Research article

Numerical study of different shape design of piston bowl for diesel engine combustion in a light duty single-cylinder engine

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ABSTRACT

Diesel engine is the prime mover on land transportation industry and used in a variety of power generation applications due to their higher fuel efficiency. However, the engine research community faces a major hurdle in the form of rigorous restrictions introduced in the Glob to reduce pollutant emissions from internal combustion engines. Different piston bowl shape designs allows more precise mixing before combustion to enhance in the optimization using computational calculations to reduce emissions. The investigation was to reduce the NOx and PM emissions using combustion simulation comparing with each piston of a single-cylinder engine at a CR of 24, 4-stroke, and water-cooled Engine. The four piston bowl shapes of DSEVL2 BMW M47T, Shallow Hesselman, Lombardini 15LD350, and DOOSANP158FE were analyzed by the Diesel-RK combustion simulation. After successful validating; the simulation model shows that the peak cylinder pressure of Piston-2 is 131bar and the peak cylinder pressure of Piston-4 is 113bar. The Maximum Cylinder Temperature of the Piston-2 is 2048.2k, and the lowest value of Cylinder Temperature of the Piston-4 is 1680.9k the cylinder temperature of Piston-2 is 18% higher than Cylinder Temperature of Piston-4. The simulation result indicates that the temperature is within the acceptable limit in between 1400-2000k except for the piston temperature of 2048.2k. The PHRR of the Piston-3 is 0.082 with great variation in between maximum and minimum due to the presence of pre-and post-injection, the HRR-P4 is 0.035 J/CA with the single injections. The HRR of the Piston-3 is the highest while HRR of the Piston-4 lowest with 39%. The NOx in the exhaust gas is 25.62 in the NOx piston-1; 16 in NOx of Piston-2, 18.2 in NOx-P3, and NOx-P4 is 12.74 g/kWh respectively. The NOx of the NOx-P2 is lower than first and second piston due to the lower fuel fraction of NWF dilution outer the sleeve, low fuel fraction in core of the free spray, low fuel fraction in fronts of the free spray, low fuel fraction in the core of the fuel free spray. The Particulate Matter emission in PM-P1 is 0.35, and PM-P2 is 0.43 g/kWh which is higher than all the other. Although there is a substantial decrease in PM, a penalty in NOx is observed for PM-P1 but PM of the P2 is higher after the peak result of emission.

1. Introduction

Diesel engines are the first choice as the main mover on land transportation industrial usage (Nghia et al., 2022; Xu et al., 2018). Due to their higher thermal efficiency, CI Engines are vastly used in a variety of applications. Due to fuel/air mixture regulated diffusion combustion, typical diesel combustion emits high amounts of NOx and PM (Ahmadi and Hosseini, 2018). The design of the chamber is a vital part of the whole engine system for achieving a standardized burn rate and increasing the engine's efficiency (S. K. A. S. A. Siddique et al., 2015). Piston bowl shape design selection has a great impact on the emission, combustion, and performance characteristics of a diesel engine (Seelam et al., 2021). The engine research community faces a major hurdle in the form of rigorous restrictions introduced in the Glob to reduce pollutant outputs from internal combustion engines. Despite its efficiency, typical mixing-controlled diesel combustion in CI diesel engines needs complicated and expensive exhaust after-treatment systems to meet the NOx, soot, and other emission limit values recommended in current standards. The high temperature gained during the CDC, as well as the oxygen availability outside of the spray plume, result in undesirable emissions (Pham, 2019; Armin et al., 2021). The piston bowl is crucial in aiding the
mixing of air-fuel within the combustion chamber; hence it’s proposed that the surface-to-volume ratio be reduced (Li et al., 2016).

As stated in the introduction, a variety of piston bowl shape designs were designed and tested using computational calculations and actual tests to reduce soot and NOx within the squish zone while lowering well heat transfer. Due to the PBSD, which allows for more correct mixing before combustion, more mixing from the piston bowl shape is necessary for the less time DI diesel fuel (AndreyKuleshov et al., 2014; Zhao et al., 2020). The purpose of this study was to reduce the NOx and PM by using combustion simulation and compare with each other by using 0.325L engine displacement, single-cylinder diesel engine at AASTU (Addis Ababa Science and Technology University), which has a compression ratio of 24, 4-stroke, and is water-cooled. To enhance thermal efficiency and minimize CDC emissions, combustion simulations using the Diesel-RK Software to evaluate bowl shape parameters in a light-duty engine. The engine running circumstances employed in the piston bowl design evaluation model were chosen to be indicative of previous CDC incidents (Sreedharan and Krishnan, 2018; Ya and Koca, 2014). The wide-shallow incineration chamber enhanced the engine’s operable load range as a consequence of the more stable combustion process and decreased likelihood of engine knocking under all operating engine load situations. The wide-shallow combustion chamber significantly reduced heat transfer loss under all engine running conditions (Naouer et al., 2010; A. Kuleshov and Mahkamov, 2008; Jason Miwa, 2021).

