AIMS Energy

AIMS Energy, 14(3): 618–639.
DOI: 10.3934/energy.2026026
Received: 01 March 2026
Revised: 05 May 2026
Accepted: 13 May 2026
Published: 28 May 2026

https://www.aimspress.com/journal/energy

Research article

Numerical study of the impact of piston bowl geometry on emissions and combustion properties in diesel engine fueled with aqueous ammonia and diesel mixture

Hussein A. Mahmood[1,*], Osam H. Attia[1] and Ali O. Al-Sulttani[2]

[1] Department of Reconstruction and Projects, University of Baghdad, Baghdad 10071, Iraq
[2] Department of Water Resources Engineering, College of Engineering, University of Baghdad, Baghdad 10071, Iraq

* **Correspondence:** Email: husseinadel@uobaghdad.edu.iq; Tel: +9647739695778.

Abstract: Numerous research studies have been conducted to enhance fuel economy and reduce emissions by converting diesel engines to run on aqueous ammonia-diesel (AAD) blends. However, only a few studies have investigated the effect of the piston bowl geometry on combustion efficiency and emissions from AAD blends. This study investigated the effect of piston bowl geometry on the combustion efficiency of a diesel engine fueled with AAD blends. Five different types of piston bowls, namely the Double lip (Case 1), 10D100 (Case 2), Mexican hat (Case 3), Inveco F1C Rollbuch (Case 4), and Peugeot DW 10 (Case 5), were used. Diesel-RK software was used for modeling and simulating the fuel combustion inside the diesel engine. Based on the numerical findings, with the same combustion chamber, the results show that the NOx emission, smoke level, and peak pressure reduced with increased engine speed from 2000 to 3000 rpm, whereas peak temperature, CO_2, and PM emissions increased. Moreover, among all designs, Case 5 achieved the highest combustion performance, with peak pressure and temperature, along with the lowest PM and smoke. However, it produced the highest NOx emissions. In contrast, Case 1 yielded the lowest NOx, peak pressure, and temperature, but the highest PM, indicating poor combustion. Case 3 achieves the second-highest peak pressure (53.13 bar at 2000 rpm and 52.61 bar at 3000 rpm) and second-highest temperature (1886.8 K at 2000 rpm and 1914 K at 3000 rpm), and only moderately elevated NOx. Case 3 offers the best overall emission balance while maintaining high combustion efficiency, making it the most suitable piston bowl geometry for sustainable operation with AAD blends.

Keywords: aqueous ammonia and diesel mixture; piston bowl geometry; Diesel-RK; combustion efficiency

1. Introduction

In an era of decreasing fossil fuel sources and the harmful influence associated with their combustion products on the environment, people are increasingly interested in generating energy from various forms of natural resources, such as wind and solar. Nevertheless, the energy obtained from these resources is finite and fluctuates according to the seasons, locations, and environmental factors. Clean energy sources such as ethanol, biodiesel, nitrogen, and natural gas are of interest to manufacturers and researchers. Alternative fuels give reciprocating engines the chance to continue developing unhindered and to be modified to meet contemporary environmental protection standards in the transportation and energy sectors [1–3]. Diesel engines are more common than other combustion engine types because of their great thermal performance, high torque, longevity, and fuel economy, especially in the power generation, transportation, and agricultural sectors. Still, diesel engines are harmful to the environment due to the elevated levels of poisonous exhaust gases such as nitrogen oxides (NOx), hydrocarbons (HC), and particulate matter (PM) [4–6]. Meanwhile, numerous researchers and producers have looked into diverse strategies to address the emissions issue, encompassing engine modification, exhaust gas treatment, and fuel alteration [7,8].

Fuel modification is a viable strategy for achieving diminished emissions in diesel engines, as it directly influences the combustion of the air–fuel combination that produces the mechanical power required to operate the engine shaft. Fuel modification involves various modes, such as multi-fuel mixing, water mixing, and the incorporation of additives, with the aim of achieving certain benefits [9–11]. The concept of water combined with fuel, known as emulsified fuel, has raised considerable interest among scientists because of its ability to reduce the use of primary fuel, e.g., diesel or biodiesel, without affecting engine functionality or raising emissions. In addition, water in the combustion chamber decreases the temperature, thus decreasing the unnecessary formation of NOx and PM [12,13]. Oliveira and Brójo [14] studied the effect of 8% water addition in diesel fuel as a Water-in-Diesel Emulsion (WiDE) on the combustion and emission of a diesel engine. Results showed an average of 7.57% improvement in specific fuel consumption (SFC), 19.14% improvement in thermal efficiency (TE), 5.54% reduction in carbon dioxide, 20.50% reduction in nitric oxide (NO), and 75.19% reduction in smoke levels. However, the mean increases in carbon monoxide (CO) and hydrocarbon (HC) emissions were 81.09% and 93.83%, respectively. Hasannuddin et al. [15] conducted 200 h of engine operation with water-diesel emulsion fuels. They found that NOx and smoke were reduced by 51% and 14%, respectively, and CO and BSFC increased. Watanabe et al. [16] tested the diesel engine with two types of emulsion fuels with 10% and 20% concentration of water. The tests showed that E20 fuel had reduced NOx emissions and smoke, which was associated with particles of water vapor absorbing the flame heat. In addition, CO and SFC of E10 and E20 were higher than those of the diesel, which was attributed to the OH radicals of water converting carbon into CO.

Carbon-neutral fuels constitute a practical and effective strategy for alleviating the environmental impact of diesel engine emissions [17–19]. The unique properties of ammonia make it an attractive alternative fuel source. Its carbon-free content and the ability to reduce NOx emissions make it a desirable substitute for conventional fuels in compression ignition engines. Ammonia fuel is recognized to be an environmentally friendly alternative to hydrogen, which addresses the issues of mobility and storage of hydrogen with its large mass, thermal dispersion, and low boiling point [20,21]. Also, the ammonia fuel contains approximately 17.6% of hydrogen by weight and is considered as being a hydrogen carrier [22,23]. Consequently, ammonia fuel represents a feasible alternative to hydrocarbon fuels. Furthermore, ammonia fuel is a crucial component of the selective catalytic reduction (SCR) system, which aims to diminish NOx emissions during diesel engine combustion. This is achieved through the chemical reaction of ammonia (NH_3) with NOx, yielding H_2O and N_2 [24–26]. Nadimi et al. [27] conducted experimental research on the effects of substituting diesel fuel with ammonia on emission levels and combustion efficiency in an ammonia/diesel dual-fuel engine. The study findings indicated that ammonia can supply 84.2% of the input energy, while demonstrating that thermal efficiency improves with greater diesel replacement. Furthermore, the findings demonstrated that adding ammonia raised NOx emissions and unburned ammonia (14,800 ppm) while decreasing CO, CO_2, and PM emissions. Reiter and Kong [28] examined the influence of diesel fuel and ammonia mixing on the combustion performance of a diesel engine, maintaining constant engine power while varying the diesel/ammonia mixing ratios between 40% and 60% diesel and 60% and 40% ammonia. The findings indicated that, compared with pure diesel-fuel engines, dual engines exhibited lower exhaust gas emissions of hydrocarbons (HC), nitrogen oxides (NOx), and carbon monoxide (CO). The substantial use of ammonia also diminished soot emissions owing to the low carbon content of the dual fuel. Huang et al [29] numerically investigated ammonia–diesel dual-fuel combustion in a CI engine using 3D CFD simulations, concluding that ammonia is predominantly consumed within the diesel spray plume, while slip occurs in regions not reached by the spray, highlighting the need to enhance flame propagation to reduce emissions.

Given the advantages of emulsion and ammonia fuels, the blending of the two fuel systems has received a lot of interest. Studies on the viability of emulsion fuel–ammonia blends in internal combustion engines are currently being conducted, and the concentration of ammonia that can be safely added to the combustor has not been defined yet. However, the low rate of flame propagation and the high auto-ignition temperature of ammonia pose combustion inefficiencies that may lead to a compromise in the use of ammonia in diesel engines. Recent research has demonstrated that aqueous ammonia can be used to mitigate some of these problems in combination with diesel, but to realize ideal combustion characteristics, complex processes are still required [30,31]. Pyrc et al. [32] carried out experimental research to identify the influence of the replacement of 25% water ammonia solution with diesel fuel on the operating properties and emissions of the compression ignition dual-fuel engine under different operating conditions. The findings indicated that combustion with ammonia reduces the stability of the test engine, though it remains within the acceptable stability indices of the reciprocating combustion engines. Moreover, the introduction of water ammonia solution induced higher emission of hydrocarbons and carbon monoxide, and reduced emission of nitrogen oxides. Al-Dawody et al. [31] examined the effect of using three different volumetric percentages of ammonia solution in mixture with diesel fuel (40% NH_4OH + 60% diesel, 50% NH_4OH + 50% diesel, and 60% NH_4OH + 40% diesel) on the efficiency of diesel engine combustion and emission of pollutants in dual-mode operation. The study concluded that the addition of ammonia solutions inhibited heat emission and pressure of the system and prolonged ignition delay. In addition, the analysis showed that the level of nitrogen oxides and soot emissions significantly decreases. Sarıtaş et al. [33] experimentally investigated aqueous ammonia (10–20 wt.%), hydrogen (10% energy), and diesel blends in an 11 L heavy-duty CI engine at 660 rpm, concluding that A20D80 maximizes brake thermal efficiency (39.18%), while NH_4OH reduces CO/HC, and hydrogen mitigates NO at low/medium loads, emphasizing the need for optimal ammonia–hydrogen balancing.

