesharwani et al. [27] adopted multi-injection strategy for injection of fuel by injecting some portion of fuel prior to the main injection which created better combustion atmosphere for principal fuel injection. Pilot injection fuel mass was taken as 5, 10 and 15% by maintaining the total mass of fuel injected invariant. The case in which fuel mass kept as pilot injection is 5%, and principal injection is 95% turned out as an optimum as soot and NO_x emission reduced by 16% and 10%, respectively, as compared to single injection. Wang et al. [28] employed a thermal barrier coating (TBC) to reduce the losses of heat from the inner side of the combustion chamber. The study revealed that the lower thermal conductivity and increased porosity of TBC resulted in 100–200 K high temperature into the combustion chamber yielding improved thermal efficiency and also higher temperature at exhaust improved after-treatment of outgoing gases to lower emission levels.

After comprehensive review of literature, it can be conjectured that there are studies undertaken to comprehend the effects of modifications to typical and toroidal combustion chamber geometries to improve performance and reduce emission. Further, a large number researcher also investigated the effects of injection parameter as well either conjunction with piston bowl geometrical variations with a view to identical objective. However, analysing the combined influence of typical combustion chamber geometry unlike the toroidal combustion chambers along with injection parameters is the subject of current research. The principal objective of the present analysis is to augment performance and minimise emission of a standard DI engine by combining variations in combustion chamber geometry and fuel injection parameters such as injection pressure and injection nozzle size. Three combustion chambers with different tumble and swirl ratios with three different injection pressures and three different nozzle diameters are chosen to comprehend the cumulative influence of these factors which are crucial to engine performance and emission control.

2 Model descriptions

In this study a four-stroke direct injection, single cylinder, naturally aspirated, water cooled diesel engine is considered. In all the simulations performed, piston bowl volume, compression ratio, engine speed and injected fuel mass are maintained constant so that results are purely based on the effect of fuel injection parameter and piston bowl geometry. Figure 1 shows the typical piston bowl design comprising of various parameters used for defining the

Fig. 1 TDC Typical outline of piston bowl geometry representing the main geometrical parameter

Table 1 Parameter defining the design of different piston bowl

Parameters	D_1	D_2	D_3
T_m (m)	0.0058	0.00656	0.0049
D_i (m)	0.053	0.05031	0.058
D_a (m)	0.06	0.05622	0.06
T (m)	0.0175	0.01999	0.01564
R_1 (m)	0.0065	0.00744	0.00475
S_1 (m)	0.0201	0.01235	0.02223

Fig. 2 Mesh distribution of three piston bowl geometries at TDC

Table 2 Geometrical specification and number of cells for different piston bowl

	Number of cells	Surface area (m ²)	Volume (m ³)
D_1	32,480	0.00572	3.6314×10^{-5}
D_2	32,156	0.00609	3.6314×10^{-5}
D_3	32,252	0.00559	3.6314×10^{-5}

geometry. Table 1 shows outline parameters for the three piston bowl designs. Figure 2 depicts the computational domain for three considered piston bowl design at TDC. The computational domains for each of the geometry D_1 , D_2 and D_3 consist of only one-third of combustion chamber for the sake of symmetry. Table 2 shows the actual number of the cell used for numerical computations of corresponding piston geometries along with the bowl volume and surface area. It is evident that volume is constant, and the surface area is varied with the design of piston bowl geometry. Tables 3 and 4 show the engine specifications and initial and boundary conditions, respectively.

Table 3 Technical details of the engine

Engine	4-Stroke, single cylinder, water cooled, diesel engine
Cylinder bore	87.5 mm
Stroke length	110 mm
Connecting rod	234 mm
Compression ratio	17.5
Speed	1500 RPM
Power	5.2 KW @1500 RPM
No. of the nozzle hole	3
Nozzle outlet diameter	0.2,0.25,0.3 mm
Type of injection	Direct injection
Start of injection	23° BTDC
Injection pressure	200,250,300 bar

Table 4 Initial and boundary condition used for the numerical model

Cylinder head temperature	425 K
Liner temperature	375 K
Piston temperature	525 K
Initial pressure	1 bar
Residual gas ratio	0.05
Initial temperature	335 K

Simulations are carried out with inlet valve closing at 35.5° crank angle (CA) after BDC and exhaust valve opening at 35.5° CA before BDC. Along with piston bowl geometries, three pressures of fuel injection (200 bar, 250 bar and 300 bar named as P_1 , P_2 and P_3 , respectively) and also three outlet diameter of fuel injecting nozzle (0.2 mm, 0.25 mm and 0.3 mm named as H_1 , H_2 and H_3 , respectively) are taken for numerical simulations.

