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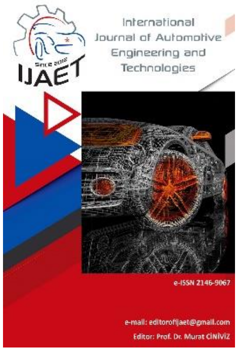

Original Research Article

Numerical analysis of various combustion chamber bowl geometries on combustion, performance, and emissions parameters in a diesel engine

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ABSTRACT

In this paper, numerical analysis of a diesel engine with various combustion chamber bowl geometries was investigated. Numerical analyses of asymmetrical bowl forms were carried out using the Ansys Forte software. Analyses were done for three different piston bowl geometries in a single-cylinder, air-cooled diesel engine at 1500 rpm engine speed. Three different bowl geometries were created, the piston-1 model is a commercial one for used engine, in the piston-2 pedestal area is removed from the piston, and in the cylindrical model bowl geometry has more straight borders. The results of the numerical study were confirmed with the results of experiments which is carried out for the piston-1 model. Simulation results showed a good correlation with experimental results. According to the results of the numerical study, the highest in-cylinder temperature was determined as 1213 K for the piston-1 model. In the piston-1 model, the combustion efficiency was improved with the increase of the swirl by the increased combustion temperature. As a result of the analysis, the thermal and combustion efficiencies of the piston-1 model were higher than the other models. The maximum turbulence velocity was observed at -12 °CA for three different piston bowl models as 4.1 m/s, 4.06 m/s, and 3.9 m/s for piston-1, piston-2, and cylindrical respectively.

Keywords: Chamber, Diesel engine, Piston, Numerical Analysis

1. Introduction

As the human population increases in the world, the demand for internal combustion vehicles also increases. The internal combustion engines continue to play an important role in the energy cycle. As a result, the global impact of exhaust emissions on the environment continues to increase.

Although the transition to hybrid and electric vehicles has accelerated in recent years, the use of internal combustion engines in the current system has not changed significantly. High performance and low emission requirements for internal combustion engines make engine design processes difficult. Carbon monoxide, nitrogen oxides, and unburned hydrocarbon emissions are being reduced through research on existing systems. Applications within this area are organized within, before, and after the combustion chamber.

Various strategies have been developed to reduce emissions. Blending with alternative fuels, changing the type and timing of fuel injection, in-cylinder flow control, EGR systems, low-temperature combustion combinations and their effects are all being studied. (1) The literature indicates that piston bowl geometry has a significant effect on the combustion of the mixture and the formation of emissions in diesel engines.

The piston bowl, which is the top part of the piston, forms the combustion chamber together with the cylinder bowl. Changing the geometry of the piston bowl affects the turbulence of the flow and the homogeneity of the air-fuel mixture. To increase air movement and improve the combustion process, it is possible to design a suitable combustion chamber that can reduce emissions without changing engine parameters. However, the effect of combustion chamber geometry on engine performance, flow field and air-fuel interaction is very complex (2).

Changing the geometry of the combustion chamber to improve the flow and movement of air in the combustion chamber affects combustion in diesel engines. Improving the air-fuel mixture improves engine performance. There are many ways to improve the air-fuel mixture in a cylinder and changing the piston bowl geometry is one of them. Studies have been carried out in the area of combustion chamber and bowl geometry design. Experimental investigation of the effects of different applications is very difficult and costly. For this reason, important predictions in engine design can be obtained with a modeling approach. (3,4). The basic approach is to optimize the piston bowl geometry to produce better rotational and crushing motions that improve mixture quality as the gas in the cylinder is swept away by the upward movement of the piston during the compression process (5).

Kakaee et al. (6) used three different geometries (stock, bathtub and cylindrical) in their study of diesel engine piston bowl geometry. They found that the bathtub geometry had better performance and emission values.

Hariharan et al. (7) used two different piston bowl geometries (re-entrant bowl and shallow bowl) in an RCCI engine. It was reported that there was no significant difference in the combustion part for these two geometries under similar conditions, while the shallow bowl geometry showed slightly better performance when the thermal efficiency was analyzed.

Gülcan and Ciniviz (8) experimentally investigated the effect of piston bowl geometry on the combustion performance and pollutant emissions of a single-cylinder methane diesel engine. High performance piston bowl geometries (toroidal and toroidal re-entrant) were used. The experimental results showed that the Toroidal Re-entrant Combustion Chamber (TRCC) geometry reduces the long ignition delay time due to methane addition and provides more stable combustion at all torque conditions. In summary, the use of the TRCC geometry is reported to be an effective way to provide more complete combustion and reduce emissions in dual fuel operation at all torque conditions from 3 to 9 Nm.

