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An experimental investigation of new chamber geometry on the combustion characteristics, performance and emissions in a light-duty diesel engine

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ABSTRACT

Piston bowl geometry directly affects many parameters such as combustion, flow, turbulence, and mixture formation in-cylinder. This causes a change in engine performance, combustion characteristics and exhaust emissions. In the present study, the effects of piston bowl geometry on mixture formation, combustion characteristics, and emission characteristics of a direct injection compression ignition engine have been experimentally tested. Novel bowl geometries namely New Combustion Chamber (NCC) were developed and made comparison Standard Combustion Chamber (SCC). Air-cooled, four-stroke, single-cylinder and direct injection compression ignition engine was used in the experiments. The diesel engine was operated under full load and different speed conditions (1800, 1900, 2000, 2100 and 2200 rpm). Within the scope of the study, brake specific fuel consumption, engine torque, exhaust emissions (CO_2, CO, O_2, HC, NO and soot) and combustion characteristics (Instantaneous heat release rate, in-cylinder pressure change, rate of pressure rise and mass fraction burned) were examined. The results showed that the eight pockets and sub-base radius in the NCC geometry significantly improved the specified parameters. The NCC geometry according to the SCC, the specific fuel consumption decreased by approximately 6.89% and engine torque increase 6.32% in the achieved maximum torque at 2000 rpm. Exhaust emissions are generally reduced thanks to the NCC bowl geometry. However, NO emissions increased due to increased temperatures, improved combustion and mixture formation at all speeds. When the combustion analyzes were examined, it was seen that the new type of bowl improved the combustion. The reason was that the increase of the bowl in the surface area, and the increased turbulence with it provided easier evaporation of the fuel droplets and better mixture formation with the air. The spread of combustion over a large area causes an increase in the flame surface area and more controlled combustion was achieved.

1. Introduction

Today, many researches are carried out to make engines more efficient and have lower emission values. To have such an engine, the mixture formation in-cylinder needs to be improved [1]. There are many ways to achieve this, and changing the bowl geometry is one of these operations [2]. Many parameters such as swirl level, turbulence, tumble, squish rate in combustion chamber are significantly affected by the bowl geometry [3–5]. Good selection of bowl form is quite effective in improving thermal efficiency, atomization, combustion characteristics and pollutant exhaust emission values in diesel engines [6]. The bowl geometry affects the in-cylinder air and fuel movements [7,8]. The higher swirl ratio improved from the bowl geometry can occur a better mixture formation [9]. The swirl ratio rises at the air flow speed, resulting in better mixture formation. Thanks to better mixture formation, exhaust emissions are reduced and combustion is improved. [10]. The shape of the combustion chamber is important. For example; the narrow entrance of the bowl can occur strong squish, especially at high speeds, thereby the mixture formation can be more easily realized [11–13]. Today, most researchers have examined the effects of some parameters and zones such as squish zone, pip shape of the combustion chamber, bowl lip shape, central pip, pip inclination, depth, top clearance, toroidal radius and throat diameter [14–17]. A re-entrant bowl form is quite common in high speed compression ignition engines [18,19]. This re-entrant piston geometry type gives the best performance with turbulent kinetic energy and high air velocity [20,21]. In the re-entrant combustion chambers, the soot emission is reduced in the

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Nomenclature

SCC	Standard Combustion Chamber
NCC	New Combustion Chamber
CA	Crank Angle
TKE	Turbulent kinetic energy
J/0	Joules/degrees
R	Radius
TDC	Top Dead Center
RPM	Revolutions per Minute
TRCC	Torroidal Re-entrant Combustion Chamber
HCC	Hemispherical Combustion Chamber
TCC	Torroidal Combustion Chamber
DLSB	Double-layer split bowls
MHCC	Hemispherical Combustion Chamber
CHRR	Cumulative heat release rate
LSPB	Lateral swirl Piston bowl
DSPB	Dual swirl Piston bowl
LDSPB	Re-entrant, lateral and dual swirl Piston bowl
HPB	Hemispherical Piston bowl

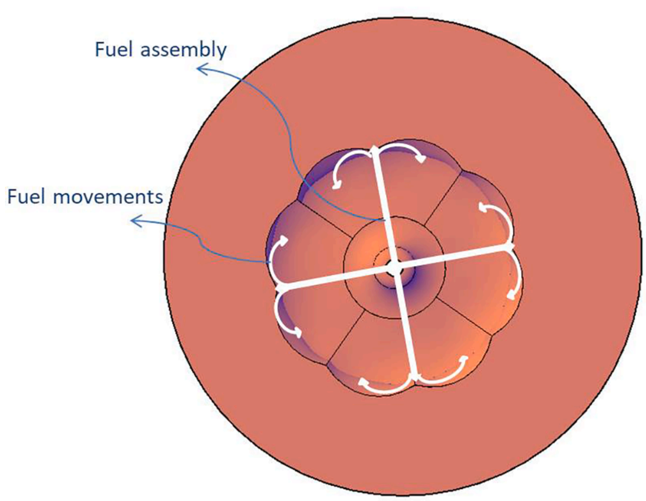

Fig. 1. Fuel spray distribution and line in chamber.

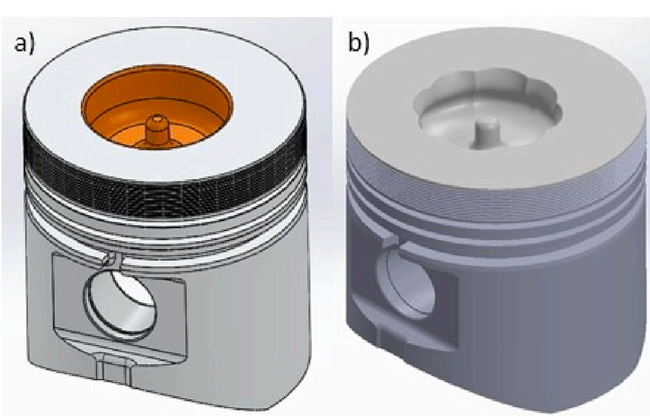

Fig 2. The SCC geometry (a) and the NCC geometry (b).

