

Article

Investigation into the Impact of Piston Bowl Size on Diesel Engine Characteristics with Changes in Fuel Injection Pressure and Boost Pressure

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Abstract: This study presents the effects of piston bowl size on the characteristics of a four-stroke single-cylinder diesel engine, which is considered in relation to changes in factors such as fuel injection pressure and turbocharger pressure. The study was carried out by 3D modeling using AVL Fire with an omega combustion chamber size and dimensions determined by the ratio between the diameter and depth of the piston bowl, which varies from 3.4 to 10.0. Additionally, the turbocharger pressure varies from 0.15 to 0.45 MPa at an engine speed of 1400 rpm and fuel injection pressure up to 300 MPa. The results show that the engine reaches the best values of indicated power, fuel efficiency, and a substantial decrease in emissions of nitrogen oxides at a turbocharger pressure from 0.25 to 0.35 MPa and with a ratio of the diameter to the depth from 7.8 to 10. However, the injection angle changes slightly, and the penetration depth and the tip velocity decrease with increasing boost pressure. While the piston bowl parameters only impact significantly on the tip velocity, the penetration and the spray angle are almost unchanged. In addition, the variation in the diameter of the combustion chamber has an influence on the fluctuation of the spray tip velocity and penetration.

Keywords: boost pressure; exhaust gas; fuel spray; piston bowl



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1. Introduction

Diesel engines are still the main source of power for construction machines, ships, agricultural machines, etc., because of their higher fuel efficiency and greater output power compared to gasoline engines. However, the emissions of diesel engines are still very high, especially particulate matter (PM) and nitrogen oxides (NO_x). To reduce the emission content of diesel engines, three main approaches are used. They include a redesign of and improvement in engine structures; installation of exhaust gas recirculation systems [1,2]; and biofuels and lubricant additives [3–6]. Emissions of NO_x and PM decrease and increase, respectively, if the emission recirculation rate, fuel injection pressure, and turbocharger pressure increase [7,8]. In addition, exhaust gas treatment on the exhaust pipe with a selective catalytic reduction (SCR) and diesel particulate filter (DPF) systems is a solution often chosen to improve the exhaust gas quality of diesel engines [9–11].

The enhancement of engine design is associated with an improvement in the fuel system, selection of the combustion chamber geometry, or increase in the boost pressure (p_{in}). The above solutions can be combined to achieve emissions quantities complying with increasingly stringent ecological standards [8].

The geometric shape of the combustion chamber enhances the rate of turbulent mixing when the piston approaches the top dead center (TDC) [12]. Furthermore, the turbulent movement in the combustion chamber improves the air–fuel mixture, which reduces the time of ignition delay [13].

Improving fuel systems is a basic development trend to perfect diesel engines, especially common rail (CR) fuel injection systems. This research direction includes increasing the injection pressure [14] or ensuring multiple injections [15–17]. Additionally, controlling fuel distribution in combustion chamber zones or using a lot of fuel are two possible solutions [18,19].

Pulse duration and pressure are two factors that define the fuel injection process [20]. The fuel injection process is also strongly dependent on the wave phenomenon in the high-pressure pipeline, especially with multiple injections [17,21,22].