2. Diesel-RK modeling

The modelling and simulation study of CDC engine and comparative study with each is by using Diesel-RK combustion simulation, with the main features similar to known thermodynamic software. Diesel-RK software developed by Bauman Moscow State Technical University of Russia had new innovative applications that the other programs do not have. It focused on optimization of incineration processes in a diesel

| Table 1. Basic engine specifications. |
|--------------------------------------|
| Make/Model | Kubota EA330-E4-NB1 |
| No of Cylinder | 1 |
| Engine type | W-cooled, 4s D-engine |
| Bore* Stroke | 77.0 mm * 70.0 mm |
| Engine displacement | 0.325L |
| Speed | 1900 rpm |
| Fuel Injection Pump | Bosch K type mini pump |
| Combustion Chamber | Spherical type (TVCS) |
| Injection Nozzle | DN-PD Mini Nozzle |
| Injection Timing | 25.5° B.T.D.C |
| Injection Pressure | 137.3 bar |
| Compression Ratio | 24 |
| IVO | 20° before T.D.C |
| IVC | 45° after B.D.C |
| EVO | 50° before B.D.C |
| EVC | 15° after T.D.C |

Figure 1. In-cylinder different Piston bowl Geometries.
engine with the different piston bowl shape designs as shown in the Figure 1 are analyzed and optimization supposing the same working of all cylinders in the engine allows a substantial increase in operating speed and makes it possible to optimize the complex diesel engine tasks and the investigation is with the engine specifications as shown in Table 1.

A search for the best alternative is a time-consuming and labor-intensive task due to a large number of accessible possibilities and their combustions. Modelling and simulation, as well as computational optimization, can be used to identify trends and suggest ways to enhance engines. The three types of diesel engines simulation models available today are: Zero (single-zone), quasi-dimensional, and multi-zone models. The PBG is critical, notably for optimal combustion, performance, and emission characteristics in diesel engines (Regan, 2021). The four alternative piston bowl shapes were analyzed and compared each other. The NOx production is simulated using the thermal Zeldovich mechanism; while the combustion is simulated using the Diesel No-2 fuel property stated in Table 2.

### 2.1. Model validation

Varies combustion chambers due to piston bowl shapes are compared with each Combustion Simulations findings generated using Diesel RK Software. The pressure variation with the crank angle of the CDC Piston bowl geometry was compared at the same load situation to validate the projected numerical results (S. A. S. K. A. Siddique et al., 2015). Figure 2 shows the pressure variation as a function of crank angle. The simulation’s trends are comparable to each other. However, there is a small

### Table 2. Basic diesel fuel Properties.

| Parameters                          | Results |
|-------------------------------------|---------|
| Density at 15 °C                    | 0.840   |
| 90% Volume recovered, °C            | 342     |
| FBP in Fuel Temperature, °C         | 380     |
| Total Sulfur, % wt                  | 0.002   |
| Flash point, PMCC, °C               | 66      |
| LHV in MJ/KG                        | 42.5    |
| IBP, °C                             | 38      |
| Kinematics viscosity @ 40 °C        | 3.5     |
| Cloud point °C                      | Max +5  |
| Cetane index                        | 48      |
| CCR, % Wt                           | 0.2     |

### Table 3. Emission, CP, and CT simulation and experiment results and its validation (Seelam et al., 2021; Muralidharan, 2021; Bawankar and Gupta, 2016; Hanson et al., 2012; Ganji et al., 2018).

| Toroidal bowl & similar piston design [21]-[24] | Hemispherical bowl & similar piston design [21]-[24] |
|--------------------------------------------------|---------------------------------------------------------|
| N° CP CT HRR NOx PM CO                           | N° CP CT HRR NOx PM CO                                   |
| P1 ↑↑ ↑ ↓ ↑ ↑ ↑                               | P2 ↑↑ ↑ ↑ ↑ ↓ ↓ |
| TRB ↓ ↓ - - ↓ -                                    | HSB ↑↑ ↑ ↑ ↑ ↓ ↓ |
| TCC ↑↑ - - ↑ -                                    | HCC ↑ - - - - - - |
| RCCI-OEM ↑↑ ↑ ↑ ↑ -                                | MRCCI ↑ - - - - - - |
| Omega ↑↑ ↑ ↑ ↑ -                                  | HM ↑ - - - - - - |