3 Mesh independency study

The mesh is generated using mesh generator tool of AVL Fire commercial software with active perpetual licence version 2013.1. Computational domain is discretised using structured hexahedron mesh. To determine the optimum mesh for onward computations, grid independence test was performed on three different grid sizes with numbers of cells as 14,052, 30,428 and 45,072, respectively. The results are compared in terms of in-cylinder pressure against the crank angle. Figure 3 depicts that peak cylinder pressure variation at TDC for grid sizes of 14,052 and 30,428 cells is 6.1687%, and variation between 30,428 and 45,072 cells is 0.2541%. It can be deduced that further refinement of mesh from 30,428 cells to 45,072 cells, no appreciable variation in cylinder pressure, hence, 30,428

Fig. 3 Effect of grid resolution on the pressure curve

numbers of cells may be taken as optimum size for further simulations. For different piston bowl geometries, small variation in number of cells is permissible.

4 The model validation

In this study, the process of combustion occurring within the engine cylinder is numerically manifested using AVL FIRE which specially designed for modelling internal combustion engines. Various numerical models representing the physical and chemical processes during combustion are listed in Table 5. K- ζ -f model, developed by Hanjalic et al. [29], is used for modelling turbulence and turbulent heat transfer inside the combustion chamber. Fuel spray-wall interaction model known as 'walljet1' introduced by Reitz et al. [30] is used for the fuel particles striking wall of combustion chamber during the injection process. The spray breakup process is modelled by using the WAVE breakup model [31]. Dukowicz model [32] is being used for fuel spray droplet heating and evaporation. Extended

Table 5 sub-model used for simulation

Heat transfer wall	Standard wall function
Combustion	ECFM-3Z
Spray Breakup	WAVE
Wall interaction	Walljet 1
Evaporation	Dukowicz
Turbulence	k- ζ -f
Soot model	Kinetic model
NO model	Extended Zeldovich
Wall treatment	Hybrid wall treatment

coherent flame model (ECFM-3Z) is used for modelling the diesel combustion process. This model manifests turbulent mixing, emission and flame spread. It can be used for both auto ignition and spark ignition engines simultaneously. Combustion model ECFM-3Z was described by Colin et al. [33]. Extended Zeldovich [34] model is used for NO_x formation during combustion. NO_x is formed during combustion at high temperature zone. A kinetic model is used for soot formation inside the combustion chamber. Soot formation is a complicated process, involving various reactions. Kinetic model of soot formation is described by Mauss et al. [35]. Since Tatschl et al. [36] performed validation of all the aforementioned numerical models. Evidently, this establishes the efficacy of the AVL FIRE code and, thus, separate validation is not necessary. However, case specific validation in the contest of the present investigation is obligatory.

Numerical simulation is carried out using the input parameter of the experimental condition at full load. For the validation of simulation results, predicted data of pressure curve and heat release curve are compared with experimental results of single cylinder DI diesel engine measured by Khan et al. [6]. Figures 4 and 5 show the simulation results indicating satisfactory agreement with the experimental data, hence, simulation process is adequately modelled.