Studies have demonstrated that the fuel injection process on diesel engines greatly affects the combustion process and diesel engine parameters. Increasing fuel injection pressure needs to be combined with other structural modifications of the injectors, the combustion chamber, and other engine advances that reduce the specific fuel consumption by up to 4% [21]. At the same time, the fuel injection pressure has a direct effect on the vaporization and air–fuel mixing efficiency in the combustion chamber [17,18,23]. Under high-load conditions, engine power is related to the penetration rate of the fuel spray and the combustion rate of the fuel–air mixture in the combustion chamber [24,25]. According to Matteo Imperato and et al., when increasing the fuel injection pressure from 150 to 240 MPa for large-sized diesel engines, the results show that ignition delays are reduced by 40%, engine efficiency is increased by 4%, and fuel combustion time in the combustion chamber is reduced. In addition, the amount of NO_x emission at 150 MPa is virtually unchanged [26]. Other studies also show that the amount of soot will decrease when the fuel injection pressure increases and the injector hole diameter decreases [27]. With different nozzle diameters and increasing injection pressure from 100 to 300 MPa, the results present that injection pressure has a small effect on the properties of the mixture, but plays an important role in providing enough energy to fuel vaporization and spray lengths by using micro-hole diameters. The micro-diameter nozzle combined with high pressure is a better combination than conventional nozzles, which increases the rate of vaporization and reduces the time of homogenization of the mixture in the combustion chamber. Injection pressures of 300 MPa and an injection hole size of 0.08 mm have shown high efficiency in increasing the speed of turbulence and reducing fuel particle sizes. This is very important in reducing the mixture preparation time and rich mixture formation for the premixed compression ignition (PCI) combustion [28–30]. Besides, smoke is lower at constant NO_x emissions with an increase in the injection pressure from 200 to 250 MPa. Alternatively, the increase in the injection pressures significantly reduced fuel consumption at constant NO_x and smoke emissions. From 250 to 300 MPa, the effect of injection pressure on smoke and fuel consumption is limited at higher NO_x levels [31]. With a fuel injection pressure of 320 MPa, the duration of combustion is shorter, while the ignition delay changes very little if the injection fuel mass and the fuel injection pressure are the same [32].

When the turbocharger is used, the engine power increases and specific fuel consumption decreases. However, its main disadvantages are the high thermal factor of the engine parts, the increase in the price of a diesel engine, and its maintenance. Additionally, the maximum value of the boost pressure also depends on the engine design [33,34]. The combination of turbocharging and increasing fuel injection pressure improves engine performance. This is a solution that can be chosen as a combination of two factors. It has been shown that density, temperature and air pressure in the cylinder increase as the fuel injection pressure is low or medium and the turbocharging pressure rises. This significantly affects fuel injection characteristics [35–37].

The purpose of this study is to find out the effect of the geometric parameters of the combustion chamber and the boost pressure on the combustion characteristics when using an ultra-high fuel injection pressure of 300 MPa. The shape of the piston bowl was determined by the ratio between its diameter and depth of the combustion chamber when the compression ratio of a single-cylinder four-stroke diesel engine is fixed at 15.4. An electro-hydraulic nozzle with eight holes of 0.1 mm diameter each was used. Simulation was performed by AVL Fire software version 2020, which is developed by AVL Ltd. (Graz,

Austria). The results of this study are the basis for designing, improving, and optimizing diesel engines to increase engine power and reduce fuel consumption and emissions.

2. Simulation Model

2.1. Three-Dimensional CFD Simulation Models

For research purposes, a 3D model was built in AVL Fire. In this model, the ECFM-3Z combustion model is selected along with the ignition, emission, turbulence models, etc. The selected submodels are shown in Table 1 [38].

Table 1. Sub-models of 3D model in AVL Fire [38].

Sub-Models	Name of Sub-Models
Combustion model	ECFM-3Z
Spray model	Wave
Turbulent dispersion model	Enable
Spray wall interaction model	Walljet1
Droplet Evaporation model	Dukowicz
Turbulence model	k-zeta-f
Ignition model	Auto-ignition
NO formation	Extended Zeldovich
Soot formation	Kinetic Model

2.2. Simulation Object and Initial Conditions

The engine used in the study was 1CH12/13. This is a single-cylinder diesel engine with a stroke of 130 mm, cylinder diameter of 120 mm and compression ratio of 15.4. In addition, the engine displacement is 1.47 L with 4 valves (2 intake valves and 2 exhaust valves). The main parameters of the diesel engine are shown in Table 2.

Table 2. The 1 CH 12/13 engine Specifications [38].

Parameter	Value
Bore, (mm)	120
Stroke, (mm)	130
Connecting rod length, (mm)	224
Number of cylinders (-)	1
Displacement, (L)	1.47
Compression ratio (-)	15.4
Number of intake valves (-)	2
Intake valves open, deg. BTDC	10
Intake valves close, deg. ABDC	15
Number of exhaust valves	2
Exhaust valves open, deg. BBDC	50
Exhaust valves close, deg. ATDC	6
Number of nozzle holes (-)	7
Nozzle hole diameter, (mm)	0.1

The engine's original combustion chamber was measured directly when removing the engine. This engine uses an omega combustion chamber. The dimensions were determined and re-drawn as shown on the longitudinal section in Figure 1a.