Dempsey et al. (9) showed that at low loads, the shallow bowl piston has significantly better combustion efficiency than an entering bowl piston due to reduced heat transfer losses and improved combustion efficiency. Using the standard re-entering piston bowl geometry, both fuel combinations achieve low NOx and PM emissions with a peak gross specified efficiency of 48%. Over the full load and speed range, the modified piston provided low NOx and PM emissions with a peak gross specified efficiency of approximately 51%.

Splitter et al. (10) experimentally investigated the effect of nominal piston bowl depth on thermal efficiency to minimize heat transfer losses and THC emissions by reducing the compression volume. In the experiments, a modified piston was used to investigate the effect of piston groove volume, compression geometry, and compression ratio on performance and efficiency. Of all the pistons tested, the tub-type piston was found to provide the highest braking efficiency, low emissions, and low-pressure rise rates at practical intake and exhaust gas recirculation

(EGR) temperatures.

Jaichandar et al. (11) worked to optimize the combination of injection timing and combustion chamber geometry to achieve higher performance and lower emissions from a biodiesel-fueled diesel engine. The experimental study used a single-cylinder direct injection (DI) diesel engine equipped with hemispherical and toroidal re-entrant combustion chamber (TRCC) geometry pistons. The TRCC application resulted in a better air-fuel mixture and delayed injection timing. A 5.64 percent improvement in brake thermal efficiency, a 4.6 percent reduction in brake-specific fuel consumption and an 11 percent reduction in nitrogen oxide (NOx) levels compared to the base engine with ULSD.

Mobasheri and Peng (12) used CFD modeling to investigate the effect of reentrant combustor geometry on mixture preparation, the combustion process and engine performance in a diesel engine. To investigate the effect of combustor, thirteen different types of combustor configurations were considered based on four categories, including piston bowl depth, piston bowl width, piston bottom surface and lip area. The results confirmed that the combustion chamber geometry has a significant effect on the combustion process. It was shown that by changing the design of the piston bowl, the amount of emission pollutants can be reduced while keeping other engine performance parameters constant.

Pham et al. (13) investigated the effect of piston bowl geometry on combustion and emissions in a four-stroke direct injection (DI) heavy duty marine diesel engine. Three types of piston bowl geometries were numerically analyzed. The results showed that ω-type and re-entrant piston seats increased cylinder power and reduced the specific fuel consumption (ISFOC) compared to U-type. In particular, ω-type and re-entrant piston crowns reduced NO (nitric oxide) emissions due to lower peak temperatures compared to U-type piston crowns. Piston bowl type was also found to have a negligible effect on soot and CO2 emission characteristics. The use of re-entrant piston bowls is highly recommended to improve engine performance and fuel consumption while reducing NO emissions.

Lee and Park (14) carried out a piston bowl optimization study to reduce emission rates in a gasoline and diesel dual fuel compression ignition engine. They found that emission parameters were reduced when the re-entrant shape was removed from the piston bowl surface. It was predicted that emission parameters could be directly related to flow separation and thus combustion efficiency.

As can be seen from the literature, rare studies have been conducted on the effect of different bowl geometries focused on the turbulence velocity and kinetic energy in the combustion chamber of a diesel engine. In this article, the effects of changing kinetic energy and turbulence velocity on combustion and emission parameters in diesel engines due to the change in piston bowl geometry were investigated numerically.

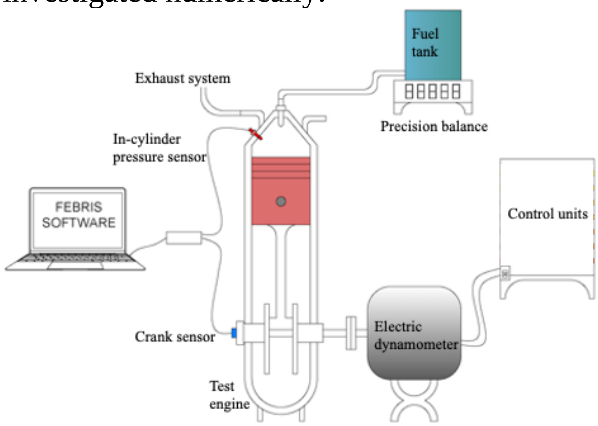


Figure 1 The experimental setup schematic diagram.

2. Materials and Methods

The experiments were carried out at the internal combustion engine laboratory of the Mechanical Engineering Department of Batman University. The experiments were carried out using a single-cylinder, direct injection, and air-cooled DDJ9500E diesel engine. The tests were performed for piston-1 at full load. The FEBRİS combustion analysis software with a pressure sensor was used to obtain combustion data. The peak pressure values versus crank angle have been collected by using this software. Engine tests were carried out with a diesel engine power generator. The tests were performed with the engine at full load and at 1500 rpm. The experimental setup schematic diagram is shown in Fig.1. The technical specifications of the single-cylinder diesel engine are given in Table 1.