| Re-entrant bowl & Similar piston design [21]-[24] | DOOSANP158FE - Bowl & similar piston design [21]-[24] |
|--------------------------------------------------|---------------------------------------------------------|
| N° CP CT HRR NOx PM CO                           | N° CP CT HRR NOx PM CO                                   |
| P3 ↓ ↑ ↑ ↑ ↑ ↑                                     | P4 ↓ ↓ ↓ ↓ ↓ ↓                                |
| REB ↓ - - - -                                      |                                    |
| SCC ↓ - - - -                                      |                                    |
| CDC ↓ - - - -                                      |                                    |
| SC ↓ - - - -                                       |                                    |

Where: ↑↑ maximum value, ↑ -second value, ↓-Third value, ↓↓- minimum value, HSB-Hemispherical bowl, TRB-Toroidal bowl, and REB-Re-entrant bowl, HCC-Hemispherical combustion chamber, SCC- shallow depth combustion chamber, TCC – toroidal Combustion chamber, OM-omega bowl, HM-hemispherical bowl, SC-single curved, DC-double curved P1-DSEVL2 BMW M47T, P2-Shallow Hesselman, P3-Lombardini 15LD350, P4 - DOOSANP158FE.

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The PBG is critical, notably for optimal combustion, performance, and emission characteristics in diesel engines (Regan, 2021). The four alternative piston bowl shapes were analyzed and compared each other. The NOx production is simulated using the thermal Zeldovich mechanism; while the combustion is simulated using the Diesel No-2 fuel property stated in Table 2.

![Figure 2. In-cylinder pressure of various PBG [bar].](image1)

![Figure 3. Engine setup configuration. Where: DAQ-is data acquisition system, AFM- Air flow meter, ECDC-Eddy current dynamometer, EGA-Exhaust gas analyzers.](image2)
differential in peak pressure, temperature, heat release rate, and emissions. Table 3 also shows a comparison of Performance metrics like indicated mean effective pressure; mechanical efficiency; and indicated efficiency. The distinction between the numerical value comparable analysis for soot and NOx, the disparity between numerical values of pressure, temperature, HRR, Oxides of nitrogen, and particulate matter is found.

The Engine setup Configuration for the combustion, and emission with the data acquisition system, air flow meter, Eddy current dynamometer, Exhaust gas analyzers is assumed to be organized in the Figure 3.

3. Result and discussion

3.1. In-cylinder pressure

The combustion simulation of piston bowl shapes, namely, DSEVL2 BMW M47T, Shallow Hesselman, Lombardini 15LD350, and DOOSANP158FE were selected for the present study. Based on the combustion, performance, and emissions, the four different Piston Bowl Shape Designs are chosen to analyze further to optimize for the performance and reduce emissions. Comparison of the piston bowl shapes pressure with CA shows peak cylinder pressure of DSEVL2 BMW M47T (Cylinder Pressure-P1) is 129 and Shallow Hesselman (Cylinder Pressure-P2) bowl geometry is 131, Lombardini 15LD350 (Cylinder Pressure-P3) which is 122, and DOOSANP158FE (Cylinder Pressure-P4) result is 113bar in which the maximum in-cylinder pressure is 13.8 percent more than the lower value.

The maximum CP values for various piston bowls at a fuel injection pressure of 137bar are given; CP1, CP2, CP3, and CP4 are 129, 131, 122, and 113bar respectively as shown in Figure 2. The dominating premixed ignition stage is the resultant of the extended period for fuel-air blending in Shallow Hesselman is better and that is why the in-cylinder pressure value increases (Seelam et al., 2021). The angle at which the Maximum Cylinder Pressures takes place in each PBSD is at $3^\circ$, $4^\circ$, $3^\circ$ and $4^\circ$ A. TDC respectively. Because of better combustion, the fuel blend has a higher peak combustion pressure (Karthikeya Sharma et al., 2015).

3.2. In-cylinder temperature and HRR

It takes into account and allows optimization of PBS, injector design and location, the shape of injection profile including multiple injections. The RK-model accounts for drop sizes, the interaction of free sprays with swirl, spray and wall impingement, evolution of near-wall flow

Figure 4. a: spray & near wall fuel dilute, b: Core of free spray, c: Front of free spray and d: Core of near wall flow.