5 Results and discussion

To examine the effects of piston bowl geometries and fuel injection parameter such as nozzle hole diameter and fuel injection pressure, twenty-seven number of cases were

Fig. 4 Comparison of experimental and simulated result for in-cylinder pressure

Fig. 5 Comparison of experimental and simulated results for the rate of heat release

Table 6 cases used for simulation

Geometry	Fuel injection pressure (bar)	Nozzle outlet diameter (mm)		
		$H_1=0.2$	$H_2=0.25$	$H_3=0.3$
D_1	$P_1=200$	$D_1P_1H_1$	$D_1P_1H_2$	$D_1P_1H_3$
	$P_2=250$	$D_1P_2H_1$	$D_1P_2H_2$	$D_1P_2H_3$
	$P_3=300$	$D_1P_3H_1$	$D_1P_3H_2$	$D_1P_3H_3$
D_2	$P_1=200$	$D_2P_1H_1$	$D_2P_1H_2$	$D_2P_1H_3$
	$P_2=250$	$D_2P_2H_1$	$D_2P_2H_2$	$D_2P_2H_3$
	$P_3=300$	$D_2P_3H_1$	$D_2P_3H_2$	$D_2P_3H_3$
D_3	$P_1=200$	$D_3P_1H_1$	$D_3P_1H_2$	$D_3P_1H_3$
	$P_2=250$	$D_3P_2H_1$	$D_3P_2H_2$	$D_3P_2H_3$
	$P_3=300$	$D_3P_3H_1$	$D_3P_3H_2$	$D_3P_3H_3$

created by combining three nozzle outlet diameters, three fuel injection pressures and three piston bowl geometries as shown in Table 6.

5.1 Effect of piston bowl geometry

To analyse the effects of piston bowl geometry, three piston bowl geometries are considered with the same fuel injection pressure and nozzle outlet diameter. Figure 6 depicts turbulence kinetic energy and equivalence ratio distribution for considered piston bowl geometries. It can be observed that TKE is maximum for D_3 and minimum for D_1 . Figure 7 shows the effects of piston bowl geometry on swirl ratio. The initial swirl ratio is unity for all cases. It can be observed that the swirl ratio value in the combustion chamber initially decreasing slightly this is because of wall friction and dissipation in the fluid due to turbulence. As the volume of combustion chamber decreases significantly, swirl magnitude increasing rapidly because of the moment of inertia of air mass and become maximum at TDC. After TDC it start decreasing because of volume expansion, wall friction and turbulence dissipation in the fluid. The depth of piston bowl D_2 ($T=0.01999$ m) is highest among the all considered piston bowl which do not let air escape from piston bowl, hence produces highest swirl motion, and the depth of D_3 ($T=0.01564$) is least so that produces lower swirl motion into the combustion chamber. Figure 8 shows pressure and temperature variation for different piston bowl geometries. Pressure and temperature start increasing rapidly as the heat release starts from injected fuel. Highest peak pressure is obtained for bowl geometry D_1 and lowest peak pressure for D_3 , however swirl ratio magnitude is maximum for D_2 but due to the large surface area of piston bowl heat

Fig. 6 Comparison between various piston bowl geometries in terms of turbulence kinetic energy and equivalence ratio

Fig. 7 Variation of swirl number with CA for different piston bowl geometry

Fig. 9 NO mass fraction for different piston bowl geometry

Fig. 8 Pressure and temperature for different piston bowl geometry

Fig. 10 CO mass fraction for different piston bowl geometry

dissipation from walls is also high which decreases peak pressure into the combustion chamber. Figure 9 depicts the NO mass fraction variation with crank angle for different piston bowl geometries which is produced during combustion due to contained nitrogen and oxygen in air. It can be seen that the NO level at the end of the expansion stroke is minimum for D_2 and maximum for D_1 . This may be attributed to the fact that NO mass fraction is temperature dependent; higher is temperature higher will be the NO emission. Figure 10 shows the CO mass variation with crank angle, which is minimum for piston bowl D_2 and maximum for D_3 at the end of the expansion stroke. This may be caused by better combustion characteristics of this combustion chamber. Figures 11 and 12 show soot

mass fraction variation and burnt fuel mass fraction with crank angle, respectively. It is observed that soot formation is higher for piston bowl geometry D_1 . This may be attributed to over-rich mixture spot formation outside the bowl as well as lower TKE inside the bowl volume which also resulted in slow fuel burning rate. Soot formation is the lowest for piston bowl D_2 because of uniformity of mixture and higher turbulence inside the bowl. This also resulted in higher fuel burning rate.