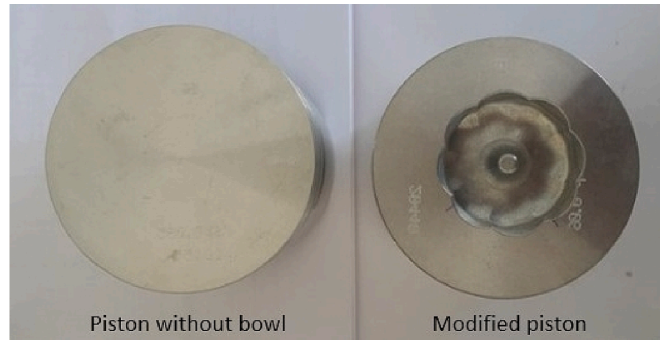

Fig. 3. Unmodified (left) and modified (right) view of the piston.

Table 1
Technical specifications of experimental engine.

Engine Type	4- stroke, direct injection diesel engine
Number of cylinders	1
Cylinder volume	510 cm^3
Bore × Stroke	85 × 90 (mm × mm)
Compression ratio	17.5:1
Maximum power	6.6@3000 (kW)
Maximum torque	32.8@2000 (Nm)
Injection angle	126°
Number of Nozzles	4

regions where the swirl and turbulence are close to the compression TDC [22]. Venkateswaran and Nagarajan [23] used a re-entrant bowl type with a re-entry angle of 43°and a throat diameter of this geometry is 10% less than the throat diameter of the original geometry. In this geometry type, while soot decreased by 46%, NO_x emission increased by 15% because of the increase in combustion efficiency and speed. Perini et al. [24] showed that classic re-entrant geometries have stronger flow separation at the intake, inhibiting chamber swirl, and however higher swirl according to stepped-lip bowl design, because its less tilted swirl and more axisymmetric squish structure. Leach et al. [25] showed that the mass fraction burned (MFB) duration increased in all operating conditions with the stepped lip design. Also, the vortex formed in the zones close to the TDC contributes to the squish flow with this design [26]. Dolak et al. [27] also stated that stepped bowl design reduces CO and soot emissions, and provides lower fuel consumption compared to the classical design. Some researchers [28,29] have studied the wider-shallow incineration chamber. Thanks to this bowl geometry, the risk of knocking in-cylinder under load conditions is reduced [30,31].

Li et al. [32] emphasized that the shallow depth geometry increased the in-cylinder temperature and pressure, but this caused an increase in NO emissions. Jyoti and Reddy [33] tested three bowl geometries (Torroidal Re-entrant Combustion Chamber (TRCC), Hemispherical Combustion Chamber (HCC) and Torroidal Combustion Chamber (TCC)) in a compression ignition engine. The thermal efficiency of TRCC and TCC geometries according to the HCC was higher 3.28% and 2.94%, respectively. However, the maximum energy release was decreased by 3.1% and 1.3%, respectively, because of the air swirl in-cylinder. Gafoor and Gupta [34] handled the effect of initial swirl ratios and bowl form on emissions and engine performance in a compression ignition engine. This study reveals that optimum performance and emission values were obtained in two cases with a diameter ratio of 70% and an initial swirl ratio of 0.5, a diameter ratio of 55% and an initial swirl ratio of 2.5. Leng et al. [35] numerically evaluated the effects of four bowl types, such as four diameter-to-depth ratios, three nozzle orifice patterns and two of which were double-layer split bowls (DLSBs). The DLSB has a shallow dish-like structure that has improved mixture formation, shortened combustion duration, and achieved higher power density. Yaliwal et al. [36] tested five different piston geometries (HPB, RPB, LSPB, DSPB and

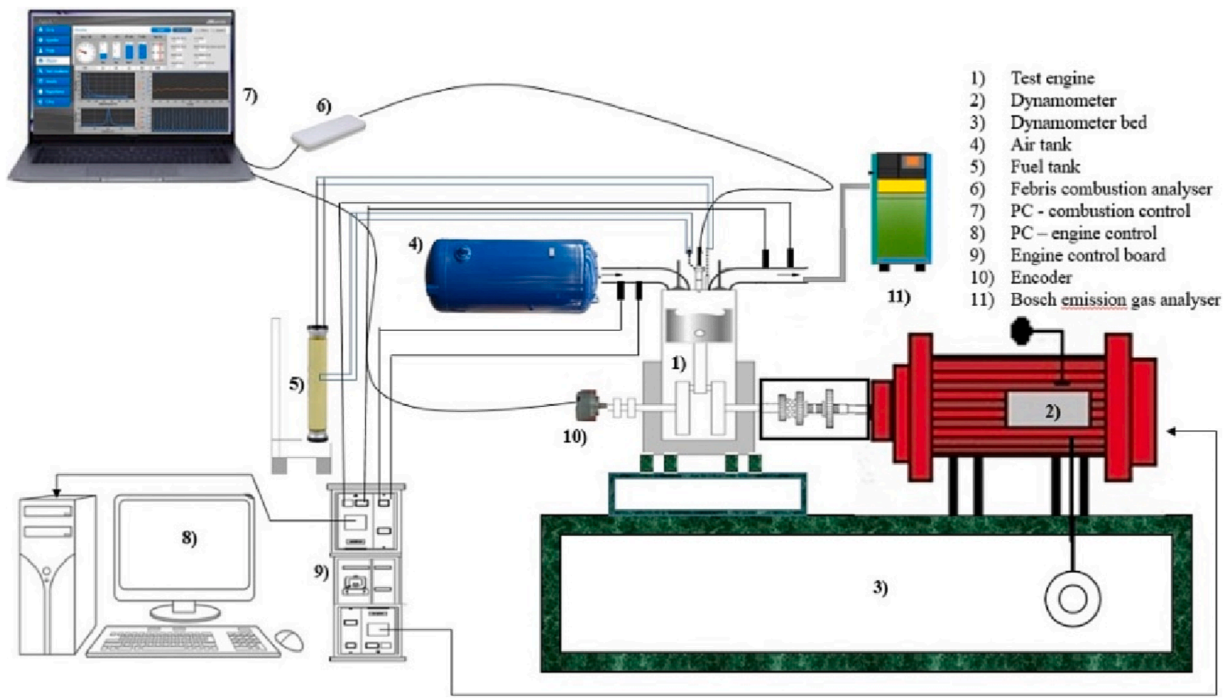

Fig. 4. View of the experimental setup as schematic.