Figure 5. In-cylinder temperature of various PBSG [K].
formed by spray, hit of fuel on cylinder head surface, on cylinder liner, crossing of NWF formed by adjacent sprays. The other reason is lower fuel fraction of near wall flow dilution, low core of the free spray, low fuel fraction in fronts of fuel free spray, and lower fuel fraction in the core of the fuel free spray is the reason way temperature and pressure is high in Shallow Hesselman Piston as shown in the Figure 4.

Figure 5 shows the Maximum Cylinder Temperature of the Shallow Hesselman is 2048k, and the minimum value of DOOSANP158FE result is 1681k. The cylinder temperature of the Shallow Hesselman piston is 18% higher than the minimum model results due to the shape and depth of its piston bowl which dominates the premixed ignition stage is the resultant of the extended period for fuel-air blending in Shallow Hesselman is better and that is why the in-cylinder temperature increases (Seelam et al., 2021). According to Paykani et al. (2016), peak local equivalence ratios should not be significantly higher than stoichiometric, the min and max temperatures should be kept within a very narrow range of 1400K–2000K to allow for complete combustion and clean combustion with low NOx and Particulate Matter emissions (Kim et al., 2009). As a result, the simulation product reveals that all the cylinder temperatures are within permissible limits, except for the Cylinder Temperature of the Piston-2 is 2048k.

Equations for the forecasting of reasonably accurate HRR for cycles calculations are offered for particular engines operating under normal conditions, for use with a basic calculation (Banack and Service, 2013).

3.3. NOx and PM emissions

The emissions from the diesel engine are NOx and PM but HC and CO are not the main issue in diesel engine because of the lean engine operation. Figure 7. Shows that the Oxides of nitrogen in the exhaust gas for P1, 2, 3, 4 is 25.6, 16.18, 2, and 12.74 g/kWh respectively. The simulation results in ppm are converted in to g/Kwh as in the Eq. (1) by the (Gopal et al., 2014)

\[
\text{NOX}_{\text{g/kWh}} = \frac{\sum (\text{NOx mass} \times \text{Wf})}{\sum (\text{BP} \times \text{Wf})}
\]  

In-cylinder temperature and NOx emission are directly proportional in combustion temperature; however, the maximum cylinder temperature of Piston-2 have a minimum NOx emission, this is due to the PBSDG effect in the fuel spray pattern. The lower fuel fraction of Near Wall Flow dilution outer the sleeve, in fronts of the free spray, low fuel fraction in the core of the fuel free spray as shown in the Figure 4 has the effect on the reduction of NOx in the piston-2 relative to all the other.

Figure 8 shows that the Particulate Matter emission is 0.35 in the Piston-1; Piston-2 is 0.42, in Piston-3 is 0.43 and, in the Piston-4 result is 0.24 g/kWh. Though there is a substantial decrease in PM, a penalty in NOx is observed for PM of the Piston-1. But PM of the Piston-2 is continuously higher than all the others after the peak result of emission. So, there is a need for further improvement of PM of the Piston-1 geometry for simultaneous reduction of NOx emissions without compromising the engine performance. Specific NOx emission reduction to NO in g/kWh (Zeldovich) are to 21, 4, 23.3, and 16 and the Summary Emission (SE) of the (PM and NOx) are 4.2, 4.3, 4.9, and, 3.6 g/kWh. Due to the increasing air swirl in the DSEVL2-BMW-M47T piston bowl, a higher PRR is observed causing increased NOx emissions as compared to all the other as shown by (Seelam et al., 2021).

Table 3 shows that the PBG in each box has the same features that are studied by different investigators and compared with the simulation results. Accordingly, in the Toroidal bowl and its group; the NOx, PM, CO, CP and CT are almost similar except in TRB which is somewhat controversial to the other four. The Hemispherical bowl & similar piston group design have the same result but P2 and MRCCI has less simulation and experimental NOx emissions results. The Re-entrant bowl and similar piston design has acceptable output except maximum NOx of Conventional Diesel Combustion, maximum Particulate Matter of P3 and SCC and maximum Carbon monoxide of REB. But P3 has low CO Production.
from the engine. The DOOSAN P158FE Bowl & double curved piston design have similar result except the Particulate Matter advantage of DOOSAN P158FE (P4) Piston Bowl.