Many studies have been conducted to increase fuel efficiency and reduce pollutant emissions through adjusting diesel engines to use aqueous ammonia-diesel blends [30,31]. However, there is a deficiency of studies investigating the role of the geometry of piston bowls on the combustion performance and emissions of diesel engines that are fueled with aqueous ammonia–diesel mixtures. Piston bowl geometry is very critical in the process of combustion in diesel engines. It has a direct effect on the air–fuel mixing, propagation of the flames, and heat release patterns of the combustion chamber [34,35]. The design of the piston bowl could either enhance or decrease the combustion performance and the level of emissions, whenever applied with aqueous ammonia–diesel mixtures. The piston bowl's design is essential for mixing an engine's air and fuel effectively. The mixture of fuel and air needs to be sufficient to offset the influence of fuel-rich regions and enable the engine to achieve its performance and emissions objectives, which is a fundamental goal in the design of the combustion chamber. Turbulence in the airflow within the combustion bowl can be utilized to enhance the mixing process and to achieve this goal. Through a suitable construction of the bowl in the piston crown, the swirl caused by the intake port can be augmented, or the squish generated by the piston as it reaches the cylinder head to create greater turbulence during the compression stroke. Rifat et al. [36] simulated the effects of three piston bowl geometries on combustion, performance, and UHC/VOC emissions in a diesel–methane dual-fuel engine with ANSYS Forte 2023 R1 and found that the modified re-entrant design had the highest thermal efficiency (34.11%) and lowest UHC/VOC emissions, but marginally higher CO because of delayed oxidation. Sefatjoo and Khaleghi [37] performed a coupled CFD study of four piston bowl configurations and swirl ratios (0–3) on spray dynamics and fuel–air mixing, and found moderate swirl (SR-2) to maximize effective penetration and uniformity of the mixture, and high swirl to create distorted jets and a smaller usable combustion volume.

Only a limited number of studies have dedicated their research to examining the influence of piston bowl geometry on combustion efficiency and emissions in diesel engines powered by aqueous ammonia-diesel blends. This study focuses on investigating the relation between aqueous ammonia-diesel blends and piston bowl geometry to address combustion and emission challenges in diesel engines fueled by diesel fuel blended with constant volumetric percentages of aqueous ammonia using the simulation software Diesel-RK. The given study fills the gap by examining the impact of piston bowl geometry on the dynamics of combustion and the emission profile of a diesel engine working with an aqueous ammonia-diesel mixture. This research aims to find a combination of piston-bowl configuration and critical performance indicators by varying the configurations systematically and determining those that optimize combustion efficiency and, at the same time, reduce environmental impacts. The findings will contribute to the development of sustainable and high-performing diesel engines compatible with alternative fuels.

2. Materials and methods

This study employed the Diesel-RK simulation program to analyze the impact of piston bowl design on the combustion procedure and emission generation characteristics of a diesel engine powered by aqueous ammonia-diesel blends. The Diesel-RK software is specially designed as a simulation and modeling application for modeling internal combustion engines. In 1981–1982, Bauman State Technical University in Moscow created the DIESEL-RK program for combustion engines (piston engines). It was mainly developed to simulate and optimize the operation of commercial internal combustion engines with various upgrades and enhancements. Diesel-RK software is employed to determine emission levels, combustion, and performance characteristics for a diesel engine [38,39]. In the current study, five different types of piston bowls were used, namely the Double Lip (Case 1), 10D100 (Case 2), Mexican Hat (Case 3), Inveco F1C Rollbuch (Case 4), and Peugeot DW 10 (Case 5). Table 1 shows the coordinates of 16 points on the Y and R axes for designing each combustion chamber, while Figure 1 illustrates the dimensions of the shapes for the five piston bowl designs. The chosen piston bowl geometries are open and re-entrant with various lip radii and bowl depths, which directly impact in-cylinder flow properties, including swirl strength, squish movement, and fuel–air mixing, as these are the basic parameters of combustion efficiency and formation of emissions in compression ignition engines. Internal combustion engine research has determined that the bowl geometry influences ignition delay, heat release behavior, and NOx/soot formation by influencing spray development and turbulent mixing. In addition, the majority of the chosen geometries are based on configurations that were reported in scientific literature and real engines [40,41].The choice will facilitate a methodical evaluation of the impact of piston bowl geometry changes on in-cylinder flow, combustion behavior, and emission properties, and will enable the identification of configurations that maximize fuel/air mixing, combustion efficiency, and reduce NOx and soot emissions in ammonia-diesel engines. This study involved a four-stroke, single-cylinder, direct-injection, naturally aspirated, water-cooled diesel engine. Table 2 shows the engine specifications. In all simulations, the piston bowl capacity, compression ratio of 15.5, and injected fuel mass of 0.03834 g/cycle at 2000 rpm and 0.03816 g/cycle at 3000 rpm were maintained constant with the different piston shapes. This guarantees that results are exclusively reliant on the effect of piston bowl form on combustion characteristics. The comparative analysis should help to clarify the influence of piston shape variations on a variety of characteristics and parameters and provide key factors and core features that are relevant to the design of piston bowl geometries.

The volumetric composition used in this study, as per the literature review, was 50% aqueous ammonia and 50% diesel fuel [31]. Table 3 describes the characteristics of diesel, aqueous ammonia (NH_4OH), and a combination of 50% NH_4OH and 50% diesel. The simulations have been done under the Diesel-RK using well-specified sub-model settings. A multi-zone combustion model was used to represent the combustion process; this model splits the in-cylinder charge into zones to resolve the fuel-air mixing, temperature, species concentration, and combustion development. Further, ignition delay was determined by applying the Tolstov mechanism, which approximates the time taken between fuel injection and initiation of combustion depending on pressure and temperature conditions. The heat release rate was also modeled using the Wiebe function, which is an empirical model commonly used to describe the phasing and duration of combustion. To model the emissions, the nitrogen-oxygen reactions of the high-temperature formation of NO were predicted by the thermal Zeldovich mechanism [42]. The estimation of PM was based on the Alkidas formula, which links the formation of soot to combustion parameters, including the equivalence ratio and temperature. The Bosch and Hartridge correlations were used to measure smoke emissions by correlating exhaust opacity and the soot concentration [43,44]. Furthermore, the heat transfer between the in-cylinder gases and the walls was also modeled through the Woschni correlation, which approximates convective heat transfer based on the cylinder pressure, temperature, and velocity of the gases. Turbulence in Diesel-RK is captured implicitly in the spray and air-fuel mixing processes through semi-empirical correlations, and not by an explicit turbulence model such as k-ε. Turbulent processes are modeled in terms of spray penetration, droplet breakup, air entrainment, and diffusion, which govern mixing and combustion processes [43,44]. A time step of 0.5 crank-angle (CA) was used in all simulations in order to resolve combustion phasing and emission formation events accurately. The models were used consistently in all cases of the simulation so that a reliable comparison of the results was possible.

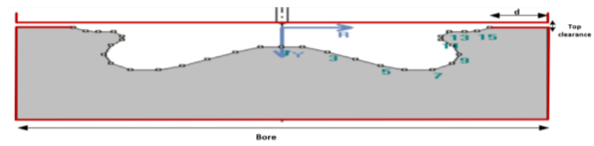
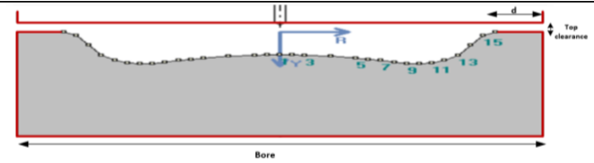
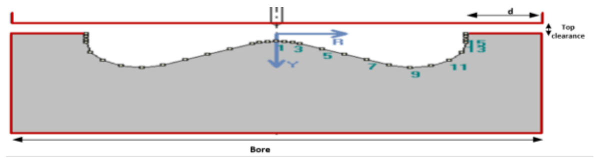
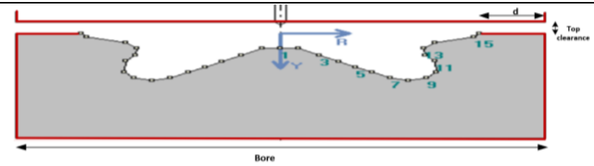
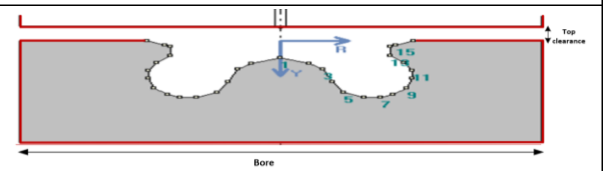
Cases	Name	Shapes of a piston
Case1	Double lip	
Case 2	10D100	
Case 3	Mexican hat	
Case 4	Inveco F1C Rollbuch	
Case 5	Peugeot DW 10	

Figure 1. Various piston bowl geometries used for examination.

Table 1. Coordinates of sixteen points on the Y and R axes for the drawing of each combustion chamber.