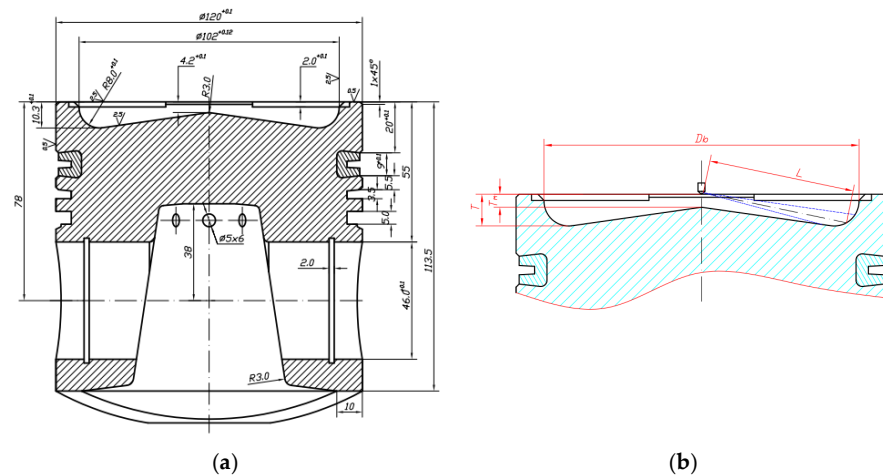


Figure 1. Piston shape of the 1CH 12/13 engine: (a) longitudinal section of the piston; (b) basic dimensions of the piston.

In this study, the piston shape was kept as omega. We investigated the change in the depth of the combustion chamber and the diameter of the combustion chamber under the condition that the engine's compression ratio remained unchanged.

Under that requirement, 5 survey modes were selected with corresponding piston sizes, as shown in Table 3. the parameters T , T_m , D_b , and L are as shown in Figure 1b.

Table 3. Main parameters of the combustion chamber in the model.

Cases	Diameter of the Combustion Chamber, D_b (mm)	Depth of the Combustion Chamber, T (mm)	Depth of the Combustion Chamber at the Center, T_m (mm)	Length of the Fuel Spray, L (mm)
1	67	19.8	16.0	73.4
2	74	16.4	13.0	74.5
3	80	14.4	10.1	79.6
4	90	11.5	8.0	88.1
5	100	10.0	4.6	98.0

Simulation conditions include temperature, pressure of inlet air, cylinder and piston temperatures, etc., which are entered into the model based on the actual measured values of the engine in test modes. These values respond to engine conditions when operating stably. The engine speed was selected to be 1400 rpm, which corresponds to the mode that promotes the maximum torque of the low-speed diesel engine group. Within the scope of the study, the change in inlet air pressure was also investigated, and it increased from 0.15 to 0.45 MPa. Table 4 presents the boundary condition values of the 3D simulation model in AVL Fire.

Table 4. Simulation conditions in the 3D model.

Parameter	Value
Engine speed, (rpm)	1400
Boost pressure, (MPa)	0.15 to 0.45
Intake air temperature, (K)	307
Cylinder head temperature, (K)	550
Cylinder wall temperature, (K)	475
Piston top temperature, (K)	575
Fuel injection temperature, (K)	330
Fuel injection pressure, (MPa)	300
Fuel mass per cycle, (mg)	60

Fuel injection characteristics were determined experimentally on a non-engine test bed. The fuel injection pressure varied from 50 to 300 MPa in steps of 50 MPa. Additionally, the fuel mass per cycle was 60 mg, the injection signal duration was 0.7 ms, and the injection pressure was 300 MPa. The fuel injection characteristics of the nozzle and the mode chosen for simulation are displayed in Figure 2.