In this work, the ANSYS Forte tool is used. Fig 2 shows the different pistons combustion chamber bowl geometry used in this study. Fig 2.a shows the piston model used in the experiment and named as piston-1. The new piston bowl geometry named as piston-2 is shown in Figure 2.b. Figure 2.c shows the piston bowl geometry of the diesel engine, which is named as cylindrical.

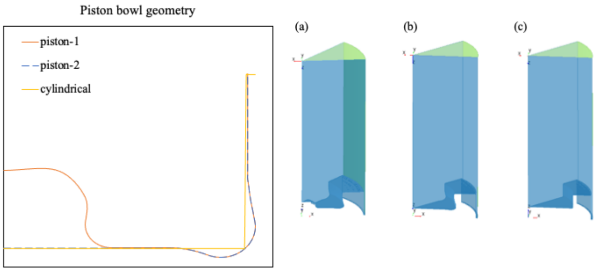


Figure 2 Piston bowl geometry (a) piston-1 (b) piston-2 (c) cylindrical

Table 1 Technical specifications of the test engine

Determinations	Descriptions
Model	DDJ9500 E
Power output	10 HP
Rated speed (rpm)	1500-1800
Rated power (kW)	7.5
Max power	8.2
BSFC	<270 kg/kWh
Cooling method	Air cooling
Intake system	Naturally aspirated
Bore*Stroke(mm)	92*75
Cylinder vol. (cm^3)	498
Cylinder number	1
İnjection system	Direct Injection
Compression ratio	19:1

For creating the piston models coordinates of the model profiles were saved as a .csv file. Then these .csv files were imported to Ansys Forte software. Topology 1 was used to create the sector mesh structure of the pistons. The minimum cell height of the mesh structure is 0.016 cm. The sector angle is defined as 45. The calculation model has used a maximum grid size of 1.6 mm and a 1/8 sector in consideration of the combustion chambers' symmetry. The mesh structure for all piston models is given in Figure 3.

In the simulations, the initial temperature was assumed as 352 K and the initial pressure was assumed as 0.235 MPa. Fuel mass was used as 0.054 g which is obtained from experiments for 1500 rpm engine speed. The Reynolds-averaged Navier Stokes (RANS) RNG k-epsilon is applied to solve the in-cylinder turbulence process.

The inflow droplet temperature and spray atomization model were defined as 368 K and KH-RT model. The start of injection and duration of injection was defined as - 20 CA and 7.75 CA. The IMEP (Indicated Mean Effective Pressure) was defined as 0.41 MPa. The simulations were carried out between -160 and 120 crank angles with 0.5-degree precision. Boundary conditions are given in Table 2.

Table 2. Simulation boundary conditions

Determinations	Descriptions
Number of cylinders	Single cylinder
Type of cooling	Air-cooled
Rated speed (rpm)	1500
Number of injection nozzle	4
Injection spray angle	140
Injection rate (mass)	0.054 g
Air inlet temperature	352 K
Air inlet pressure	0.235 MPa
Fuel injection temperature	368 K
Turbulence model	RNG k-epsilon
Wall heat transfer	Han & Reitz
Soot emission model	Kinetic model
Cylinder head temperature	470 K
Cylinder liner temperature	420 K

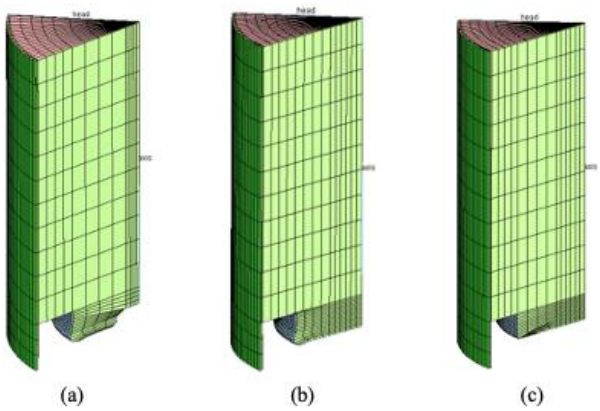


Figure 3 Mesh structure of the piston bowl geometry (a) piston-1 (b) piston-2 (c) cylindrical

2. Results and Discussion

To verify the reliability of the simulation with the experimental study for the model piston-1, the obtained in-cylinder pressure values were compared. In-cylinder pressure comparison of experimental and simulation results is shown in Fig. 4. FEBRİS combustion analysis software was used to read the in-cylinder pressure of the engine. A good agreement was found between simulation results and experimental results for in-cylinder pressure on the piston-1 model.