	Case 1		**Case 2**		**Case 3**		**Case 4**		**Case 5**	
points/axes	**Y(mm)**	**R(mm)**	**Y(mm)**	**R(mm)**	**Y(mm)**	**R(mm)**	**Y(mm)**	**R(mm)**	**Y(mm)**	**R(mm)**
point 1	0	6.95	0	6.98	0	2.34	0	4.42	0	5.12
point 2	3.16	6.95	1.93	7.03	1.25	2.43	2.66	4.42	4.48	6.69
point 3	6.95	8.84	3.86	7.13	2.46	2.72	5.89	6.53	6.43	8.47
point 4	11.16	11.37	7.73	7.46	3.62	3.18	8.74	8.52	7.98	12.22
point 5	14.95	13.68	11.59	8.02	7	4.85	11.37	10.11	9.73	15.27
point 6	18.53	15.16	13.53	8.37	10.39	6.52	13.89	11.58	12.85	16.75
point 7	22.74	15.16	15.46	8.73	13.77	8.19	16.59	13.22	15.37	16.84
point 8	25.68	12.84	17.39	9.2	17.16	9.87	19.37	14.16	17.33	15.95
point 9	26.74	9.68	19.32	9.69	20.25	10.68	22.11	13.26	19.37	14.11

Continued on next page

	Case 1		Case 2		Case 3		Case 4		Case 5	
points/axes	**Y(mm)**	**R(mm)**	**Y(mm)**	**R(mm)**	**Y(mm)**	**R(mm)**	**Y(mm)**	**R(mm)**	**Y(mm)**	**R(mm)**
point 10	25.68	6.32	21.26	9.76	23.42	10.2	23.67	11.66	20.21	11.16
point 11	24	4.42	23.19	9.45	26.14	8.51	23.37	8.21	20.21	8.84
point 12	24	3.16	25.12	8.54	27.97	5.88	21.89	6.53	18.95	6.53
point 13	25.26	1.47	27.05	7.06	28.61	2.74	21.68	4.42	16.84	4.42
point 14	27.58	1.47	28.99	4.06	28.61	1.83	23.37	2.74	16.84	1.68
point 15	29.47	1.26	30.92	0.96	28.61	0.91	29.31	1.05	17.68	1.26
point 16	31.37	0	32.85	0	28.61	0	30.15	0	20.44	0
d (mm)	8.63		7.15		11.39		9.85		19.56	

Table 2. Specifications of the engine [45].

Model	Diesel engine (Kiloskar)
Kind	Single-cylinder, 4-stroke
Rated output	3.7 KW
Compression ratio	15.5
Bore × stroke	80 mm × 110 mm
Duration of injection	30 CA
Injection pressure	160 bar
Injection timing	20 CA BTDC

Table 3. Properties of diesel and aqueous ammonia [31].

Property	**Iraqi diesel**	**100% NH_4OH (with 25% NH_3) aqueous ammonia**	**50% NH_4OH + 50% diesel**
Molecular weight (g/mol)	190	35.046	112.523
Density (kg/m³)	830	890.3	860.15
Viscosity (cP)	2.5	0.475	1.49
Surface tension (mN/m)	25	55	40
Flash point (°C)	70	Non-flammable	50
Lower heating value (LHV) (MJ/kg)	45.83	9.83	27.83
Cetane number	53.4	--------	26.7
Autoignition temperature (°C)	210	~630	350
Stoichiometric air–fuel ratio	14.33	2.93	8.63
Heat of evaporation (kJ/kg)	250	2034.25	1142.125
Mass fraction of carbon (%)	0.87	0	0.435
Mass fraction of hydrogen (%)	0.126	0.142	0.134
Mass fraction of oxygen (%)	0.004	0.457	0.2305
Sulfur fraction (%)	0.2	0	0.1
Activation energy	22	70–80	50

3. Validation

A sample of the simulation findings, displayed by heat release ratio (HRR) and pressure, has been validated and confirmed against the study conducted by Mohamed F. Al-Dawody [31]. The results were validated using the identical fuel characteristics and operating conditions, which comprised a 1500 rpm engine speed, a 15.5 compression ratio, and a diesel fuel. Figure 2 depicts the pressure history validation using crank angle. Furthermore, Figure 3 compares the HRR observed in this study to that found in previous research. Moreover, Table 4 presents a comparison of the brake power, peak temperature, maximum NO emission, and soot concentration obtained in this study with those reported by Mohamed F. Al-Dawody. As shown in Figure 2 and Table 4, the combustion chamber pressure, brake power peak, temperature, maximum NO emission, and soot concentration observed in this study are within 5% of those found in Mohamed F. Al-Dawody's study. Furthermore, as demonstrated in Figure 3, the HRR values observed are reasonably consistent with those obtained from [31]. Furthermore, the figures show the same pattern with a minor divergence, which is encouraging enough to demonstrate the reliability of the Diesel-RK simulation software.

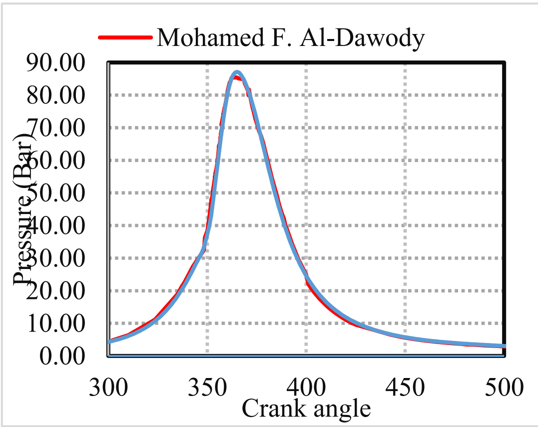


Figure 2. Validation of pressure history relative to crank angle.

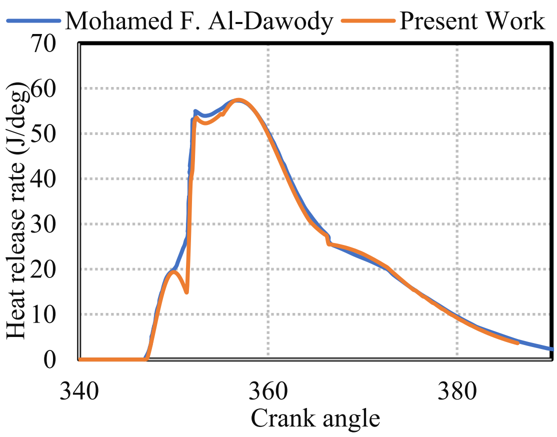


Figure 3. Comparison of the heat release rate (HRR) obtained in this study with that reported by Mohamed F. Al-Dawody [31].

Table 4. Comparison of the brake power, temperature, NO emission, and maximum soot concentration obtained in this study with those reported by Mohamed F. Al-Dawody [31].

Parameter	**Findings according to Mohamed F. Al-Dawody's study**	**Numerical findings in the present study**	**Error percentage**
Brake power (kW)	4.625	4.6006	0.52%
Temperature (K)	2040	1996.30	2.14%
NO emission (g/kWh)	18	17.4	3.33%
Maximum soot concentration (g/cm^3)	6.33	6.324	0.094%

4. Results and discussion

Five different types of piston bowls, namely the Double lip (Case 1), 10D100 (Case 2), Mexican hat (Case 3), Inveco F1C Rollbuch (Case 4), and Peugeot DW 10 (Case 5), were used to study the impact of combustion chamber shapes on the combustion performance and emission formation characteristics of a diesel engine fueled with aqueous ammonia-diesel. The Diesel-RK simulation program was utilized. The simulation findings for the combustion characteristics of diesel fuels with five different types of piston bowls are presented below.

4.1. Pressure

Figures 4 and 5 demonstrate the correlation between the pressure within the various combustion chambers and the crank angle at engine speeds of 2000 and 3000 rpm. Figure 6 depicts the effect of combustion chamber geometries on the value of the peak pressure within the engine. The numerical findings revealed that the combustion pressure was quite different in the six cases, indicating changes in piston design and the combustion performance of the 50% NH_4OH and 50% diesel fuel mixture. The in-cylinder pressure is strongly influenced by piston bowl geometry due to its effect on air–fuel mixing and turbulence inside the combustion chamber. Performance (power output) is strongly influenced by in-cylinder pressure and the homogeneity of the air–fuel mixture. High pressure improves fuel atomization and distribution, which helps achieve more complete combustion. Still, if the air–fuel mixture is not well distributed or homogeneous, local rich zones can form, leading to emissions. The key to achieving low emissions lies not only in high pressure and temperature but also in the homogeneity of the air–fuel mixture. Figures 4, 5, and 6 show that the maximum pressure value drops as the engine speed increases under the same combustion chamber. Moreover, according to the results, Case 5 (Peugeot DW 10) achieved the highest pressure of 53.86 bar at 2000 rpm and 52.67 bar at 3000 rpm, closely followed by Case 3 (Mexican hat) at 53.13 bar at 2000 rpm and 52.61 at 3000 rpm, while Case 1 (Double lip) exhibited the lowest pressure of 52.35 bar at 2000 rpm and 50.96 bar at 3000 rpm. In Case 5, the Peugeot DW 10 design generates intense turbulence, elevating peak pressure and improving combustion efficiency. This is critical for ammonia-diesel blends, which require thorough atomization to compensate for ammonia's slower combustion kinetics. On the other hand, Case 1, including a bowl with two lips, reduces compression pressure due to the fact that fuel and air do not mix completely, therefore creating a weaker combustion. Case 3 (Mexican Hat) was characterized by equal compression and swirl flows, maintaining moderate pressure, ideal for stable combustion without excessive thermal stress. Increased pressure tends to increase efficiency but has to be controlled so that it does not generate too much heat, which worsens NOx emissions.