4. Simulation of fuel sprays in the swirling air flow

The Interaction of spray and their near-wall flow with swirl and walls are depicted in the equations below as follows:

\[ l_{ij} = k_j B_{sw} \tau_w^0 + c_{ij} U_j \tau_w d_{32}^{-1.5} \int_0^w (w_i - u) \cos \beta \tau_w \]

\[ \tau_w = \tau_s - \tau_{sw} \]

\[ k_j = \sqrt{\frac{\sin \gamma_1 \sin \gamma_3 + 1.2(1 - \sin \gamma_j) - 2(\cos \gamma_j)}{\cos \gamma_j}} \]

where:
- \( k_j \) - Effect of impingement Angles.
- \( W_j \) - Effect of local swirl velocity
- \( l_{ij} \) - length of NWF spot in each direction (\( j = 1, 2, 3, 4 \)).
- \( \gamma_j \) - the shape of NWF spot depends on impingement angles (\( j = 1, 2, 3, 4 \)).

The elementary fuel mass motion from injection to spray front zone \( l_k \) and spray tip \( l_m \) is shown in the Figure 9. The equation of motion of an Elementary Fuel Mass (EFM) injected during a small time-step and moving from the injector to the spray tip is extremely dependent upon the shape of the piston bowl as shown in Eq. (5).

The current distance, velocity, and the EFM penetration length until the spray stop; and the Momentum conservation defines the mass of entrained air \( \Delta m_a \) for each Elementary Fuel Masses (EFM) is computed in Eq. (5) by (A. Kuleshov and Mahkamov, 2008).

\[ u \frac{\Delta m_f}{\Delta m_a} = u \frac{l_k}{l_m} \]

where: \( l \) is the current distance from EFM to injector;
- \( U = dl/dt \) is the current EFM velocity;
- \( U_{in} \) is the EFM velocity in the injector nozzle; and
- \( l_m \) is the EFM penetration length until the spray stop.

Eq. (6) shows the Momentum conservation defines the mass of entrained air \( \Delta m_a \) for each Elementary Fuel Masses (EFM) \( \Delta m_f \):

\[ u \Delta m_f = u \left( c_1 \Delta m_f + \Delta m_a \right) \]

As shown in Figure 10, the Diesel-RK spray model categorizes the fuel sprayed in the cylinder into 7 (seven) zones. Each zone’s evaporation and burning circumstances are unique. Free spray before and after impingement are the two basic divisions of diesel fuel free spray zones. Before impingement, there is a free spray that has three distinct zones: 1, 2, and 3-Dense axial core, Dense forward front, and Dilute outside the sleeve. Because the near-wall flow created after impingement is not homogenous in structure, density, or temperature, calculating fuel evaporation is problematic; therefore, by similarity with free spray, it is practical to
designate typical zones with averaged heat and mass transfer coefficients in the NWF. A new set of zones is 4, 5, 6, 7 formed after wall impingement: the axial conical core of a NWF, dense core of a NWF on a piston bowl surface, dense forward front of an NWF, and dilute outer sleeve of near-wall flow.

The corresponding zones are taken into account when fuel knockouts the cylinder liner and cylinder head surface. Spray forward front depth in Eq. (7) is computed as follows:

$$b_m = A_m \frac{\nu}{M} \frac{\rho}{C_0}$$

(7)
where: \( Am = 0.7 \) is empirical coefficient Distribution of fuel among the spray zone is defined for each time steps by the following expressions:

In the Core:
\[
\sigma_{Core} = (\sigma_i - \sigma_r)(1 - 0.1k) / l
\]  

(8)

In Front:
\[
\sigma_{Front} = 0.8(\sigma_i - \sigma_r)A
\]  

(9)

In dilute:
\[
\sigma_{dilute} = \sigma_i + 0.2(\sigma_i - \sigma_r) + 0.1(\sigma_i - \sigma_r)0.1k/l
\]  

(10)

In NWF:
\[
\sigma_w = 0.8(\sigma_i - \sigma_r)(1 - A)
\]  

(11)

where: \( A = 1 \) before wall impingement and \( A = 0 \) for after impingement.

The PBD affects the engine efficiency because of that it has different effects on the fuel spray mechanisms like impingement angles, local swirl velocity, length of NWF spot in each direction, the shape of NWF spot depending on impingement angles. Fuel dilution, free core spray fraction, fuel-free spray front, and near-wall effects on the fuel spray mechanisms like impingement angles, local swirl velocity, length of NWF spot in each direction, the shape of NWF spot

Due to the single injection of the first combustion chamber (DSEVL2 BMW M47T) and the fourth combustion chamber (DOOSAN P158FE) engine reduced the injection velocity, the peak HRR-P4 results is 0.035J/C14/CA at the maximum injection velocity since they have single injections. Due to the existence of pilot injection before the main injections, the peak HRR-P2 velocity has increased to 0.061 J/C14/CA, which is some distance ahead of the peak main injection velocity, and the angle of HRR-P2 termination, has grown to 410°. Because of the advanced pilot, the HRR-P3 peak is 0.082J/C14/CA, which is higher than HRR-P1 and HRR-P2, and HRR-P4. This in general shows that the heat release rate increases with multiple increased injections and decreases for the single injections fuels (Verma and Bhosale, 2020; Taşkiran and Ergeneman, 2011).