5.2 Effect of fuel injection pressure

To analyse the effect of fuel injection pressure, three fuel injection pressures is considered with the same piston

Fig. 11 Soot mass fraction for different piston bowl geometry

Fig. 12 Burnt fuel mass fraction for different piston bowl geometry

bowl geometry and nozzle outlet diameter. As the fuel injection pressure increases injection duration decreases due to increment in fuel injection velocity into the combustion chamber. TKE is surely get affected by fuel injection pressure, Fig. 13 depicts that as fuel injection pressure increases turbulent kinetic energy increases. P_1 and P_3 case have minimum and maximum values of TKE, respectively. The equivalence ratio is also getting better by increasing fuel injection pressure this is because atomization of fuel particle is getting improved by increase in fuel injection pressure as we can see in Fig. 13. Figure 14 shows the swirl ratio with crank angle. Fuel injection pressure does not affect swirl magnitude much, it slightly affects swirl at TDC due to the higher fuel injection rate. These effects lead to

better combustion, and heat release rate got increases with fuel injection pressure due to this pressure and temperature get increases at a higher rate as the combustion take place. Figure 15 shows pressure and temperature variation with crank angle. P_1 have minimum pressure and temperature, and P_3 have maximum pressure and temperature. NO mass fraction is increases with increase in fuel injection pressure because the temperature of combustion chamber is increased. Figure 16 shows NO formation maximum for the case P_3 and minimum for P_1 . CO mass fraction decrement rate after reaching peak value is higher for case P_3 and lower for P_1 as shown in Fig. 17 and gives minimum value for P_3 at the end of the expansion stroke. Figures 18 and 19 clearly show that soot formation is reduced with increase in fuel injection pressure owing to higher burning rate of fuel.

5.3 Effect of nozzle diameter

To analyse the effect of nozzle diameter, three nozzle outlet diameters are considered with same piston bowl geometry and fuel injection pressure. As nozzle diameter increases, fuel injection duration is decreases. Here the cause of injection duration decrement is increased area of flow for the fuel into the combustion chamber. TKE is affected due to fuel injection rate and because of this TKE is high with greater nozzle outlet diameter as shown in Fig. 20. TKE is minimum for H_1 and maximum for H_3 also equivalence ratio is higher for greater nozzle outlet diameter which leads to a high combustion rate. Figure 20 shows maximum equivalence ratio for H_3 and minimum for H_1 . Figure 21 shows swirl ratio variation with crank angle for different nozzle outlet diameters. It can be seen that influence of nozzle outlet diameter on swirl ratio is not significant, however, it slightly affects swirl ratio at TDC. This may be due to variation of fuel injection rate with nozzle outlet diameter. Figure 22 depicts variation of pressure and temperature with crank angle for different nozzle outlet diameters. It can be seen that as the heat release rate increases with increasing nozzle outlet diameter which in turn increases the peak value of pressure and temperature. H_1 and H_3 show the lowest and highest value for pressure and temperature inside the combustion chamber, respectively. Because of temperature increment, NO formation increases with nozzle outlet diameter as we can see in Fig. 23; NO mass fraction is maximum for H_3 and minimum for H_1 . In the case of CO mass fraction, CO level increases rapidly as combustion takes place and have the highest peak for H_3 as shown in Fig. 24. After attainment of peak value, it decreases, and decrement rate is higher for H_3 which gives minimum value to H_3 at the end of combustion because of higher swirl

Fig. 13 Comparison between various fuel injection pressures in terms of turbulence kinetic energy and equivalence ratio

Fig. 14 Variation of swirl number with CA for different fuel injection pressure

Fig. 15 Pressure and temperature for different fuel injection pressure

magnitude at the end of the expansion stroke. Figure 25 depicts soot mass fraction variation with crank angle for nozzle outlet diameter; soot formation appeared to decrease with increase in nozzle outlet diameter. At the end of the expansion stroke, soot mass fraction

maximum for H_1 and minimum for H_3 . Figure 26 shows that the burnt fuel mass fraction is increases with a nozzle outlet diameter.