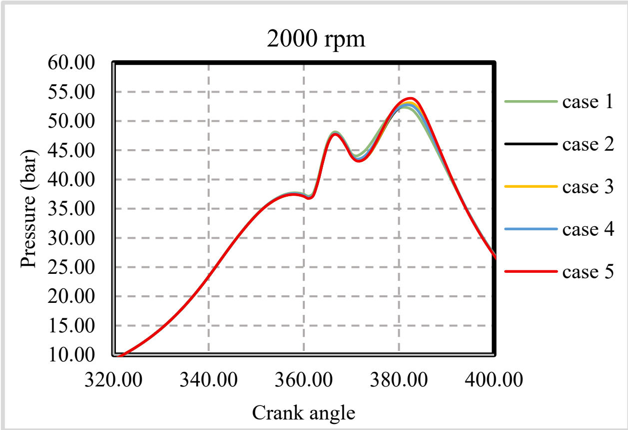


Figure 4. Association between pressure and crank angle at different combustion chambers and an engine speed of 2000 rpm.

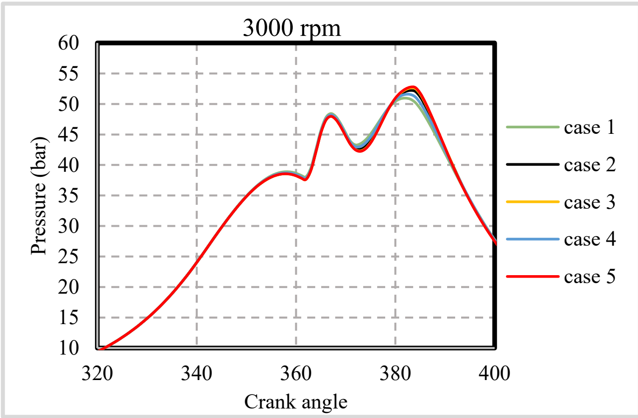


Figure 5. Association between pressure and crank angle at different combustion chambers and an engine speed of 3000 rpm.

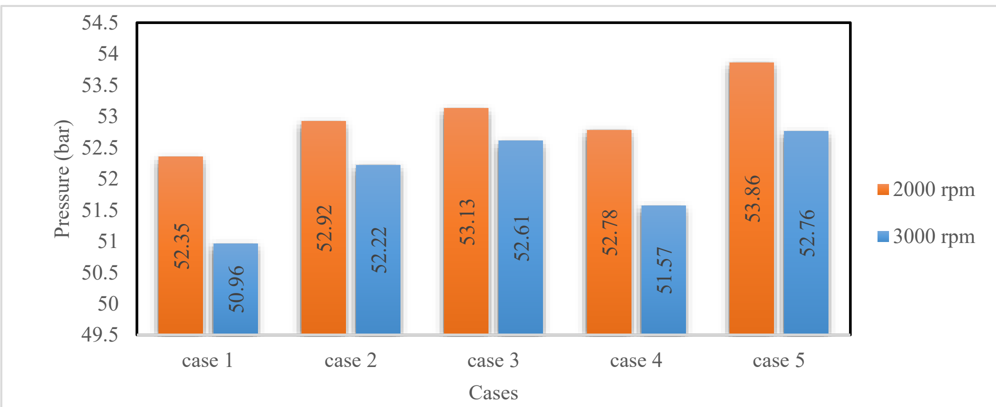


Figure 6. Relationship between pressure and the different combustion chambers.

4.2. Temperature

The relationship between temperatures in the different combustion chambers and the crank angle at an engine speed of 2000 and 3000 rpm is depicted in Figures 7 and 8. Figure 9 explains the impact of combustion chamber shapes on the temperature inside the engine. Numerical analysis demonstrated that the combustion temperatures of six piston designs were noticeably variable when operated with a mix of 50% NH_4OH and 50% diesel. This is because the piston design influences the efficient mixing of the air–fuel mixture and increases flow turbulence in the combustion chamber, improving the combustion performance and total combustion efficacy. Combustion quality is strongly influenced by in-cylinder temperature, as well as the homogeneity of the air–fuel mixture. High temperature accelerates chemical reactions, promoting better oxidation of fuel. As shown in Figures 7, 8, and 9, under the same combustion chamber, the results explain that the value of the temperature rises as the engine speed increases. Combustion temperature findings showed that Case 5 (Peugeot DW 10) had the greatest temperature (1908 K) at 2000 rpm engine speed and 1915 K at 3000 rpm engine speed, subsequent to Case 3 (Mexican hat), with 1886.8 K at 2000 rpm engine speed and 1914 K at 3000 rpm engine speed. Case 1 (Double lip) had the lowest temperature (1857.9 K) at 2000 rpm engine speed and 1864 K at 3000 rpm engine speed. The elevated temperatures in Case 5 correlate with their superior heat release rates and combustion pressures, suggesting that their piston geometries promote efficient fuel–air mixing, heat retention, and optimize flame propagation. Conversely, Case 1's "Double lip" piston underperformed, possibly due to incomplete combustion, resulting in lower thermal energy. Temperature management is vital for ammonia blends, as excessive heat negates their NOx-reduction potential, while insufficient heat risks incomplete combustion.

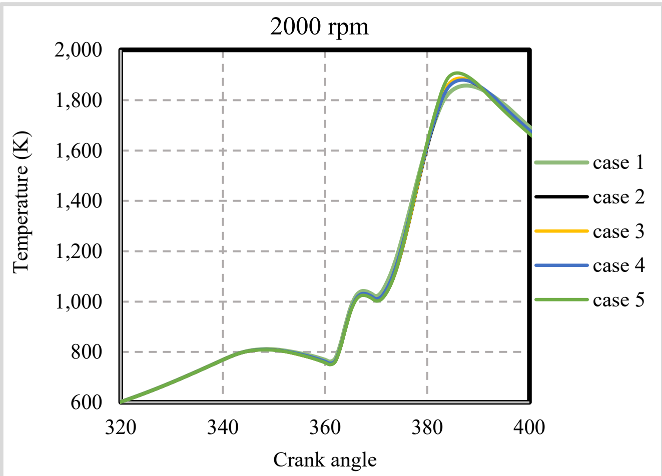


Figure 7. Association between temperature and crank angle at different combustion chambers and an engine speed of 2000 rpm.

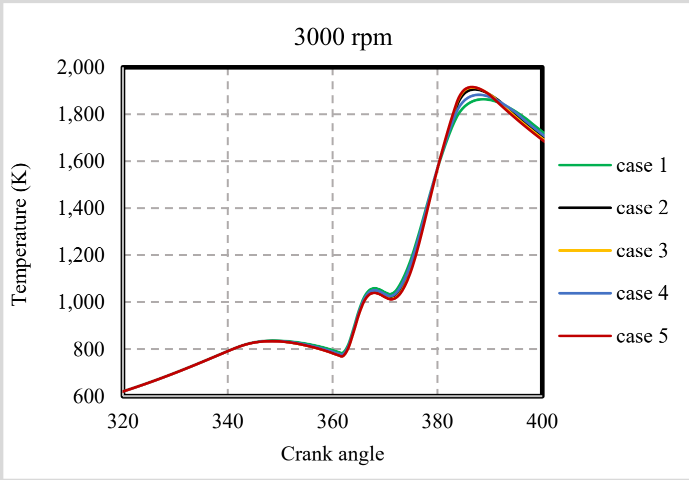


Figure 8. Association between temperature and crank angle at different combustion chambers and an engine speed of 3000 rpm.

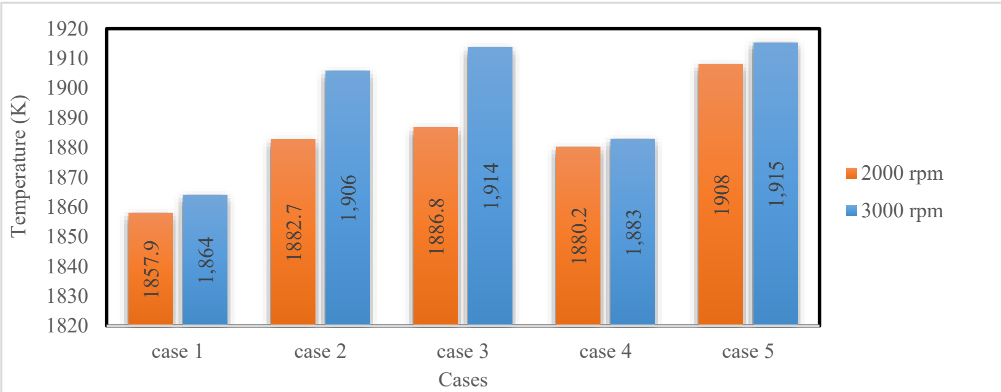


Figure 9. Relationship between temperature and combustion chamber shapes.

4.3. Heat release rate (HRR)

Figures 10 and 11 show the correlation between heat release rates and the crank angle at the various combustion chambers. Figure 12 explains the impact of combustion chamber shapes on the heat release rate. HRR reflects combustion phasing and energy release dynamics. As shown in Figures 10, 11, and 12, under the same combustion chamber, the value of the heat release rate increases as the engine speed increases. Numerical findings indicate that HRR correlates with combustion phasing and energy release dynamics. Case 5 (Peugeot DW 10) demonstrates a pronounced HRR peak of 73.68 J/deg at 2000 rpm and 75.3 at 3000 rpm, attributable to significant turbulence resulting in accelerated combustion. In contrast, Case 1 (Double Lip) shows a delayed and lower HRR (65.28 J/deg) at 2000 rpm and 65.6 J/deg at 3000 rpm, likely due to reduced turbulence and incomplete combustion. Case 3 (Mexican Hat) offers a balanced HRR of 71.8 J/deg at 2000 rpm and 74.6 at 3000 rpm, which is more appropriate for ammonia's slower combustion rate, guaranteeing a continuous energy release without sudden pressure surges. This balance is necessary to extend the life of the engine and reduce emissions in a dual-fuel setup. Figure 13 shows simulation results of spray formation in the combustion chamber in six different piston-bowl geometries using a fuel mixture of 50% NH_4OH and 50% diesel at 2000 and 3000 rpm engine speeds.