Table 2
Uncertainties in the measured parameters.

Pressure Sensor	Encoder	Electric dynamometer
Optrand brand fiber optic sensor	Cubler	Baturalp Tayland brand brake
Measuring range 0–200 bar	Measuring range 0–12000 rpm	Reached maximum torque 80 Nm
0.025 V/bar sensibility	Encoder resolution 360x1°CA	%±0.02 accuracy
120 kHz natural frequency	Crank angle is converted to TTL signal.	Torque sensor ± 12 VDC output
% ≤±0.5 accuracy	may be operated by 5 V or 120 mA	It works with 220 V.
Operating temperature range −40 and 360 °C	Operating temperature range −40 and 85 °C	Operating temperature range −10 and 60 °C
BEA 060 exhaust emission device		
Technical data	**BEA 060**	
Power supply unit operating voltage	100–240 V, 50–60H	
Operating temperature	5 °C to 45 °C	
Designation	**Measuring range**	**Resolution**
CO	0–10 vol%	0.001 vol%
CO_2	0–18 vol%	0.010 vol%
HC	0–9.999 ppm	1.0 ppm
O_2	0–22 vol%	0.010 vol%
NO	0–5000 ppm	1.0 ppm
Soot	0–9.99 m^{-1}	0.01 m^{-1}

LDSPB) in a diesel engine. The LDSPB type compared to other geometries, thermal efficiency increased by 10.6%, carbon emissions decreased by around 5–26%, and NO_x emission also increased by 25.4%. Sener et al. [37] tested six different bowl geometries (DA, DB, DC, DD, DE and DF). The DF design was the best design due to its curvature rear

Table 4
Exhaust emission values measured at different speeds and full load.

		Engine speed (rpm)				
		1800	**1900**	**2000**	**2100**	**2200**
SCC	**CO (% vol)**	3.3	3.24	3.58	3.89	3.702
	NO (ppmvol)	468	523	584	626–630	635–640
	Soot (1/m)	9.9	9.9	9.8	9.9	9.7
	HC (ppmvol)	119	144	158	161–162	158
	O_2 (% vol)	15.6–15.45	14.45	13.72	13.41	13.39
	CO_2 (% vol)	1.76–1.78	2.04	2.22–2.36	2.51	2.49
NCC	**CO (% vol)**	2.584	3.05	1.9	2.132	2.12
	NO (ppmvol)	351	582	600	676.5	715
	Soot (1/m)	9.7	9.8	9	9.1	9.3
	HC (ppmvol)	83–77	124	148	149	160
	O_2 (% vol)	13.46	14.36	13.48	13.26	13.32
	CO_2 (% vol)	2	2.10	2.27	2.51	2.52

Table 3
Engine performance measurements of two different bowl geometries.

	Combustion chamber type	Engine speed (rpm)				
		1800	**1900**	**2000**	**2100**	**2200**
Torque (Nm)	SCC	30.57	29.88	29.9	28.69	28.04
	NCC	31.6	31.34	31.79	30.2	29.62
bsfc (g/kWh)	SCC	252.35	282	219	281.4	274.3
	NCC	244.25	266.1	203.9	257.6	259.4
Exhaust temperature (K)	SCC	451	473.6	515.1	537.4	554.3
	NCC	453.1	487.3	520	540	559.36

Table 5
Soot emissions values for idle speed.

Soot (1/m)	Engine speed (rpm) 1800	1900	2000	2100	2200
SCC	0.3	0.41	0.54	0.89	1.3
NCC	0.18	0.4	0.49	0.85	1.12

side shape according to combustion temperature distribution, pressure, heat release rate and emissions (CO, UHC, NO_x and soot). Bapu et al. [38] compared a modified Hemispherical Combustion Chamber (MHCC) piston with HCC in a compression ignition engine. Presented study using Ansys Fluent software, the MHCC compared to HCC, peak pressure increased by 4.7% ignition delay period was shorter, soot emissions decreased by about 35%, and NO increased due to improved combustion. Similar studies are available in the literature [39,40], Venu et al. [41] emphasized that it obtained the best conclusions in chamber named TRCC was obtained according to cylinder pressure, heat release rate (HRR), cumulative heat release rate (CHRR) and emissions. In another study, TRCC geometry gave better results than TCC and HCC geometry in terms of combustion characteristics. It was also emphasized that the spray angle is very important for the bowl geometry [42]. In addition,