Fig. 5. shows the effect of different piston bowl geometries on in-cylinder pressure. Figure 5.a shows the in-cylinder pressure between -160 to 120 degrees which is the total crank angle for the whole simulation.

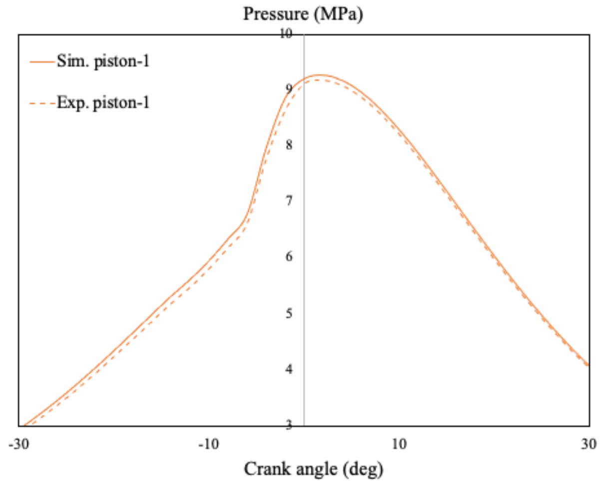
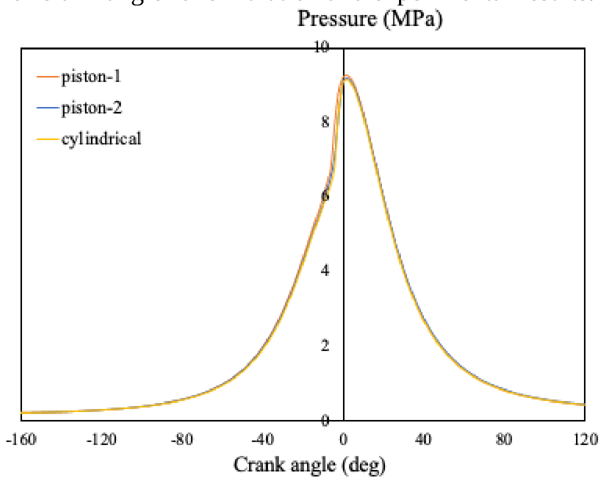
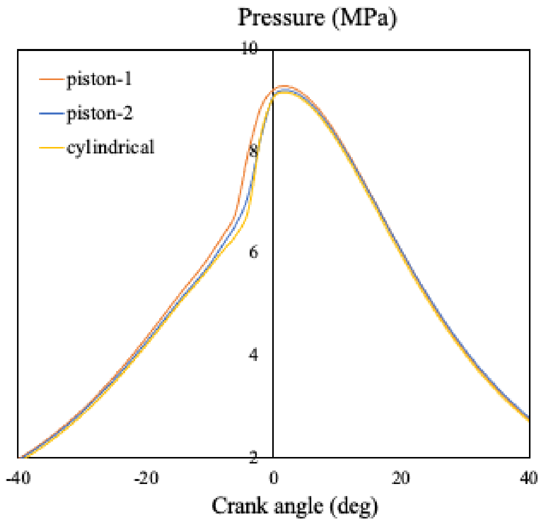


Figure 4. Comparison of pressure change as a function of crank angle for simulation and experimental results.



(a)



(b)

Figure 5. Comparison of pressure for three different piston bowl geometry

The in-cylinder pressure values were observed as 9.28 MPa, 9.19 MPa and 9.15 MPa for piston-1, piston-2, and cylindrical, respectively. While the lowest in-cylinder pressure was in the cylindrical piston, the highest in-cylinder pressure value was observed in the piston-1. The start of combustion was earliest for the piston-1 model which can mainly be attributed to the turbulence increased in the combustion chamber. Turbulence enhances the mixing of air and fuel within the combustion chamber. This leads to a more homogeneous air-fuel mixture, promoting efficient combustion and reducing the potential for localized rich or lean regions. As a result, cylinder pressure rises more uniformly during the combustion process. The enhanced mixing and faster combustion resulting from turbulence typically lead to higher peak cylinder pressures. This increase in pressure can result in higher torque output and better engine performance.