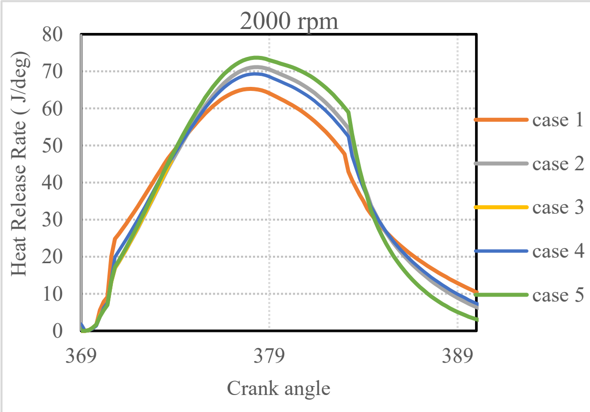


Figure 10. Association between HRR and crank angle at different combustion chambers and an engine speed of 2000 rpm.

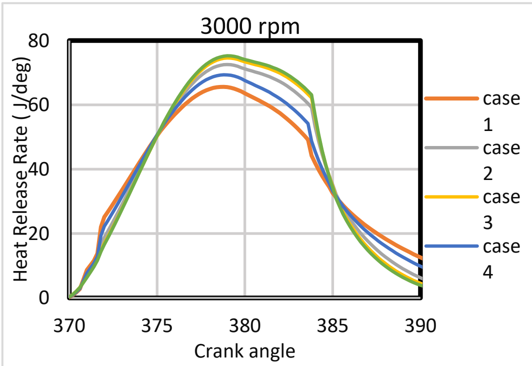


Figure 11. Association between HRR and crank angle at different combustion chambers and an engine speed of 3000 rpm.

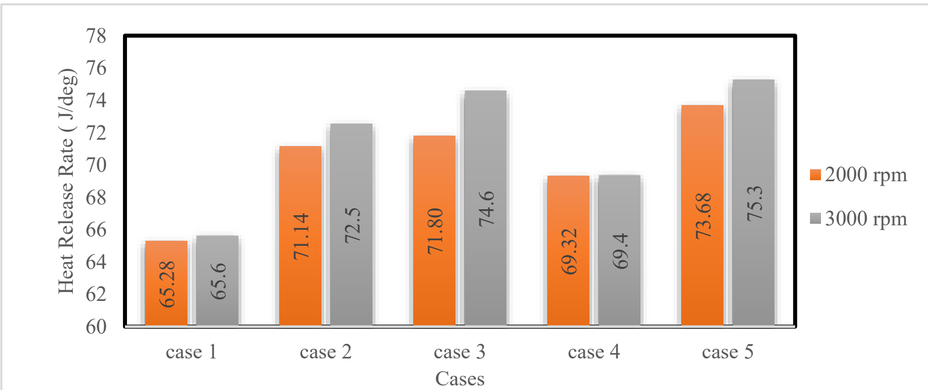


Figure 12. Correlation between HRR and combustion chamber shapes at speeds of 2000 and 3000 rpm of an engine.

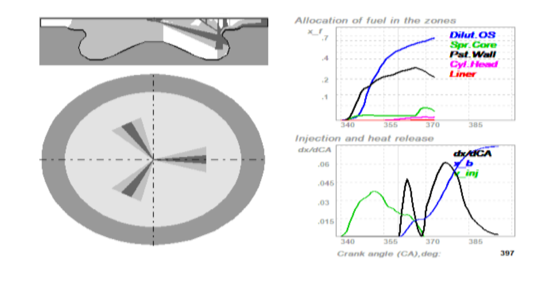
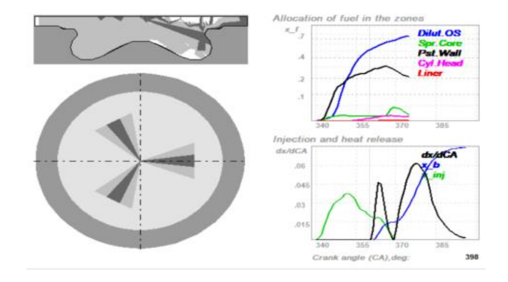
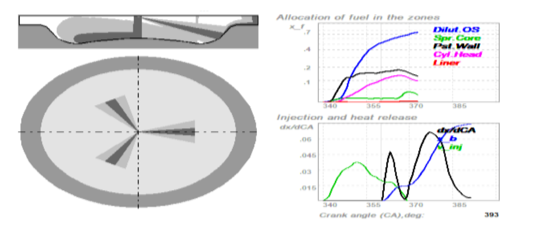
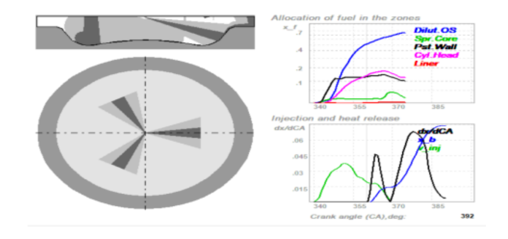
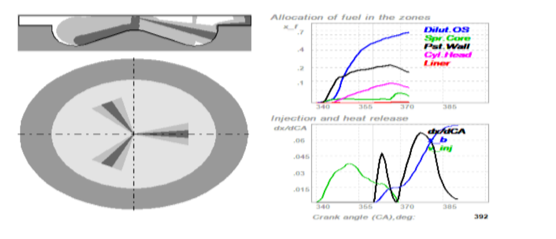
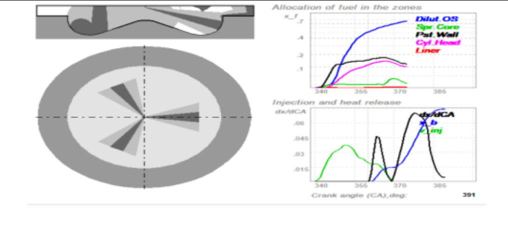
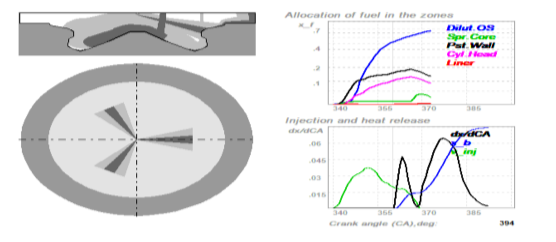
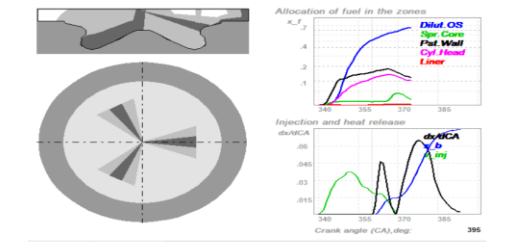
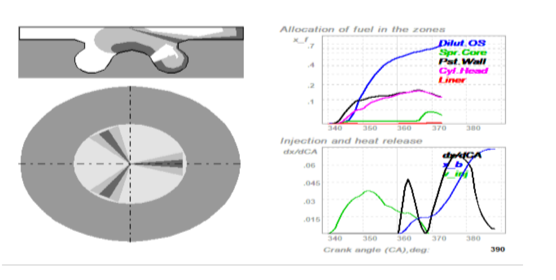
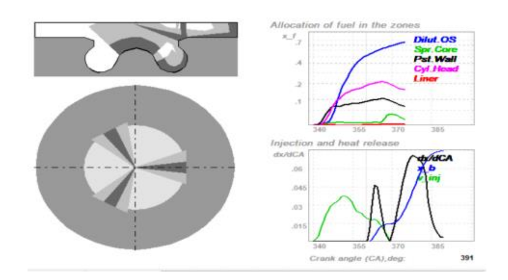
	2000 rpm	3000 rpm
Case 1 (Double lip)		
Case 2 (10D100)		
Case 3 (Mexican hat)		
Case 4 (Inveco F1C Rollbuch)		
Case 5 (Peugeot DW 10)		

Figure 13. Spray formation of the combustion chamber at six different piston bowl geometries at an engine speed of 2000 and 3000 rpm using a blend of 50% NH_4OH and 50% diesel fuel.

4.4. Nitric oxide emission (NO)

The level of nitric oxide formed depends on the highest temperature within the engine, retention time, and the oxygen level. The process of forming NO is very dependent on the temperature of combustion, which also depends on the degree of turbulent mixing in the chamber and the air–fuel mixture. The piston design affects the effective mixing of the air–fuel mixture and flow turbulence in the combustion chamber, which in turn influences combustion performance and overall combustion effectiveness. Moreover, a lower combustion temperature reduces the NO percentage in the combustion chamber, while the reverse is true. Figure 14 illustrates the impact of combustion chamber shapes on the nitric oxide emissions when operated with a mix of 50% NH_4OH and 50% diesel. Figure 14 shows that the concentration of nitric oxide emissions decreases as the engine speed increases for the same fuel type in the combustion chamber. Moreover, the NO emissions results across the five different cases (Double Lip, 10D100, Mexican Hat, Inveco F1C Rollbuch, and Peugeot DW 10) indicate apparent variations in nitrogen oxide formation under similar engine operating conditions (2000 rpm). As shown in Figure 14, findings regarding NO emissions indicate that Case 5 (Peugeot DW10) had the greatest NO emission (2.49 g/kWh at 2000 rpm and 1.99 g/kWh at 3000 rpm), followed by Case 3 (Mexican Hat), with 2.40 g/kWh at 2000 rpm and 1.974g/kWh at 3000 rpm. Case 1 (Double Lip) had the lowest NO emission (2.31 g/kWh at 2000 rpm and 1.819 g/kWh at 3000 rpm). The NO emissions results for Case 2 and Case 4 are 2.35 g/kWh at 2000 rpm and 1.93g/kWh at 3000 rpm, and 2.344 g/kWh at 2000 rpm and 1.871 g/kWh at 3000 rpm, respectively. The increased level of NO emissions in Case 5 is associated with high heat release rates and the combustion temperature. On the other hand, the performance of Case 1 in the Double Lip piston was poor, which could be attributed to the low temperature. Temperature affects ammonia blends the most because excessive heat reduces NOx-reduction capacity, and an insufficient amount of heat increases the likelihood of incomplete combustion.