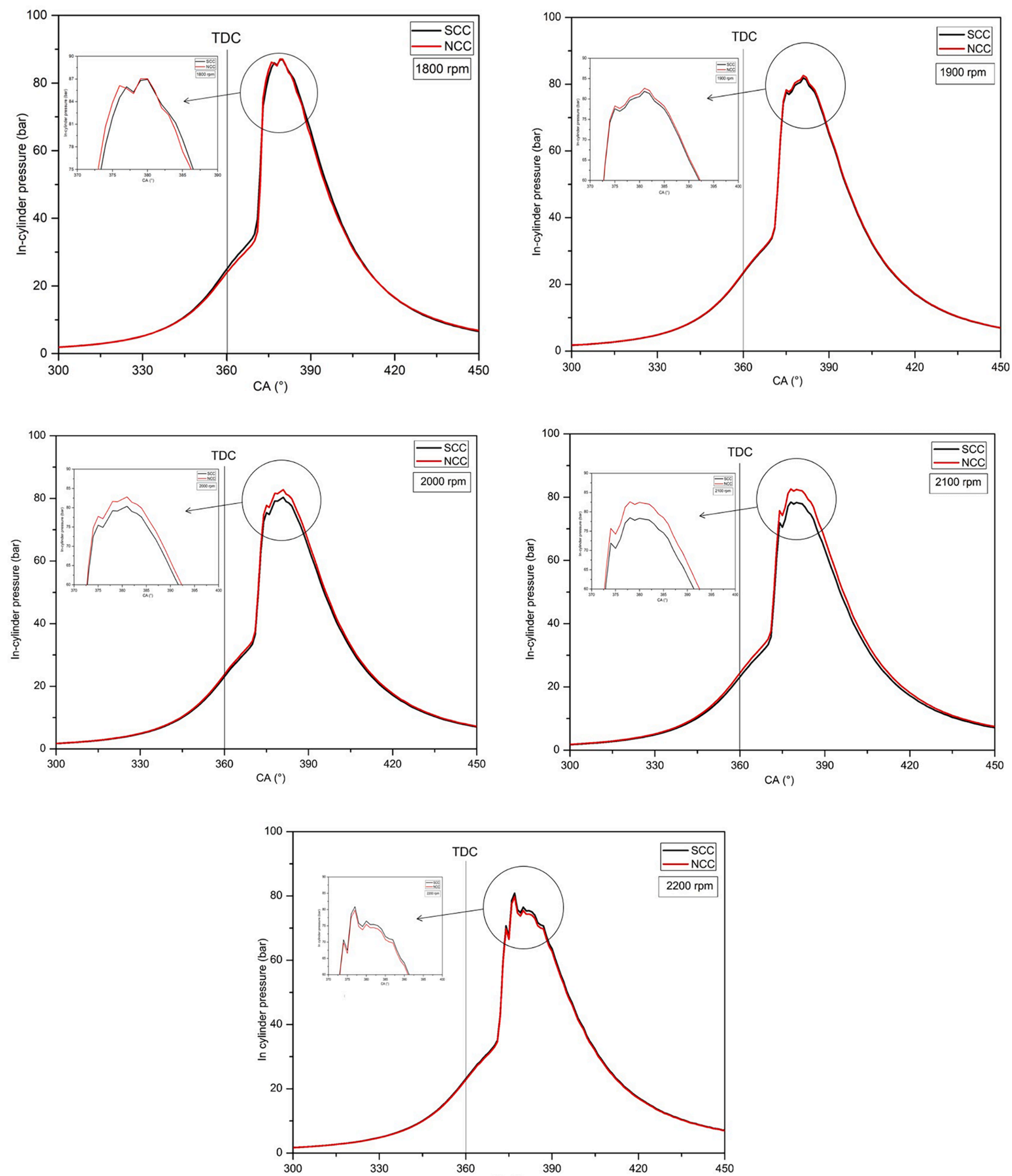

Fig. 5. Pressure/crank angle change of the SCC and NCC bowl geometries at different speeds.

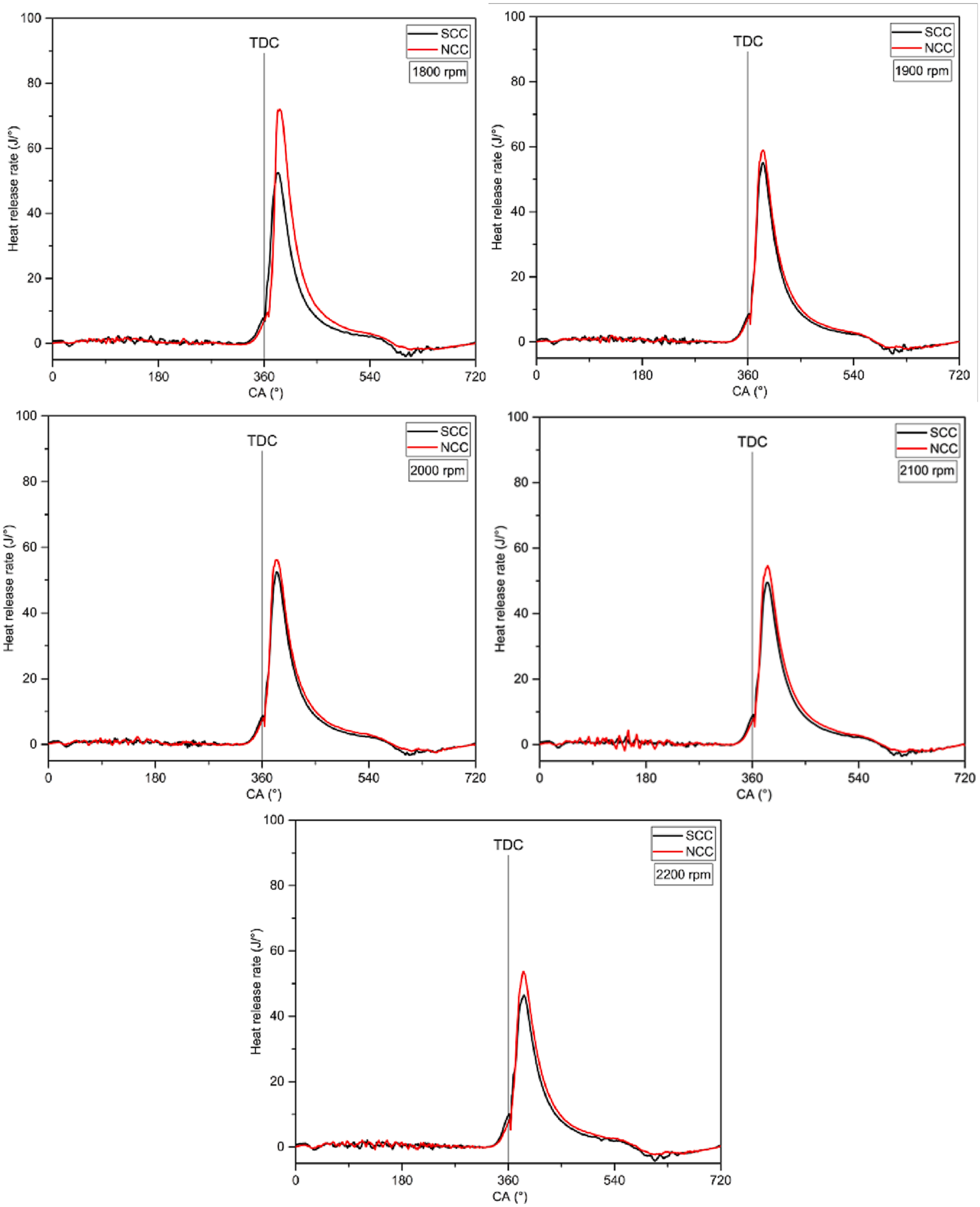

Fig. 6. Heat release rates/crank angle change of the SCC and NCC bowl geometries at different speeds.

increasing the number of injector holes contributed to the TRCC geometry in terms of combustion and emission [43]. Karthickeyan [44] performed exergy and energy analyzes for TCC, TRCC and HCC geometries. It was concluded that TCC geometry has better swirl and squish behaviour, better mixture formation and is closer to complete combustion than the others. Risi et al. [45] numerically evaluated different bowl types under different load and speed conditions in a compression ignition engine. As a result, it reached that the combustion chamber must be deep and narrow, with a shallow re-entry and a low projection in bowl geometry, the injection must be directed towards the bowl entrance for decrease NOx emissions. Soot emissions are a very important issue in diesel engines. Lim and Min [46] emphasized that the most important

Table 6
Numerical values of combustion analysis.