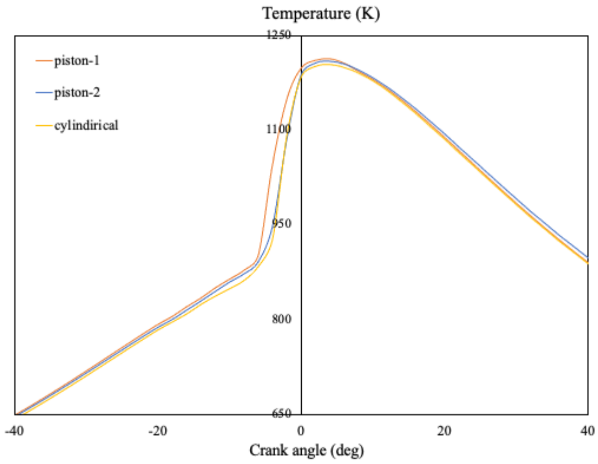


Figure 6. Comparison of temperature for three different piston bowl geometry

Fig. 6. shows the effect of different piston bowl geometry on in-cylinder temperature. The in-cylinder temperature values were observed as 1213 K, 1209 K and 1204 K for piston-1, piston-2 and cylindrical, respectively. The highest cylinder temperature values are found for model piston-1. It can also be attributed to the formation of turbulence in the combustion chamber. Turbulence leads to more efficient heat transfer within the combustion chamber. This results in better control of temperature and pressure distribution, creating conditions that favor quicker and more uniform ignition across the combustion chamber. Improved mixing and faster combustion due to turbulence can lead to higher peak combustion

temperatures. However, excessive peak temperatures can potentially contribute to increased nitrogen oxide (NOx) emissions. Modern diesel engine designs often seek to balance the benefits of higher peak temperatures for efficiency with the need to control emissions. However, properly managed turbulence can help reduce the variability in in-cylinder temperature. This is important for maintaining consistent combustion characteristics from cycle to cycle, which improves engine stability and reduces the risk of knocking and other combustion-related issues.

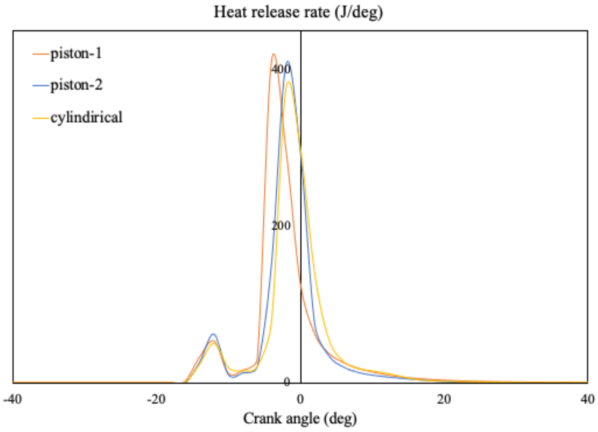


Figure 7. Comparison of HRR for three different piston bowl geometry

The change of heat release rate for different piston bowl models is shown in Fig. 7. Maximum HRR was observed at -2 °CA for piston-2 and cylindrical. Maximum HRR was found for piston-1 at -4 °CA. The HRR was observed as 410 J/deg, 407.3 J/deg, and 375.7 J/deg for piston-1, piston-2 and cylindrical, respectively. It is clear from the figure that the piston-1 model has a much lower ignition delay, in comparison to the model of piston-2 and cylindrical. It can be directly attributed to the increase in turbulence formation due to the shape of the piston surface.

Turbulence helps create a more uniform air-fuel mixture by preventing fuel-rich or fuel-lean regions. This homogeneity facilitates a more consistent ignition process, as the ignition energy required is spread evenly throughout the mixture. As a result, the ignition delay is reduced. Besides, Turbulence promotes faster flame propagation through the combustion chamber. The chaotic mixing of the air-fuel mixture causes small-scale eddies and turbulence, which expose different parts of the mixture to the ignition source (typically a high-temperature region). This rapid flame spread reduces the time required for the flame to travel from the ignition source to other parts of the mixture, thus reducing the ignition delay.

Wall heat transfer change for different piston bowl models is shown in Fig 8. The wall heat transfer corresponding to 2 °CA was observed as 267.84 J, 250.13 J, and 241.98 J for piston-1, piston-2, and cylindrical, respectively. Even though the heat transfer rate through the piston walls was higher for the piston-1 model, the effective turbulence can lead to shorter combustion durations, which means that the combustion process occurs over a shorter period of the piston's travel. This results in improved combustion stability and reduced heat loss to the cylinder walls, contributing to higher thermal efficiency.