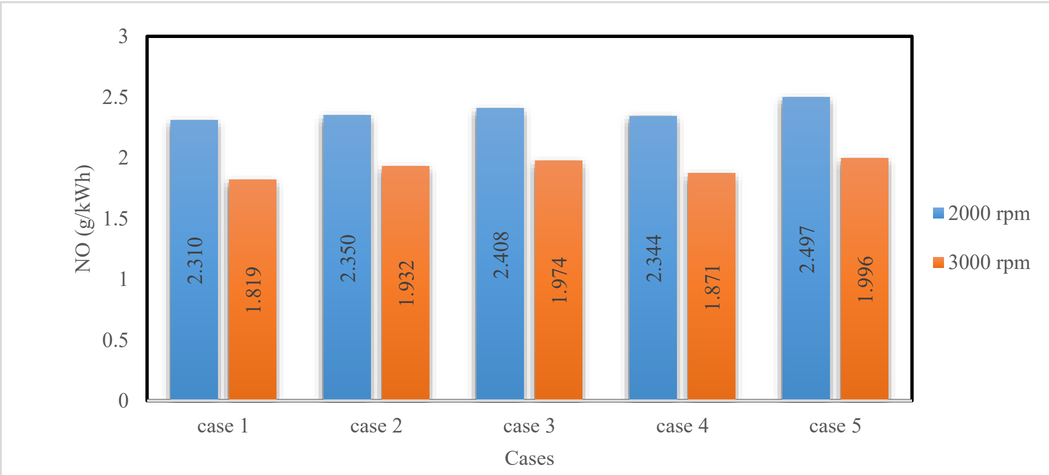


Figure 14. Correlation between NO and combustion chamber shapes at speeds of 2000 and 3000 rpm of an engine.

4.5. Particulate matter (PM) emissions

Figure 15 describes the effects of the shapes of the combustion chambers on the emissions of PM. The PM emissions in diesel engines are influenced by various factors, such as fuel composition, air–fuel ratio, mixing efficiency, temperature, and pressure. The increase in the combustion temperatures boosts the rate of chemical reactions, resulting in more thorough fuel combustion. This should result in fewer unburned hydrocarbons and reduced PM emissions. Numerical analysis demonstrated that the PM emissions of six piston designs were noticeably variable when operated with a mix of 50% NH_4OH and 50% diesel. PM emissions in a diesel engine are heavily influenced by the piston shape, as evidenced by the specific PM emission results at 2000 and 3000 rpm across the five different designs: Double Lip had 1.02 g/kWh at 2000 rpm and 1.126 g/kWh at 3000 rpm; 10D100 had 0.965 g/kWh at 2000 rpm and 0.933 g/kWh at 3000 rpm; Mexican Hat had 0.95 g/kWh at 2000 rpm and 0.903933 g/kWh at 3000 rpm; Inveco F1C Rollbuch had 0.97 g/kWh at 2000 rpm and 1.006 g/kWh at 3000 rpm; and Peugeot DW10 had 0.91 g/kWh at 2000 rpm and 0.892 g/kWh at 3000 rpm. This is because the piston design affects the combustion efficiency, including mixing the air–fuel mixture and flow turbulence in the combustion chamber. The Double Lip design has the largest PM emissions due to poor combustion efficiency and incomplete combustion of fuel, resulting in increased soot formation and the formation of particulates. Conversely, the Peugeot design of the DW 10 emits the least amount of PM, indicating that the air–fuel mixing and combustion process are more efficient and complete, thereby reducing the production of the particulate. The Mexican hat-10D100 and Inveco F1C Rollbuch designs are intermediate, with moderate PM levels, reflecting their varied degrees of combustion efficiency. The findings highlight the significance of piston geometry in regulating the quantity of particulate emissions, since an appropriate design can minimize the creation of harmful particles when the diesel burns.

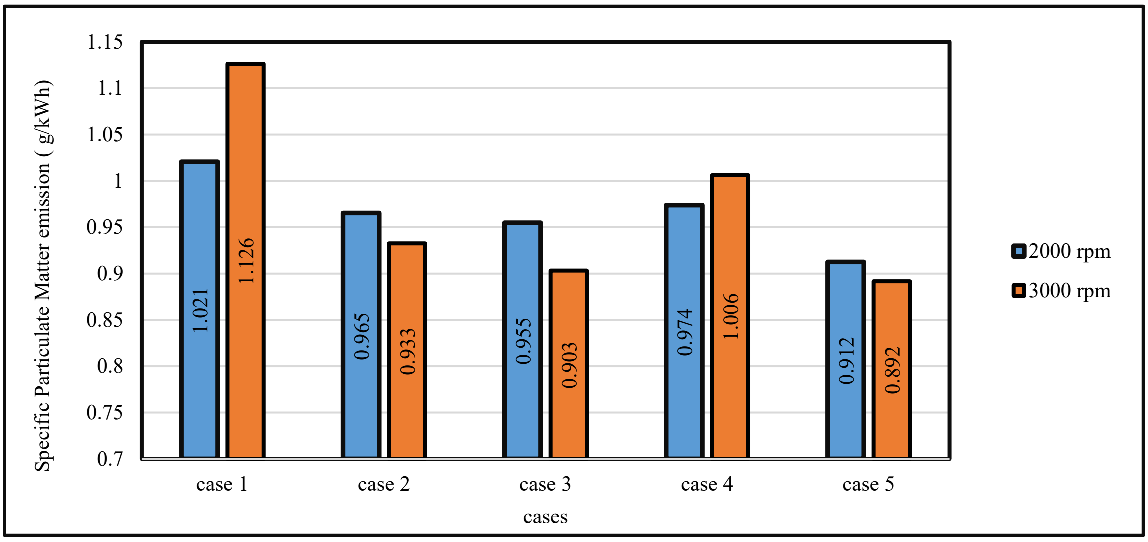


Figure 15. Relationship between PM emissions and different combustion chambers.

4.6. Hartridge smoke level

Figure 16 illustrates the impact of combustion chamber shapes on the Hartridge smoke level. Figure 16 shows that the concentration of smoke emissions decreases as the engine speed increases for the same fuel type in the combustion chamber. Moreover, smoke emissions in a diesel engine are significantly influenced by the shape of the piston, as demonstrated by the Hartridge Smoke Level results across five different piston designs: Double Lip (36.41 at 2000 rpm and 36.20 at 3000 rpm), 10D100 (34.05 at 2000 rpm and 29.8 at 3000 rpm), Mexican Hat (33.68 at 2000 rpm and 28.93 at 3000 rpm), Inveco F1C Rollbuch (34.52 at 2000 rpm and 32.17 at 3000 rpm), and Peugeot DW 10 (32.14 at 2000 rpm and 28.48 at 3000 rpm). The results indicate that the Peugeot DW 10 design produces the lowest smoke emissions, suggesting superior air–fuel mixing and combustion efficiency, which minimizes incomplete combustion and smoke formation. The Double Lip piston, on the other hand, produces the greatest smoke levels, which is probably due to ineffective distribution of the fuel–air mixture as a result of overly high turbulence or unfavorable swirl patterns. The Mexican Hat and 10D100 designs perform relatively well, showing moderate smoke levels, while the Inveco F1C Rollbuch exhibits slightly higher emissions than both. These variations highlight the critical role of combustion chamber geometry and piston bowl design in controlling smoke generation during diesel combustion.

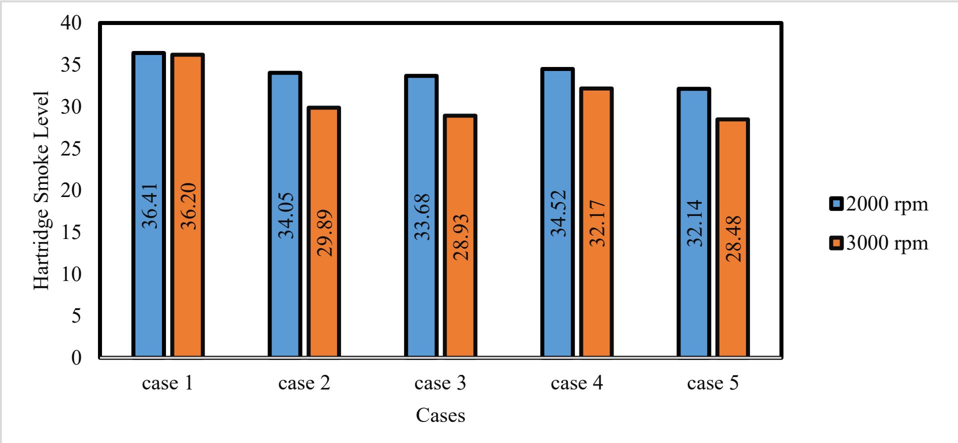


Figure 16. Correlation between smoke level in the Hartridge and various combustion chambers.