Fig. 16 NO mass fraction for different fuel injection pressure

Fig. 18 SOOT mass fraction for different fuel injection pressure

Fig. 17 CO mass fraction for different fuel injection pressure

Fig. 19 Burnt fuel mass fraction for different fuel injection pressure

6 Optimization

After the discussion of results, an optimization study is being done on above twenty-seven cases to find an optimum case. Fig. 27 shows CO and soot are plotted against the NO mass fraction for all cases. $D_3H_1P_1$ is taken as the reference case. Further ten cases are taken for comparison which is having minimum emission whose results are shown in Table 7. Taking emission as priority $D_2H_1P_1$ is an optimum case as we can see in Fig. 30. Figures 28, 29 and 30 show that CO, NO and soot emission reduced by 57.108%, 62.299% and 57.807%, respectively, while BSFC decreases by 3.136%. However, from a performance point of view $D_2H_2P_2$ may be considered as an

alternate case. In this case, an increase of 6.3872% and 23.0392% is noted in IMEP and NO, respectively, while CO, soot and BSFC decreased by 84.0347%, 91.6052% and 7.622%, respectively. Increase in NO mass fraction can be controlled by proper handling of exhaust gases by after-treatment method.

7 Conclusion

In the present work, the combined effects of fuel injection pressure, injection nozzle size and piston bowl geometry on emission characteristics and thermal performance of a compression ignition engine were investigated. The

Fig. 20 Comparison between various nozzle outlet diameters in terms of turbulence kinetic energy and equivalence ratio

Fig. 21 Variation of swirl number with CA for different nozzle outlet diameter

Fig. 22 Pressure and temperature for different nozzle outlet diameter

Fig. 23 NO mass fraction for different nozzle outlet diameter

Fig. 25 SOOT mass fraction for different nozzle outlet diameter

Fig. 24 CO mass fraction for different nozzle outlet diameter

Fig. 26 Burnt fuel mass fraction for different nozzle outlet diameter

conclusions drawn from the present research are encapsulated as follows:

1. Swirl and TKE tend to increase with increasing depth of piston bowl geometry during upward movement of the piston up to TDC. However after TDC, swirl takes a sharp downhill trend while TKE remains almost invariant till the end of the expansion stroke. Further, NO is high for higher piston depth.
2. With increasing fuel injection pressure, TKE and equivalence ratio increases during fuel injection period leading to better combustion and, hence, reducing the soot formation and increasing NO_x emission.
3. Increase in nozzle outlet diameter reduces the duration of fuel injection. TKE and equivalence ratio are

high for higher nozzle outlet diameter. NO_x emission increases with increasing nozzle outlet diameter while soot formation decreases due to higher combustion rate.

4. Amongst twenty-seven cases analysed, $D_2H_1P_1$ is found to be optimum in view of low emission. However, $D_2H_2P_2$ may be considered as an alternate case from the performance point of view. Higher NO_x level in this case can be minimised by using an appropriate after-treatment.

Fig. 27 CO and soot versus NO mass fraction for all cases

Fig. 29 BSFC for the selected ten cases

Table 7 comparison of results of selected cases

CASES	NO _x	CO	SOOT	IMEP	BSFC
D1H1P1	0.000233	0.000594	5.69E-07	5.25	0.2372
D2H1P1	0.0000875	0.000869	0.000000481	5.14	0.244
D2H1P2	0.000162	0.000599	2.08E-07	5.24	0.2379
D2H1P3	0.000233	0.000439	9.36E-08	5.3	0.2341
D2H2P1	0.000136	0.000664	1.83E-07	5.23	0.2381
D2H2P2	0.000251	0.000368	0.0000000957	5.33	0.2327
D2H3P1	0.000236	0.000401	1.72E-07	5.31	0.2335
D3H1P1	0.000204	0.002305	0.00000114	5.01	0.2519
D3H2P1	0.000234	0.003315	3.35E-07	5.02	0.2518
D3H3P1	0.000246	0.003225	4.07E-07	5.06	0.2448

Fig. 28 IMEP for the selected ten cases

Fig. 30 Soot and NO mass fraction for the selected ten cases

8 Future scope of work

This work can be further extended by taking crevice volume which is the volume between piston and the cylinder liner, in to account since it can affect the emission as well as the engine performance significantly owing to fuel getting trapped into it. Effect of load and speed variation may be investigated since they affect the actual engine performance.

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Declarations

Conflict of interest The authors declare that they have no conflict of interest.

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