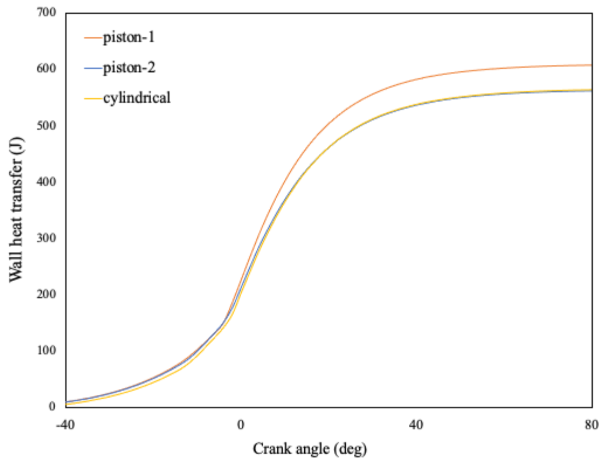


Figure 8. Wall heat transfer for three different piston bowl geometry

Fig 9. shows the influence of different piston bowl geometries on exhaust emission. The simulation results of the NO_X value were observed as 0.22 g/kWh, 0.16 g/kWh, and 0.15 g/kWh for piston-1, piston-2, and cylindrical, respectively. The simulation results of the UHC value were observed as 2.51 g/kWh, 3.01 g/kWh, and 3.22 g/kWh for piston-1, piston-2, and cylindrical, respectively. The simulation results of the CO value were observed as 5.47 g/kWh, 6.19 g/kWh, and 6.71 g/kWh for piston-1, piston-2, and cylindrical, respectively. The highest NOx emissions for the piston-1 model can be attributed to the increased combustion temperature. The simulation results of the soot value were observed as 0.082 g/kWh, 0.091 g/kWh, and

0.095 g/kWh for piston-1, piston-2, and cylindrical, respectively. Excessive turbulence or poor combustion control can lead to increased NOx emissions. Very high turbulence levels can lead to incomplete combustion, causing fuel-rich regions where NOx can be formed. This is especially true during low load or cold-start conditions. However, enhanced turbulence, for the model of piston-1, improves combustion efficiency and helps to burn more of the hydrocarbons present in the fuel.

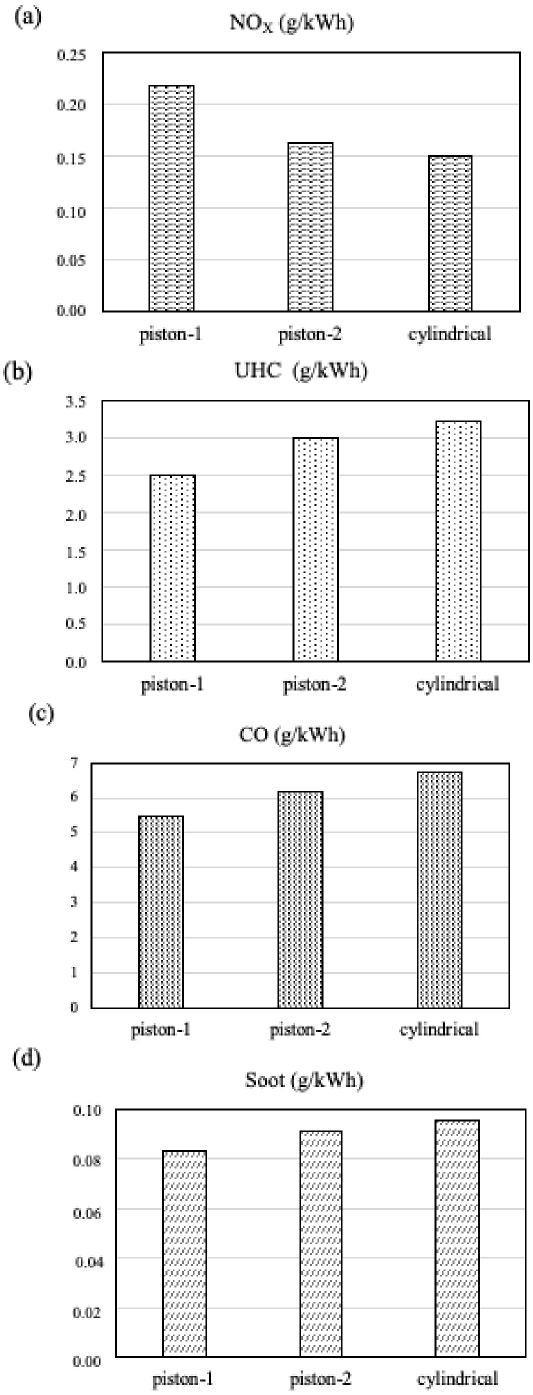


Figure 9. Influence of different piston bowl geometry on emissions (a) NO_X (b) UHC (c) CO (d) Soot

This can lead to reduced unburned hydrocarbon emissions. Besides, improved mixing and faster combustion due to turbulence result in better oxidation of carbon monoxide. This leads to lower CO emissions as more CO is converted into CO_2 during combustion. The effects of combustion chamber turbulence on emissions in diesel engines are closely intertwined with the overall combustion process. Properly managed turbulence through optimized engine combustion chamber design, injector technology, and combustion control strategies can lead to improved emissions performance. However, an incorrect balance of turbulence can result in higher emissions, particularly under certain operating conditions. Modern engine development often involves sophisticated simulations and experimentation to achieve the right level of turbulence for optimal emissions control and engine performance. Therefore, the design of combustion chamber is very important issue for optimal reduction of all emissions.