	Engine speed (rpm) Geometry type	1800	1900	2000	2100	2200
Mean cumulative heat release rate (J)	SCC	1357	1375	1330	1283	1224
	NCC	1615	1481	1457	1428	1390
Maximum cumulative heat release rate (J)	SCC	3035	3110	3017	2917	2786
	NCC	3775	3373	3316	3244	3156
Maximum in-cylinder pressure (bar)	SCC	86.9	81.87	80.37	78.47	80.87
	NCC	87.11	82.63	82.8	82.61	79.77
Mean Indicated pressure (bar)	SCC	7.78	7.80	7.83	7.67	7.63
	NCC	7.79	7.79	7.88	7.75	7.69

parameters in reducing soot emissions are the spray angle and the bowl form.

In this presented study, combustion chamber bowl geometry has been developed for air cooled, four-stroke, single-cylinder, light-duty diesel engine. The new combustion chamber (NCC) geometry is compared with the standard combustion chamber (SCC) geometry. The effect of NCC which has eight-cavity bowl geometry was investigated on engine performance, emissions and combustion characteristics.

2. Materıal and methods

2.1. Experimental study

The movement of the piston in the compression process is considered an important step for the mixture of fuel/air. Although many radius and geometric lengths are emphasized in the NCC in terms of construction features, the total volume of the chamber is equal to the SCC. The guidance of the fuel droplets in the chamber is accepted as an important parameter in the design of the bowl geometry. Fuel is injected with the help of four nozzles of the fuel injector with standard direct injection. The injected fuel moves along the distance dy, and is directed towards the center of the pocket with radius R3 by hitting the wall where the two pockets meet. This is achieved by the NCC geometry (Fig. 1). The wall is connected to the bottom of the bowl with a radius so that the fuel assembly can easily mix with the air flow. In the piston geometry consisting of eight cavities, the best model that can allow swirl, turbulence and air–fuel mobility and improve combustion is emphasized.

Fig. 2a shows the piston with the SCC on the existing test engine. The piston with the NCC is given in Fig. 2b. The state of the piston with and without a bowl is given in Fig. 3. The NCC is compared to the SCC, the injected fuel assembly by the injector with four nozzles, is hit to the junction of the two cavities, which will both evaporate with the effect of temperature and be directed towards the cavity center where there is high air movement. Thus, instead of a multi-hole injector and high spray pressures, a bowl geometry with different pockets was used. Thanks to this geometry, combustion was improved at low injection pressures, and PM (Particulate matter), CO and HC values, which are incompletely burned products, were reduced.

The experiments were carried out in a single cylinder, air-cooled and direct injection diesel engine. The technical properties of the engine are given in Table 1. The test engine was controlled by an electrical dynamometer. The engine was operated at 1800–2200 rpm (by 200 cycle) and full load. K-type PT temperature sensors were used to measure the ambient and exhaust gases temperatures. The test setup was connected to a computer with connections, and the instant results were directly transferred to the computer environment. The crank angle sensor, pressure sensor, and Febris combustion analysis system were engine integrated to determine combustion parameters such as the cylinder pressure, heat release, and mass fraction burned. In the combustion analysis, the in-cylinder pressure change was measured using the OPRANT brand optical pressure sensor, and the crank angle was measured using the KUBLER brand encoder. The experimental setup is given in Fig. 4.

Bosch BEA 060 model emission device was used for exhaust emission measurements. Bosch 070 model smoke measurement kit was used for soot emission measurement. The exhaust gas components of CO, HC, CO_2, O_2 and NO were measured with the Bosch BEA 060 emission device. The lambda air value is calculated from the measured gas values. Non-dispersive infrared method (NDIR-non-dispersive infrared spectroscopy) is applied to measure the CO, CO_2 and HC ratios. Oxygen is determined by the electrochemically acting sensor. From the measured HC, CO, CO_2 and oxygen concentration, the emission device calculates the lambda air value. The uncertainty analysis of the exhaust emission devices and other test setup is given in Table 2.