As it is illustrated in Table 4 and Figure 11; the dilute outside sleeve before impingement in each combustion chamber is higher. Comparison between calculated spray tip penetration and free spray angle for diesel engine with \( d_n = 0.12200 \) mm at fuel pressure of 137.3 bar of different piston bowls are shown in Figure 11 and Figure 12. The free spray contour angle and sprays tip penetration is for the two single fuel injections and the two triple fuel injections. It is possible to conclude that the spray tip penetration of the single injection as compared to the triple injection is more and the main injection has more spray tip penetration as compared to the post and pilot injected fuel. FSP1 is 60 mm and FSP4 is about 39 mm.

The free spray contour angle was calculated by the Eqs (12), (13) shown below:

\[
\tau_a = 2\arctan\left(E_s W_e^{-0.35} M^{-0.07} \frac{\gamma - 0.12}{\gamma^2 - 0.5} \rho^{-0.07} \sqrt{\gamma} \right)
\]  

(12)

\[
\tau_b = 2\arctan\left(F_s W_e^{-0.32} M^{-0.07} \frac{\gamma - 0.12}{\gamma^2 - 0.5} \rho^{-0.5} \sqrt{\gamma} \right)
\]  

(13)

where: \( E_s = 0.0932F_e W_e^{-0.03} \gamma^{0.12} \) & \( F_e = 0.0075/0.009 \) for diesel cylinder conditions.

Figure 11 shows the results of spray angle evolutions when applying injection strategy under the same injection pressure, and
density of the fuel conditions had shown with only different cylinder combustion chamber geometry. The reason for the large spray angle in the initial injection phase because that the fuel was injected into a relatively stationary air environment and the air at the nozzle outlet was squeezed and pushed into the radial direction however, the spray angle remains nearly stable during the later injection stage (Tasirkiran and Ergeneman, 2011). This can be attributed to the increased momentum exchange at higher surrounding gas densities as the fuel droplets are more closely surrounded by environmental gases. It should be noted that the increased ambient density can also reduce the maximum liquid length and better avoid the spray-wall impingement (Zhou et al., 2019; Nadu, 2015). Figures 12 and 13 illustrate the average spray angle under the same condition. The trend of the spray angle is still that it changes drastically during the initial stage of fuel injection. With the increased surrounding gas density, the spray angle tends to rise.

Figure 14 shows that the Main injection spray tip penetration for the triple injection of FSP2 is about 46 and the pilot is 41 and post-injection is 35 mm for the spray penetration but the spray penetration for the FSP3 and FSP4 is maximum at about the beginning of injections for the pilot, main and post-injection due to the resistance of stationary air environment and the squeezed and pushed air into the radial direction at the nozzle outlet (Tasirkiran and Ergeneman, 2011). The spray angle for the triple injection of FSA2 is at about 20 mm STP for the main injection and 15mm STP for the pilot and 15 mm STP for the post-injection but the maximum spray angle in the STP3 for the pilot, main and post-injection are at the spray tip of 10, 15 and 12 mm STP respectively. As compared to the single and multiple injections; the single injection fuel supply method has greater fuel spray tip penetration.

5. Conclusion

In this work, the investigation was by using combustion simulation comparison of piston bowl shapes of DSEVL2 BMW M47T, Shallow Hesselman, Lombardini 15LD350, DOOSANP158FE and their effects were analyzed and validated with the other work and have the following conclusions as follow:

- The simulation model with variation DSDoPG shows that the peak cylinder pressure and Temperature of the Shallow Hesselman are 131bar and 2048k respectively which is higher than all the other combustion chambers and the lower cylinder pressure and temperature in the DOOSANP158FE which are 112bar and 1680k respectively. The simulation result indicates that the temperature is within the acceptable limit in between 1400-2000k except for the piston temperature of 2048.2k.
- The PHRR-P3 is 0.82 with great variation in between maximum and minimum due to pre-and post-injection, the PHRR-P4 is lower due to single injections 0.033J/deg. The HRR in HRR-P3 is higher in 39% J/deg as compared to the minimum one.
- The NOx-P4 of 12.74 is lower than all of the others and NOx-P1 emission is much higher 25.6 g/KWh than the other three because of cylinder pressure, which causes an increase in temperature. Even though; in-cylinder temperature and NOx emission are directly proportional in the combustion chamber, NOx of piston-2 is lower than that of piston-1 and 3 due to the lower fuel fraction of NWF dilution outer the sleeve, low fuel fraction in core of the free spray, low fuel fraction in fronts of the free spray, low fuel fraction in the core of the fuel free spray reduce NOX.
- The Particulate Matter emission in PM-P1 is 0.35, and PM-P2 is 0.43 g/KWh which is higher than all the other. Although there is a substantial decrease in PM, a penalty in NOX is observed for PM-P1. But PM of the P2 is higher after the peak result of emission.