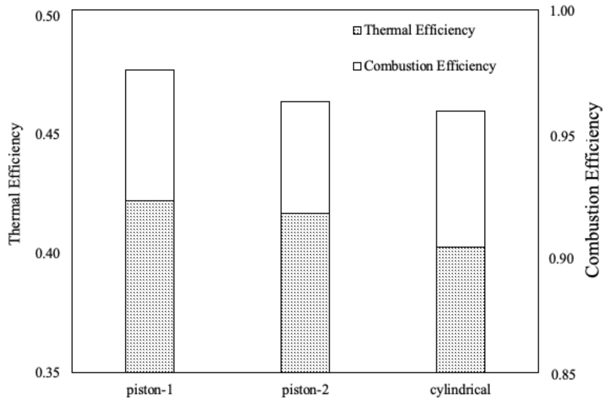


Figure 10. Influence of different piston bowl geometry on efficiency

Fig 10. shows the influence of different piston bowl geometries on thermal efficiency and combustion efficiency. The thermal efficiency values for piston-1, piston-2, and cylindrical are 0.42, 0.41, and 0.4 respectively. The combustion efficiency values for piston-1, piston-2, and cylindrical are 0.98, 0.96, and 0.95 respectively. Both the thermal efficiency and combustion efficiency are increased when the in-cylinder temperature increases and the heat transfer rate decreases. Turbulence increases the flame propagation speed by promoting faster and more complete combustion of the air-fuel mixture. This leads to a steeper pressure rise rate during the combustion process, which can contribute to improved engine efficiency. Combustion chamber turbulence in diesel engines has a direct impact on thermal efficiency, which refers to the efficiency with which the energy content of the fuel is converted into useful mechanical work. Turbulence influences various aspects of the combustion process that ultimately affect thermal efficiency. The turbulence that was created in the combustion chamber by a well-designed profile can enhance the mixing of air and fuel within the combustion chamber. This leads to a more uniform air-fuel mixture, ensuring that all fuel molecules have an opportunity to react with oxygen. A well-mixed mixture promotes more complete combustion, which in turn increases thermal efficiency. Effective turbulence ensures that fuel is thoroughly mixed with air, reducing the likelihood of unburned fuel being expelled in the exhaust gases. This not only reduces waste but also contributes to higher thermal efficiency by utilizing more of the fuel's energy content.

The change of turbulent kinetic energy for different piston models is shown in Fig. 11. Maximum turbulent kinetic energy values were observed at -14 °CA. The turbulent kinetic energy corresponding to -14 °CA was observed as 77.86 m^2/s^2, 71.48 m^2/s^2 and 72.81 m^2/s^2 for piston-1, piston-2, and cylindrical, respectively. Combustion chamber turbulence plays a crucial role in diesel engines by influencing the turbulence of the combustion process and, subsequently, the cylinder pressure. Turbulence is the chaotic and rapid mixing of air and fuel within the combustion chamber, and it has significant effects on various aspects of diesel combustion, performance, and emissions. The highest turbulence energy values were obtained for the piston- model-1.

Fig. 12. shows the change in turbulence velocity for different piston models. Maximum turbulence velocity for different piston model values was observed at -12 °CA and -2 °CA. The turbulence velocity values corresponding to -12 °CA for piston-1, piston-2, and cylindrical were 4.1 m/s, 4.06 m/s, and 3.9 m/s respectively. The turbulence velocity corresponding to -2 °CA was observed as 4.1 m/s, 3.54 m/s, and 3.47 m/s for piston-1, piston-2, and cylindrical, respectively.