3. Results and discussion

3.1. Engine performance and emissions

Torque, specific fuel consumption, exhaust temperature, exhaust emissions (CO, CO_2, HC, NO and soot) were evaluated under the same conditions for both combustion chamber geometries. The injector used in ANTOR 3 LD 510 diesel engine has 4 holes, and it has the ability to spray the fuel into the combustion chamber at an angle of 90°. Combustion analyzes showed that; in combustion chambers with directed fuel injection, the point where the fuel meets the wall and the spray distributions directly affect both combustion and exhaust emission values. For this reason, a certain angle φ (fuel assembly directly angle) was defined in the new piston with eight pocket geometries. The pocket in the bowl geometry was aligned with the spray axis of the injector, using the φ angle (in 10°) where optimum results were obtained from four different φ angles (0°, 7.5°, 10° and 15°). Thus, the injection axis and the fuel wall impact axis of the piston were aligned. Thus, it was aimed to accelerate the mixture formation by ensuring that the fuel reaches the vapor phase without undergoing pyrolysis. In addition, in order to ensure rapid evaporation of the fuel, the design was carried out within the optimum limits by considering the squish velocity in the vertical direction of the air compressed into the pocket during the compression of the piston and the ratio of the smallest diameter value of the pocket to the cylinder diameter. By directing the vaporized fuel droplets by hitting the pocket area of the wall at 10° angle, the mixture with the air was improved. This reflected in the engine performance and emission values. First of all, engine characteristics tests were carried out at the engine with SCC. The engine was operated without load for 15 min, and it was ensured to reach a stable state. The throttle lever was brought to the full throttle position, the engine was operated at the desired speed by dynamometer and was brought to full load gradually. A similar test procedure was performed for the NCC. Table 3 shows the measured torque, bsfc and exhaust temperature changes at different engine speeds of two different bowl geometries. When the results were examined, the torque value of the engine with NCC for all engine speeds was higher than the SCC. The highest engine torque was obtained with the NCC geometry at 2000 rpm. At this speed, it increased by about 6.32% compared to the SCC. This increase means that there is an improvement in efficiency with the NCC bowl geometry. This result was also reflected in the Febris combustion analysis. The mean indicated pressure values increased when the NCC geometry was compared to the SCC. By keeping many parameters constant (Injection pressure, injection advance, air excess coefficient, etc.), the improvement of combustion in engines can be explained by the formation of mixture and the combustion dynamics that develop afterwards. It can be said that the flow rate form peculiar to the newly designed combustion chamber provides some positive effects on mixture formation and combustion. While designing the NCC bowl geometry, one of the issues that was emphasized was the flow movements of the air. Combustion analyzes showed that; although the flow movements were slightly reduced with the high number of

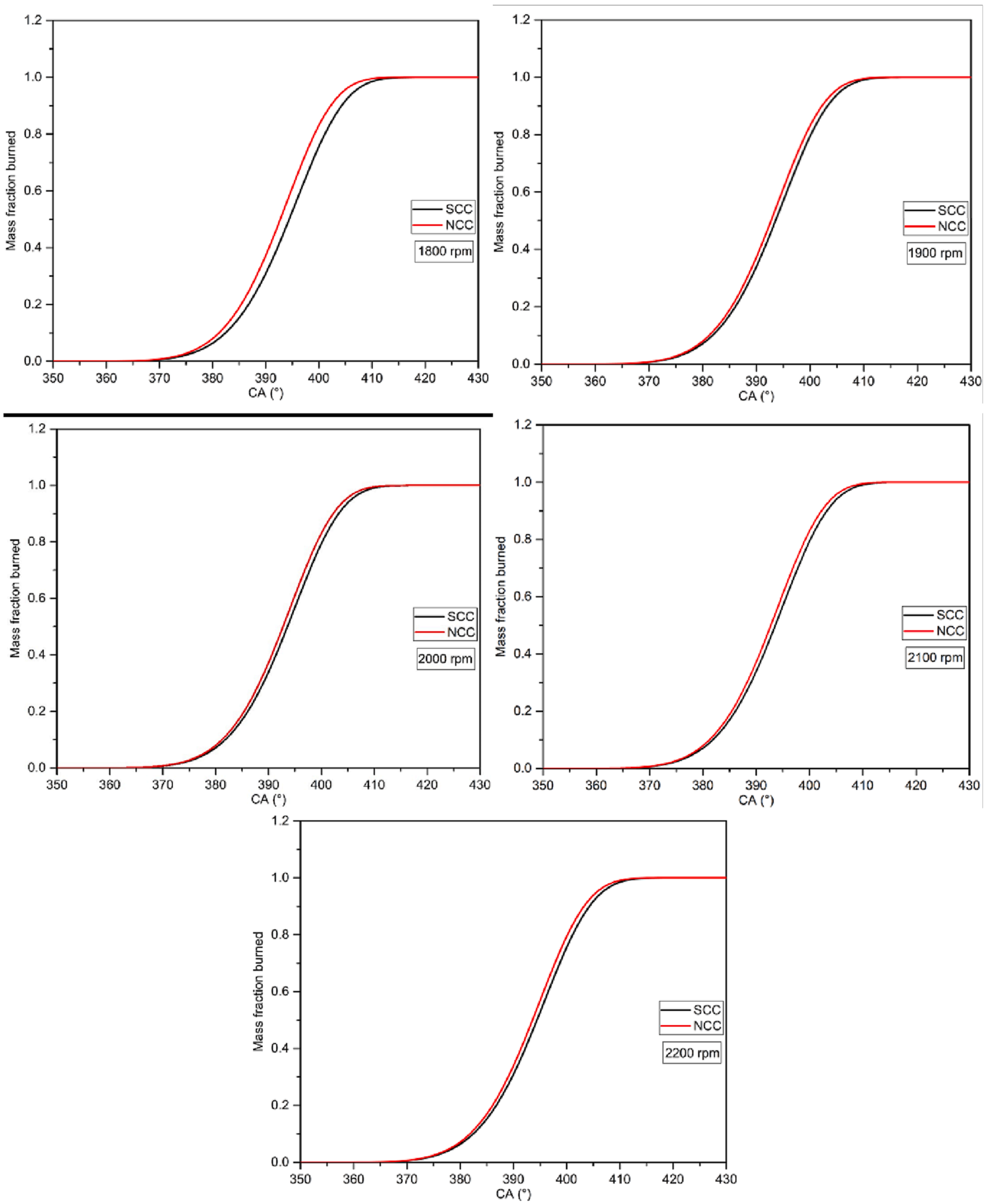

Fig. 7. Mass fraction burned/crank angle change of the SCC and NCC bowl geometries at different speeds.