Declarations

Author contribution statement

Habtamu Deresso: Conceived and designed the experiments; Performed the experiments; Analyzed and interpreted the data; Wrote the paper.
Ramesh Babu Nallamothu; Venkata Ramayya & Bisrat Yosef: Performed the experiments; Analyzed and interpreted the data.

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Data availability statement

Data will be made available on request.

Declaration of interest’s statement

The authors declare no conflict of interest.

Additional information

No additional information is available for this paper.

References

Abani, N., Kokjohn, S., Park, S.W., Bergin, M., Munnannur, A., Ning, W., Sun, Y., Reitz, R.D., 2008. An improved spray model for reducing numerical parameter dependencies in diesel engine CFD simulations. SAE Tech. Paper. 2008 (724), 776–796.
Ahmadi, R., Hosseini, S.M., 2018. Numerical investigation on adding/substituting hydrogen in the CDC and RCCI combustion in a heavy duty engine. Appl. Energy 213, 450–468.
Amin, M., Gholinia, M., Pourfalah, M., Akbar, A., 2021. Investigation of the fuel injection angle/time on combustion , energy , and emissions of a heavy-duty dual-fuel diesel engine with reactivity control combustion ignition mode. Energy Rep. 7, 5239–5247.
Banack, B., Service, A.H., 2013. Inject. Data 184 (September), 243–253.
Bawankar, C.S., Gupta, R., 2016. Effects of Piston Bowl Geometry on Combustion and Emission Characteristics on Diesel Engine: A Cfd Case Study.
Ganjil, P.R., Singh, R.N., Raja, V.R.K., Strinivas Rao, S., 2018. Design of piston bowl geometry for better combustion in direct-injection compression ignition engine. Sadhana - Acad. Proc. Eng. Sci. 43 (6), 1–9.
Gopal, R., Kavandappa Goundar, M., Ramasamy, S., Natarajan, N., Ramasamy, V., 2014. Experimental and regression analysis for multi cylinder diesel engine operated with hybrid fuel blends. Therm. Sci. 18 (1), 193–203.
Hanson, R., Curran, S., Wagner, R., Kokjohn, S., Spliter, D., Reitz, R.D., 2012. Piston bowl optimization for RCCI combustion in a light-duty multi-cylinder engine. SAE Int. J. Eng. 5 (2), 286–299.
Jason Miwa, AVL-35, C. R. N., 2021. Advanced Combustion Literature Survey (Issue May).
Karthikeya Sharma, T., Amba Prasad Rao, G., Madhu Murthy, K., 2015. Effective reduction of in-cylinder peak pressures in homogeneous charge compression ignition engine - a computational study. Alex. Eng. J. 54 (3), 373–382.
Kim, D., Ekofo, I., Colban, W.F., Miles, P.C., 2009. In-cylinder CO and UHC imaging in a light-duty diesel engine during PPC low-temperature combustion. SAE Int. J. Fuel. Lubr. 1 (1), 933–956.
Knox, B.W., 2016. End-of-Injection Effects on Diesel Spray Combustion. Ph. D. Thesis, December.
Kuleshov, A.S., 2006a. Use of multi-zone DI diesel spray combustion model for simulation and optimization of performance and emissions of engines with multiple injection. SAE Tech. Paper. 1385.
Kuleshov, A.S., 2006b. Use of multi-zone DI diesel spray combustion model for simulation and optimization of performance and emissions of engines with multiple injection. SAE Tech. Paper. 724.
Kuleshov, A., Mahkamov, K., 2008. Multi-zone diesel fuel spray combustion model for the simulation of a diesel engine running on biofuel. Proc. IME J. Power Energy 222 (3), 309–321.
Kuleshov, Andrey, Mahkamov, K., Kozlov, A., Fadeev, Y., 2014. Simulation of dual-fuel diesel combustion with multi-zone fuel spray combustion model. In: ASME 2014
Li, J., Yang, W.M., Zhou, D.Z., 2016. Modeling study on the effect of piston bowl geometries in a gasoline/biodiesel fueled RCCI engine at high speed. Energy Convers. Manag. 112, 359–368.