4.7. Carbon dioxide (CO_2)

The generation of CO_2 in a diesel engine is largely influenced by the level of completeness of combustion, which is affected by elements such as air-to-fuel ratio, presence of oxygen, temperature of combustion process, and fuel composition. Elevated CO_2 levels indicate that the fuel is combusted more efficiently and thoroughly. As such, there is a need to optimize engine parameters to improve combustion efficiency both in terms of performance and environmental aspects. Additionally, the piston form design significantly influences air–fuel mixing and combustion efficiency, potentially resulting in more complete combustion and increased CO_2 emissions. The effect of the shapes of the combustion chambers on the CO_2 emission is explained in Figure 17. CO_2 emissions in a diesel engine are significantly influenced by the shape of the piston, as demonstrated by the CO_2 results across five different piston designs: Double Lip had 614.94 g/kWh at 2000 rpm and 676.8g/kWh at 3000 rpm; 10D100 had 617.58 g/kWh at 2000 rpm and 677.4 g/kWh at 3000 rpm; Mexican Hat had 617.67 g/kWh at 2000 rpm and 677.94 g/kWh at 3000 rpm; Inveco F1C Rollbuch had 616.48 g/kWh at 2000 rpm and 676.33 g/kWh at 3000 rpm; and Peugeot DW 10 had 617.93 g/kWh at 2000 rpm and 679.93 g/kWh at 3000 rpm. Figure 17 shows that the concentration of CO_2 emissions increases as the engine speed increases for the same fuel type in the combustion chamber. The results indicate that the Double Lip design produces the lowest CO_2 emissions, likely due to poor fuel–air distribution caused by excessive turbulence or improper swirl patterns, as well as low temperatures inside the combustion chamber. On the other hand, the Peugeot DW 10 piston produces maximum CO_2 emissions, indicating excellent air–fuel mixing and combustion efficiency, thereby reducing the uncompleted combustion and formation of soot. These findings indicate that the piston geometry plays a significant role in regulating CO_2 emission, whereby piston design can greatly minimize or increase the CO_2 emission during the burning of diesel.

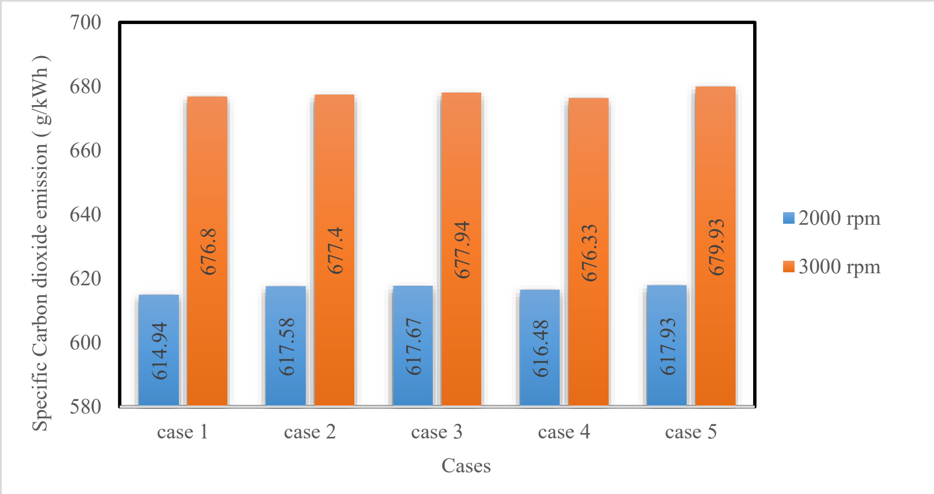


Figure 17. Correlation between CO_2 emissions and various combustion chambers.

5. Conclusions

The present work introduces the effect of combustion chamber geometry on emission formation properties and combustion behavior of a diesel engine under various engine speeds (2000 and 3000 rpm) fueled with aqueous ammonia-diesel. Five types of piston bowls, i.e., Double lip (Case 1), 10D100 (Case 2), Mexican hat (Case 3), Inveco F1C Rollbuch (Case 4), and Peugeot DW 10 (Case 5), were implemented. Numerical computations were performed on a single-cylinder, 4-stroke, and direct-injection diesel engine. The study outcomes led to the following conclusions:

- The numerical results indicated that combustion efficiency and emissions are strongly influenced by piston bowl geometry due to its effect on air–fuel mixing and turbulence inside the combustion chamber. Moreover, based on the numerical findings, with the same combustion chamber, the results explain that the NO emission concentration, Hartridge smoke level, and maximum pressure reduced with increased engine speed from 2000 to 3000 rpm, whereas maximum temperature, HRR, CO_2 emission, and PM emissions increased.
- According to numerical results, the Peugeot DW 10 (Case 5) piston bowl demonstrated superior combustion performance, reflecting greater air–fuel mixing and completeness of combustion, achieving the highest in-cylinder pressure, peak temperature, and HRR. It had the lowest PM (0.91 g/kWh and 0.892 g/kWh) and the lowest Hartridge smoke level (32.14 and 28.48) at 2000 and 3000 rpm, respectively, but also produced the highest NOx emissions (2.49 g/kWh at 2000 rpm and 1.99 g/kWh at 3000 rpm) because of high combustion temperatures. In addition, the Double Lip design (Case 1) had the worst combustion parameters, reflecting inefficient mixing and partial combustion. It recorded the lowest temperature, pressure, HRR, CO_2, and NO alongside the highest PM. Moreover, the most balanced design was Case 3 (Mexican Hat), which provided the best second-highest combustion performance. Case 3 had the second-largest peak pressure and second-largest temperature, and moderate and stable HRR. It also produced low PM (0.95 and 0.903933 g/kWh), low smoke (33.68 and 28.93), high CO_2 (617.67 and 677.94g/kWh), and only moderately increased NO_x (2.40 and 1.974 g/kWh) at 2000 and 3000 rpm, respectively.
- According to findings, the Peugeot DW 10 presents the most optimal balance of high efficiency

and low soot/smoke, although with more NO. Case 3 (Mexican Hat) provides an optimal overall balance of emissions, both soot and NO_x are reduced, and the combustion efficiency remains high. Case 3 is the most appropriate for operation with aqueous ammonia-diesel mixtures on a sustainable basis.

6. Future work

Our future work will examine the impact of injector spatial alteration, injection pressure, and nozzle size on combustion efficiency and emissions in a diesel engine fueled with a mixture of aqueous ammonia and diesel.

Use of AI tools declaration

The authors declare that no artificial intelligence tools were used in this manuscript.

Acknowledgments

The authors would like to thank the University of Baghdad for their support.

Conflict of interest

The authors state that there are no conflicts of interest related to the publication of this article.

Author contributions

Hussein A. Mahmood: Writing preparation of the original draft. Osam H. Attia: Checking and verifying the simulation. Ali O. Al-Sulttani: Review the Original draft preparation.

References

1. Sehili Y, Loubar K, Tarabet L, et al. (2024) Computational investigation of the influence of combustion chamber characteristics on a heavy-duty ammonia diesel dual fuel engine. *Energies* 17: 1231. https://doi.org/10.3390/en17051231
2. Naife TM (2022) Improvement of diesel fuel engine performance by nanoparticles additives. *J Eng* 28: 77–90. https://doi.org/10.31026/j.eng.2022.04.06
3. Ghareeb HO, Anjal HAAW (2024) Ignition delay period prediction for compression ignition engines fueled with ethanol/diesel blends. *J Eng* 30: 48–70. https://doi.org/10.31026/j.eng.2024.08.04
4. Mahmood HA, Al-Sulttani AO, Mousa NA, et al. (2022) Impact of lambda value on combustion characteristics and emissions of syngas-diesel dual-fuel engine. *Int J Technol* 13: 179–189. https://doi.org/10.14716/ijtech.v13i1.5060