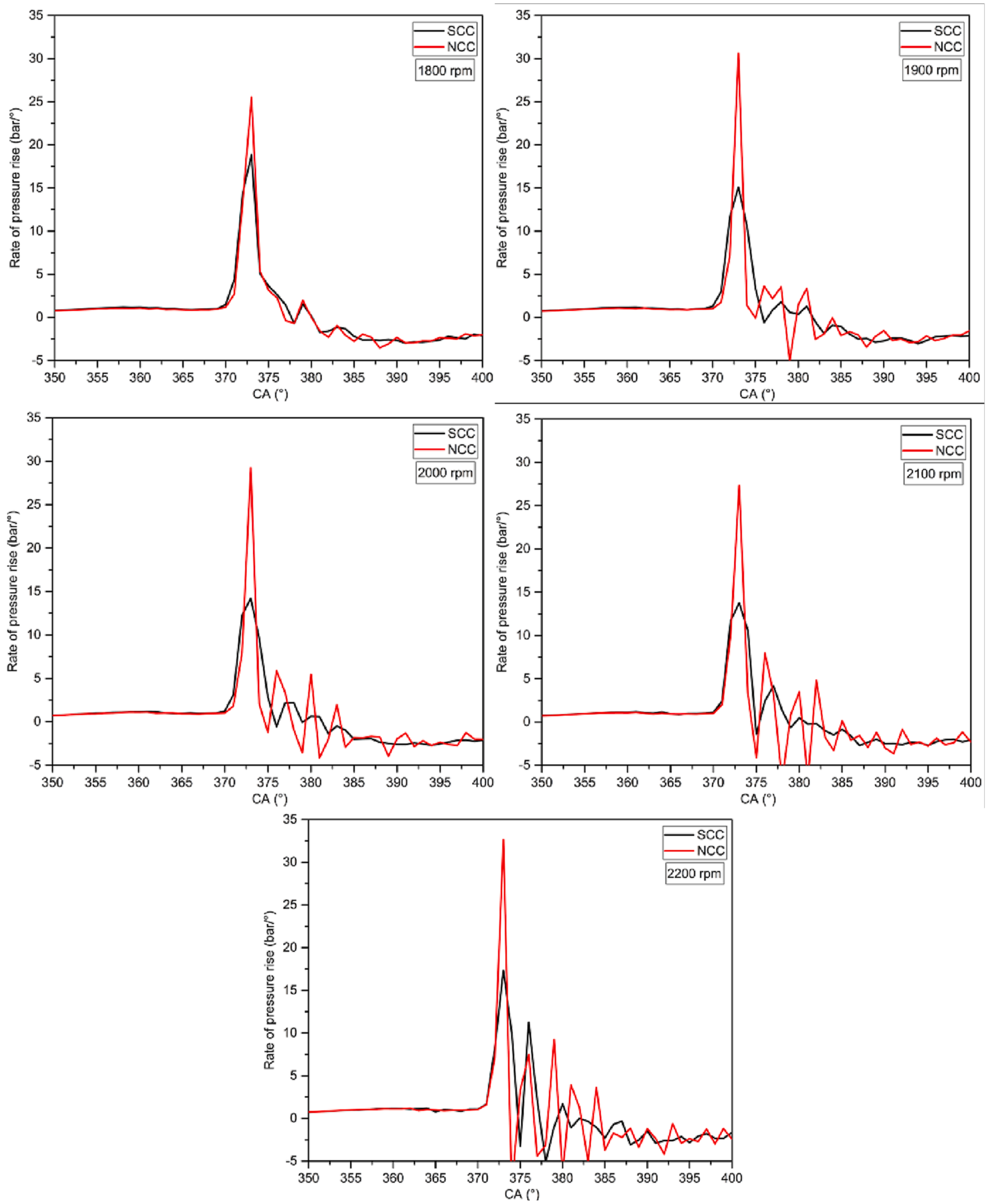

Fig. 8. Change of the rate of pressure rise with regard to crankshaft angle of SCC and NCC bowl geometries at different speeds.

pockets in the engine with NCC, the swirl ratios had been brought to values close to the SCC with the presence of the radius connecting the bowl wall to the sub-base. In addition, high swirl ratios are known to cause increased NO levels in diesel engines [47]. It could be thought the pocket structure of the NCC caused an increase in turbulent kinetic energy inside the chamber. When all these are considered together, the mixture formation with the NCC geometry is improved and this is reflected in the combustion and exhaust parameters. With this increase in combustion efficiency, the bsfc values also decreased. The NCC geometry compared to the SCC, in the specific fuel consumption value of the engine caused a reduction of approximately 6.89% at 2000 rpm.

The shape of chambers in engines affects the flow rates and regimes. Therefore, these changing factors also affect mixture formation and combustion rates. The increased turbulence is particularly important for wall-directed fuel/air mixtures. Especially around TDC, increasing turbulence contributed to the distribution of fuel droplets in the chamber. The increase of the in-bowl surface area and the turbulence which increased considered with it provided easier evaporation of the fuel droplets and butter mixture formation with the air. The spread of combustion over a large area caused an increase in the flame surface area and more controlled combustion was achieved. In Table 4 and Table 5, the exhaust emission values for two different combustion chambers were given depending on the engine speed. It observed that there were significant reductions in some emissions. It is thought that the change in the distance between the wall-injector nozzle and the increased turbulence values of the in-cylinder flow shortens the evaporation times of the fuel droplets. This resulted in a shortening of the combustion duration and an increase in combustion efficiency. In the NCC model compared to the SCC, CO emissions decreased. The reduction in CO emissions was greater at engine speed 2000 rpm and above operating conditions. Mixture formation and burning durations know to be very important in this reduction.

NO emissions are significantly affected by minor changes to the engine. NO emission increased with mixture formation and improved combustion. However, keeping it at a controllable level is very important for other emissions.

Another important type of emission for diesel engines is soot emissions. The fact that the soot emission measured for the SCC was above the boundary conditions of the device prevented the improvement in what proportions for the NCC. However, fewer soot emissions were obtained with the NCC geometry, according to both idle and full throttle conditions. It was evaluated that this decrease in soot emission occurs in parallel with the improvement of combustion. As a result of the tendency of C and H atoms to bond more with O atoms, the oxygen ratios in the exhaust gases decreased. CO_2 emissions increased as a result of shortening the combustion duration and increasing the burning speed with the NCC geometry. Another improvement was seen in HC emission by using the piston with the NCC bowl geometry. The HC emissions increased as engine speed increased. As the flame propagation speed increases, it causes a decrease in HC emissions. Here, the surface area/volume ratio is very important. By directing the fuel droplets of the pocket structure of the NCC, the spread and speed of the flame increased. In the bowl, both the lower base radius and the gradual narrowing of the bowl diameter cause the speed of the flame on the piston crown and the ignition of the fuel vapour in this region. This is more evident in the NCC geometry. As seen in the mass fraction burned/crank angle change of the SCC and NCC graphs (Fig. 6), especially (more pronounced for 1800 rpm) the mass of the burning fuel is higher in NCC geometry, and at the same time, the heat release rate has peaked more than SCC (compared to other engine speeds). The rapid burning of the fuel during the expansion stroke may have prevented sufficient time for NO chain reactions to develop. Therefore, this obstacle in front of NO chain reactions in the controlled combustion and afterburning phases may be caused by the reduction of the level.