Mahesh, G., Yerremangoudar, H., S.S, R.H.M., Prakash, B.B., 2021. CFD simulation of in-cylinder air swirl by modification in piston bowl. Int. J. Eng. Res. Technol. 10 (2), 575–578.

Muralidharan, C.S.R.K., 2021. Effects of partially stabilized zirconia fueled with Borassus biofuel at different piston bowl geometries in LHR engine. J. Therm. Anal. Calorim. 1–17.

Nadu, T., 2015. Modeling the spray characteristics of biodiesel. M.Jayamurugan Appl. Mech. Mater. 25 (2), 846–850, 1.

Naouer, B.P.O.M., Tanguy, A., Cedex, A., 2010. Modelling and Simulation of the Ophthalmology. 金属学报, January.

Nghia, N.T., Khoa, N.X., Cho, W., Lim, O., 2022. A study the effect of biodiesel blends and the injection timing on performance and emissions of common rail diesel engines. Energies 15 (242).

Paykani, A., Kakaei, A.-H., Rahnama, P., Reitz, R.D., et al., 2016. Progress and recent trends in reactivity-controlled compression ignition engines. Int. J. Engine Res. 17 (5), 481–524.

Payri, R., Salvador, F.J., Bracho, G., Viera, A., 2017. Vapor phase penetration measurements with both single and double-pass Schlieren for the same injection event. In: IIsus Europe. 28th European Conference on Liquid Atomization and Spray Systems. September.

Pham, V.V., 2019. Research on the application of diesel-RK in the calculation and evaluation of technical and economic criteria of marine diesel engines using the unified ULSD and biodiesel blended fuel. J. Mech. Eng. Res. Dev. 42 (2), 87–97, Regan, J.W., 2021. Heat release rate characterization of NFPA 1403 compliant training fuels. Fire Technol. 57 (4), 1847–1867.

Seelam, N., Kumar, S., Burra, G., Sivasurya, B., Gadepalli, M., 2021. Investigating the role of fuel injection pressure and piston bowl geometries to enhance performance and emission characteristics of hydrogen - enriched diesel/1 - pentanol fueled in CRDI diesel engine. Environ. Sci. Pollut. Control Ser. 123456789.

Siddique, S.K.A., Vijaya, K., Reddy, K., 2015. Experimental validation and combustion chamber geometry optimization of diesel engine by using diesel –rk. Int. J. Mech. Eng. Technol. 6 (2), 976–6340.

Sreedharan, S.N., Krishnan, R., 2018. Development of tool to design piston bowl considering spray parameters to reduce emissions. IOP Conf. Ser. Mater. Sci. Eng. 396 (1).

Sureshkannan, G., Vellangiri, M., Kalaiyanan, M., Kavin Kumar, K., Ragul, T., Kumar, K.K., 2019. Diesel-Rk Analysis and Experimental Investigation of Single Cylinder Variable Compression Ratio Diesel Engine. May.

Tas¸kiran, O.O., Ergeneman, M., 2011. Experimental study on diesel spray characteristics and autoignition process. J. Combust. 2011.

Verma, A., Bhosale, P., 2020. Model Based Study on Effects of Valve Lifts and Events on the Performance of an Engine. pp. 580–586.

Wu, G., Zhou, X., Li, T., 2019. Temporal evolution of split-injected fuel spray at elevated chamber pressures. Energies 12 (22).

Xu, G., Jia, M., Li, Y., Chang, Y., Wang, T., 2018. Potential of reactivity controlled compression ignition (RCCI) combustion coupled with variable valve timing (VVT) strategy for meeting Euro 6 emission regulations and high fuel efficiency in a heavy-duty diesel engine. Energy Convers. Manag. 171 (May), 683–698.

Ya, A., Koca, E., 2014. Investigation on the performance and emissions of a biodiesel engine fueled with soybean biodiesel and diesel fuel. Tarım Makinaları Bilimleri Dergisi 10 (4), 301–306.

Zhao, W., Zhang, Y., Huang, G., Li, Z., 2020. An experimental study of the injection strategies on engine performance of the butanol/biodiesel dual-fuel Intelligent Charge Compression Ignition mode. Int. J. Engine Res. 800.

Zhou, X., Li, T., Wei, Y., Wu, S., 2019. Scaling spray combustion processes in marine low-speed diesel engines. Fuel 258 (August), 116133.