5. Mahmood HA, Al-Sulttani AO, Attia OH (2021) Simulation of syngas addition effect on emissions characteristics, combustion, and performance of the diesel engine working under dual fuel mode and lambda value of 1.6. *IOP Conf Ser: Earth Environ Sci* 779: 012116. https://doi.org/10.1088/1755-1315/779/1/012116
6. Chaichan MT (2011) Exhaust analysis and performance of a single cylinder diesel engine run on dual fuels mode. *J Eng* 17: 873–885. https://doi.org/10.31026/j.eng.2011.04.17
7. Mahmood HA, Adam NM, Sahari BB, et al. (2016) Investigation on the air-gas characteristics of air-gas mixer designed for bi-engines. *Int J Appl Eng Res* 11: 7786–7794.
8. Luo W, Liu H, Liu L, et al. (2023) Effects of scavenging port angle and combustion chamber geometry on combustion and emmission of a high-pressure direct-injection natural gas marine engine. *Int J Green Energy* 20: 616–628. https://doi.org/10.1080/15435075.2022.2079380
9. Telaumbanua IK, Imai K, Sasaki K, et al. (2024) A pathway for implementing ammonia solutions as fuel blends for achieving low-emission combustion in diesel engines. *J Energy Inst* 116: 101750. https://doi.org/10.1016/j.joei.2024.101750
10. Ali, Lim O (2024) Numerical investigation of combustion and emission characteristics of the single-cylinder diesel engine fueled with diesel-ammonia mixture. *Energies* 17: 5782. https://doi.org/10.3390/en17225782
11. Chaichan MT, Abaas KI, Mohammed BA (2017) Experimental study of the effect of fuel type on the emitted emissions from SIE at idle period. *Al-Khwarizmi Eng J* 13: 1–12. https://doi.org/10.22153/kej.2017.11.001
12. Abdollahi M, Ghobadian B, Najafi G, et al. (2020) Impact of water–biodiesel–diesel nano-emulsion fuel on performance parameters and diesel engine emission. *Fuel* 280: 118576. https://doi.org/10.1016/j.fuel.2020.118576
13. Mahmood HA, Al-Sulttani AO, Attia OH (2025) Numerical investigations of SiO_2 and TiO_2 nanoparticle's addition impact on performance, combustion features, and emission characteristics of a diesel engine operating with water diesel emulsified fuel at different lambda values and engine speeds. *Int J Technol* 16: 1731–1751. https://doi.org/10.14716/ijtech.v16i5.7685
14. Oliveira P, Brójo F (2024) An experimental study on the performance and emissions of an 8% water-in-diesel emulsion stabilized by a hydrophilic surfactant blend. *Energies* 17: 1328. https://doi.org/10.3390/en17061328
15. Hasannuddin AK, Wira JY, Sarah S, et al. (2016) Performance, emissions and lubricant oil analysis of diesel engine running on emulsion fuel. *Energy Convers Manage* 117: 548–557. https://doi.org/10.1016/j.enconman.2016.03.057
16. Watanabe S, Yahya WJ, Ithnin AM, et al. (2017) Performance and emissions of diesel engine fuelled with water-in-diesel emulsion. *JSAEM* 1: 12–19. https://doi.org/10.56381/jsaem.v1i1.3
17. Berwal P, Kumar S, Khandelwal B (2021) A comprehensive review on synthesis, chemical kinetics, and practical application of ammonia as future fuel for combustion. *J Energy Inst* 99: 273–298. https://doi.org/10.1016/j.joei.2021.10.001
18. Xiao H, Ying W, Chen A, et al. (2024) Study on the impact of ammonia–diesel dual-fuel combustion on performance of a medium-speed diesel engine. *J Mar Sci Eng* 12: 806. https://doi.org/10.3390/jmse12050806
19. Guo L, Zhu J, Fu L, et al. (2023) Effects of pre-injection strategy on combustion characteristics of ammonia/diesel dual-fuel compression ignition mode. *Energies* 16: 7687. https://doi.org/10.3390/en16237687

20. Mikulčić H, Baleta J, Wang X, et al. (2021) Numerical simulation of ammonia/methane/air combustion using reduced chemical kinetics models. *Int J Hydrogen Energy* 46: 23548–23563. https://doi.org/10.1016/j.ijhydene.2021.01.109
21. Zhao Z, Miao X, Chen X, et al. (2024) Simulation study of diesel spray tilt angle and ammonia energy ratio effect on ammonia-diesel dual-fuel engine performance. *Energy Eng* 121: 2603–2620. https://doi.org/10.32604/ee.2024.051237
22. Frankl S, Gleis S, Karmann S, et al. (2021) Investigation of ammonia and hydrogen as CO_2-free fuels for heavy duty engines using a high pressure dual fuel combustion process. *Int J Engine Res* 22: 3196–3208. https://doi.org/10.1177/146808742096787
23. Zhang Z, Long W, Dong P, et al. (2023) Performance characteristics of a two-stroke low speed engine applying ammonia/diesel dual direct injection strategy. *Fuel* 332: 126086. https://doi.org/10.1016/j.fuel.2022.126086
24. Zhang H, Wang J (2015) Ammonia coverage ratio and input simultaneous estimation in ground vehicle selective catalytic reduction (SCR) systems. *J Frankl Inst* 352: 708–723. https://doi.org/10.1016/j.jfranklin.2014.06.009
25. Scharl V, Sattelmayer T (2024) Spectroscopic investigation of diesel-piloted ammonia spray combustion. *Fuel* 358: 130201. https://doi.org/10.1016/j.fuel.2023.130201
26. Scharl V, Sattelmayer T (2022) Ignition and combustion characteristics of diesel piloted ammonia injections. *Fuel Commun* 11: 100068. https://doi.org/10.1016/j.jfueco.2022.100068
27. Nadimi E, Przybyła G, Lewandowski MT, et al. (2023) Effects of ammonia on combustion, emissions, and performance of the ammonia/diesel dual-fuel compression ignition engine. *J Energy Inst* 107: 101158. https://doi.org/10.1016/j.joei.2022.101158
28. Reiter AJ, Kong SC (2011) Combustion and emissions characteristics of compression-ignition engine using dual ammonia-diesel fuel. *Fuel* 90: 87–97. https://doi.org/10.1016/j.fuel.2010.07.055
29. Huang Q, Yang R, Liu J, et al. (2025) CFD-based numerical investigation of ammonia combustion and slip characteristics in an ammonia-diesel dual-fuel engine. *J Energy Inst* 122: 102217. https://doi.org/10.1016/j.joei.2025.102217
30. Frost J, Tall A, Sheriff AM, et al. (2021) An experimental and modelling study of dual fuel aqueous ammonia and diesel combustion in a single cylinder compression ignition engine. *Int J Hydrogen Energy* 46: 35495–35510. https://doi.org/10.1016/j.ijhydene.2021.08.089
31. Al-Dawody MF, Al-Obaidi W, Aboud ED, et al. (2023) Mechanical engineering advantages of a dual fuel diesel engine powered by diesel and aqueous ammonia blends. *Fuel* 346: 128398. https://doi.org/10.1016/j.fuel.2023.128398
32. Pyrc M, Gruca M, Jamrozik A, et al. (2021) An experimental investigation of the performance, emission and combustion stability of compression ignition engine powered by diesel and ammonia solution (NH_4OH). *Int J Engine Res* 22: 2639–2653. https://doi.org/10.1177/1468087420940942
33. Sarıtaş M, Kül VS, Akansu SO, et al. (2025) Compression ignition engine performance and emissions: An experimental study on the impact of aqueous ammonia, hydrogen, and diesel fuel blends. *Int J Hydrogen Energy* 137: 679–688. https://doi.org/10.1016/j.ijhydene.2025.05.121
34. Deresso H, Nallamothu RB, Ancha VR, et al. (2022) Numerical study of different shape design of piston bowl for diesel engine combustion in a light duty single-cylinder engine. *Heliyon* 8: e09602. https://doi.org/10.1016/j.heliyon.2022.e09602

35. Rizvi IH, Gupta R (2021) Numerical investigation of injection parameters and piston bowl geometries on emission and thermal performance of DI diesel engine. *SN Appl Sci* 3: 626. https://doi.org/10.1007/s42452-021-04633-1
36. Al Rifat A, Rahman MM, Rahman MA (2026) Effect of piston bowl geometry on combustion, performance, and emission characteristics of a dual-fuel engine. *Future Energy* 5: 1–9. Available from: https://fupubco.com/fuen/article/view/556.
37. Sefatjoo H, Khaleghi H (2026) Influence of piston bowl geometry and swirl ratio on spray dynamics and fuel–air mixing in compression ignition diesel engines. *Appl Therm Eng* 289: 129908. https://doi.org/10.1016/j.applthermaleng.2026.129908
38. Siddique SA, Vijaya K, Reddy K (2015) Theoretical investigation on combustion chamber geometry of DI diesel engine to improve the performance by using Diesel-RK. *Int J Innov Technol Explor Eng* 4: 27–31. Available from: https://www.ijitee.org/wp-content/uploads/papers/v4i10/J19980341015.pdf.
39. Venu H, Soudagar MEM, Kiong TS, et al. (2025) Nanotechnology and LSTM machine learning algorithms in advanced fuel spray dynamics in CI engines with different bowl geometries. *Sci Rep* 15: 983. https://doi.org/10.1038/s41598-024-83211-y
40. Pham VC, Kim JK, Lee W-J, et al. (2022) Effects of piston bowl geometry on combustion and emissions of a four-stroke heavy-duty diesel marine engine. *Appl Sci* 12: 13012. https://doi.org/10.3390/app122413012
41. Abdelrazek MK, Abdelaal MM, El-Nahas AM (2022) Piston bowl shape and biodiesel fuel effects on combustion and emission of diesel engines. *J Eng Appl Sci* 69: 103. https://doi.org/10.1186/s44147-022-00158-5
42. Mahmood HA, Al-Sulttani AO, Alrazen HA, et al. (2024) The impact of different compression ratios on emissions, and combustion characteristics of a biodiesel engine. *AIMS Energy* 12: 924–945. https://doi.org/10.3934/energy.2024043
43. Adib AR, Rahman MM, Hassan T, et al. (2024) Novel biofuel blends for diesel engines: optimizing engine performance and emissions with *C. cohnii* microalgae biodiesel and algae-derived renewable diesel blends. *Energy Convers Manage: X* 23: 100688. https://doi.org/10.1016/j.ecmx.2024.100688
44. Rehman M, Kesharvani S, Dwivedi G (2023) Numerical investigation of performance, combustion, and emission characteristics of various microalgae biodiesel on CI engine. *Fuels* 4: 132–155. https://doi.org/10.3390/fuels4020009
45. Mohsen MJ, Al-Dawody MF (2022) The combustion characteristics of compression ignition engine fuelled partially by LPG-Diesel blends (Simulation study). *Al-Qadisiyah J Eng Sci* 15: 156–163. https://doi.org/10.30772/qjes.v15i3.828