3.2. Combustion chamber analyzes

The mean indicated pressure is a very important parameter for the comparison of combustion parameters in engines. In all measurements, the data from the instant sensors were transferred to the computer. By taking the average of approximately 200 cycles, the average cycle results were prepared. In the engine tests for both piston models, the mean indicated pressure values obtained throughout the cycle are shown in Table 6. Indicated pressure changes are a factor directly related to the indicated torque values. When the indicated pressure changes obtained with the help of the Febris combustion analysis program were examined, it was seen that a decrease occurs after 2000 rpm of the engine in both combustion chambers. This was in parallel with the engine performance characteristics. Especially at 2000 rpm operating conditions, while the mean indicated pressure value was 7.83 bar for the SCC, this value was determined as 7.88 bar for the NCC. In this engine with a fixed injection advance, this pressure was higher in the NCC geometry. The examined other parameter is the in-cylinder pressure distribution. This parameter gives important information about mixture formation and ignition. When the maximum in-cylinder pressure values were examined for both geometries, the NCC geometry was higher than the SCC (Table 4). The combustion improved in the NCC due to better air–fuel mixture [48]. As a matter of fact, sudden and rising maximum pressures can be effective on NO emissions [49]. At all speeds, the maximum pressures were slightly higher in the NCC geometry. Increased turbulence in-cylinder and shortened ignition delay durations improved mixture formation and combustion process. It was seen that these improvements were also reflected in the heat release rates. Fig. 5 and Fig. 6 show the pressure and heat release rate curves of the engine in which both geometries were given at different engine speeds. For the maximum torque, the maximum instantaneous heat release rate with the NCC geometry was 56.12 J/° at 385° CA, while it was 52.54 J/° at 385 °CA with the SCC geometry. This result was also consistent with in-cylinder pressure. Table 6 shows some calculated combustion analysis data at all speeds.

Fuel plastered on the combustion chamber pocket walls can rapidly evaporate along the main pocket radius. The evaporation of the fuel with the help of the wall relatively prevents the pyrolysis of the fuel vapors and the formation of soot. In order to provide an effective air/fuel mixture, two different radius were defined for the piston, the radius of the main cavity and the radius of the bottom base. The combustion process is intended to take place largely in the center of the combustion chamber pocket. As a result, significant changes were detected in PM, CO and HC values. Although it was considered an issue that increases NO emissions, it had been observed that the NO values obtained in the NCC geometry and the SCC were close to each other. In the NCC bowl geometry, the fuel evaporated by taking advantage of the wall temperature and the shortening of the penetration, and this improved the mixture formation. When the results were examined, the maximum heat release was obtained with the NCC bowl geometry at all speeds.

The change of the mass fraction burned of the NCC and the SCC geometries with respect to the CA is given in Fig. 7 at all speeds. Combustion earlier started and higher combustion velocity was obtained in the NCC geometry compared to the SCC. As the reason for this, it could be said that the ignition delay is shortened as a result of the fuel assembly at the NCC geometry hitting the intersections of the eight pockets and providing evaporation in a shorter duration. This showed that the NCC geometry had better turbulence density, improved mixture formation and higher quality flame kernel than the SCC. Start of injection was the same angle for both geometries. It was thought that the area of the flame increased, and the flame front expands and the burning speed increased with the increase in turbulence intensity in the NCC. This result is consistent with studies in the literature [50–52].

Fig. 8 shows the pressure change/crank angle variation of the SCC and the NCC geometries. The NCC geometry according to the SCC type, the increase in both the maximum pressure and heat release caused the rates of pressure rise to increase. This situation can also be effective in increasing the mean indicated pressures. When the rates of pressure rise are examined, it is seen that the curves are close to each combustion chamber up to a point and the pressure rise is higher in the NCC from this point on. Many reasons are effective here, such as improved combustion and increased homogeneity of the mixture. However, the point that needs to be emphasized here is that this increase is controllable. Increases in combustion rates cause NO emissions to increase. According to the optimum combustion law, slowing down the rate of pressure rise and reducing the combustion rate of the fuel is a suitable method to bring the NO emission increases to a controllable point.

4. Conclusions

In this study was carried out to analysis of the engine used the NCC bowl geometry and compare it with the SCC. The engine was operated for both bowl geometries at different engine speeds (1800, 1900, 2000, 2100 and 2200 rpm) and full load, and has the following conclusions as follow:

- The spray axis of the injector, using the φ angle (10°) where optimum results were obtained for both bowl geometries.
- The NCC geometry according to the SCC, it increased torque and fuel consumption decreased at all speeds. The torque increased by 6.32% and the specific fuel consumption decreased by 6.89% with NCC at 2000 rpm where the maximum torque was obtained.
- While CO, CO_2, soot emissions decreased, NO exhaust emissions increased because of the increase in combustion efficiency and chamber temperature in the NCC.
- In the bowl, both the lower base radius and the gradual narrowing of the bowl diameter caused the speed of the flame on the piston bowl.
- The maximum instantaneous heat release rate with the NCC geometry was 56.12 J/° at 385 °CA, while it was 52.54 J/° at 385 °CA with the SCC geometry at 2000 rpm engine speed. This situation is also reflected in the in-cylinder pressure values. This caused the rates of pressure rise to increase.
- Increased in-cylinder turbulence and shortened ignition delay durations improve mixture formation and combustion process.
- Looked at the mass combustion rates, thanks to the NCC geometry combustion earlier started and higher combustion velocity was obtained due to turbulence density, improved mixture formation and higher quality flame cernel.
- A combustion chamber with a directing feature affects the dynamics of both air and fuel particles, mixture formation, combustion and exhaust values. Especially by making long-term engine studies, ring, valve, etc. on the combustion chamber elements in the engine with such a combustion chamber.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Data availability

No data was used for the research described in the article.

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