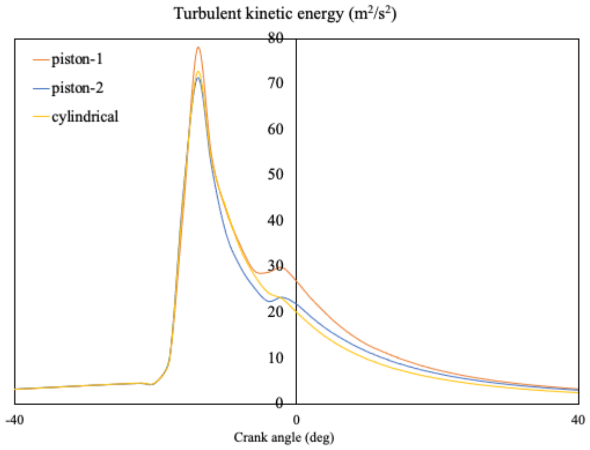


Figure 11. The change of turbulent kinetic energy for different piston models

Combustion chamber design plays a critical role in influencing turbulence levels within diesel engines. Turbulence can be enhanced by incorporating features such as squish areas, tumble ports, or tangential inlet flows. These features create swirling or tumbling motions within the air-fuel mixture, leading to better mixing and combustion. Different designs can promote either tumble or swirl flow patterns. Tumble is characterized by an axial rotation of the mixture around the cylinder axis, while swirl involves a rotational motion of the mixture about a vertical axis. Tumble enhances mixing near the injector, leading to faster and more controlled combustion, while swirl can improve mixture distribution across the combustion chamber. Carefully designed combustion chamber shapes can optimize the flow patterns and promote thorough mixing of the air and fuel. When all three piston designs are taken into consideration, the highest turbulence velocity values are obtained for the piston-1 model.

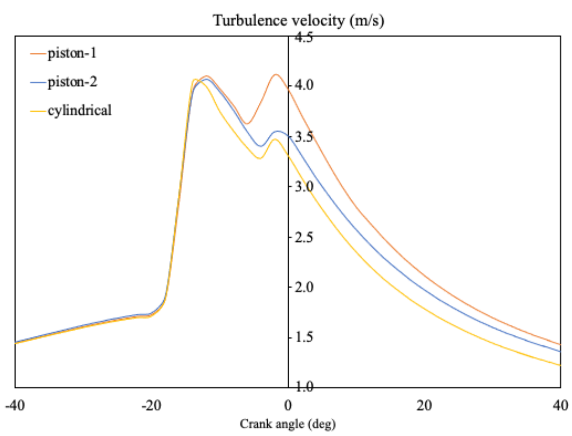


Figure 12. The change of turbulence velocity for different piston models

In-cylinder temperature distribution changes with crank angles are given in Fig. 13. for different piston models. The in-cylinder maximum temperature corresponding to -2 °CA was observed as 2180 K, 2154 K, and 2145 K for piston-1, piston-2, and cylindrical, respectively. The in-cylinder maximum temperature equivalent to 8 °CA was found to be 2170 K, 2090 K, and 2090 K for piston-1, piston-2, and cylindrical, respectively. When Figure 13. is observed, the higher levels of temperature distributions are more clustered around the pedestal area for the model of piston-1. Therefore, it can be concluded that the overall engine performance values are higher for a model of piston-1 which can also lead to an increased in-cylinder temperature and pressure values that eventually maintain the increased engine efficiency.

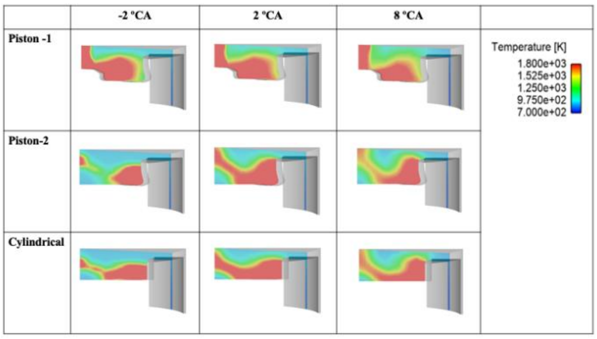


Figure 13. In-cylinder temperature distribution for different piston models and CA

4. Conclusion

We investigate the effect of the piston bowl geometry on the combustion efficiency, emissions, and cylinder turbulence formation numerically. The following conclusions can be made according to the results of the limited study.

- The investigation revealed that the piston-1 model had the highest in-cylinder pressure. In this model, combustion started early due to the pedestal area which has a turbulence-increasing effect in the combustion chamber.
- Maximum HRR was detected at -4 °CA in the piston-1 model but at -2 °CA in the piston-2 and cylinder models. The piston-1 model has a lower ignition delay than other models because it can be explained by the increase in turbulence speed depending on the piston bowl shape.
- The thermal efficiency and combustion efficiency values were observed from highest to lowest for piston-1, piston-2, and cylindrical, respectively.
- According to the in-cylinder temperature distribution, the temperature values of the piston-1 model are higher than the piston-2 and cylindrical models. However, this caused the NO_x value to be higher in the piston-1 model.

It was observed that the turbulence velocity and kinetic energy values in the piston-1 model were higher than in other models. The turbulent kinetic energy was observed as piston-1, piston-2 and cylindrical for 77.86 m^2/s^2, 71.48 m^2/s^2 and 72.81 m^2/s^2, respectively. The maximum turbulent kinetic energy values were determined at -14 °CA in the piston-1 model.

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