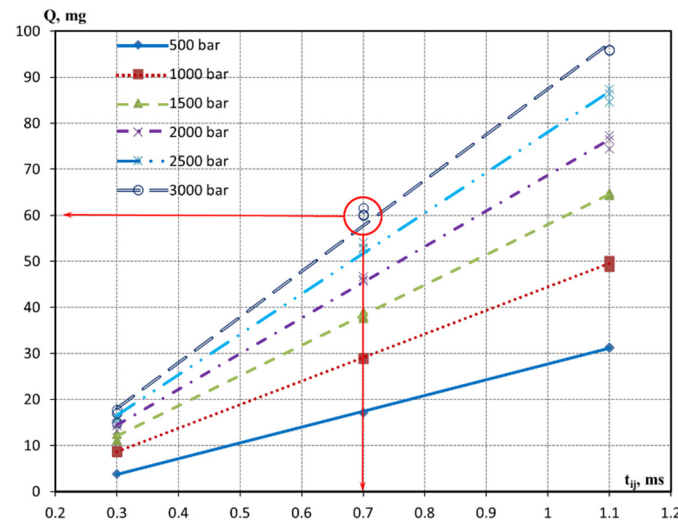


Figure 2. Fuel injection characteristics and the selection mode ($p_{ij} = 300$ MPa, $Q = 60$ mg).

2.3. Mesh Independence Analysis

The size and number of the mesh affect the results of calculations in the CFD model. This study analyzed the effect of the mesh size on the results with 3 cases including coarse, medium, and fine mesh. Figure 3 shows 3 surveys' meshing results with parameters and calculation time at the top dead center (TDC) position. The mesh values and the calculation time are shown in Table 5 in the case where the combustion chamber diameter is 100 (mm).

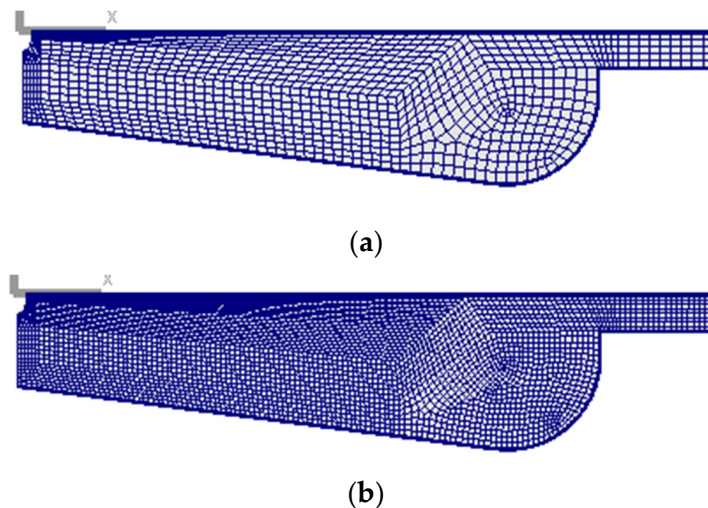


Figure 3. Cont.

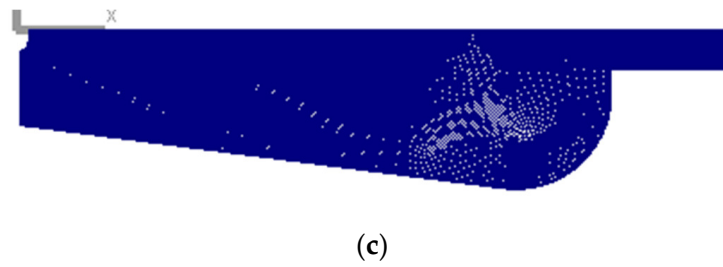


Figure 3. The cases of the computational mesh: (a) coarse; (b) medium; (c) fine.

Table 5. The mesh values and the calculation time.

Mesh Resolutions	Coarse	Medium	Fine
Average cell size	1 mm	0.5 mm	0.2 mm
Number of faces	1518	4512	25,159
Number of triangle faces	0	0	0
Number of boundary faces	183	329	807
Total number of cells	105,768	363,452	3,027,552
Calculation time with computer 64 bit, Intel® Core™i5-3210M CPU @ 2.50 GHz	(12–16) h	(32–36) h	(80–82) h

The comparison results between mean pressure, mean temperature, mean NO mass fraction, and mean CO₂ mass fraction in three cases are presented in Figure 4.

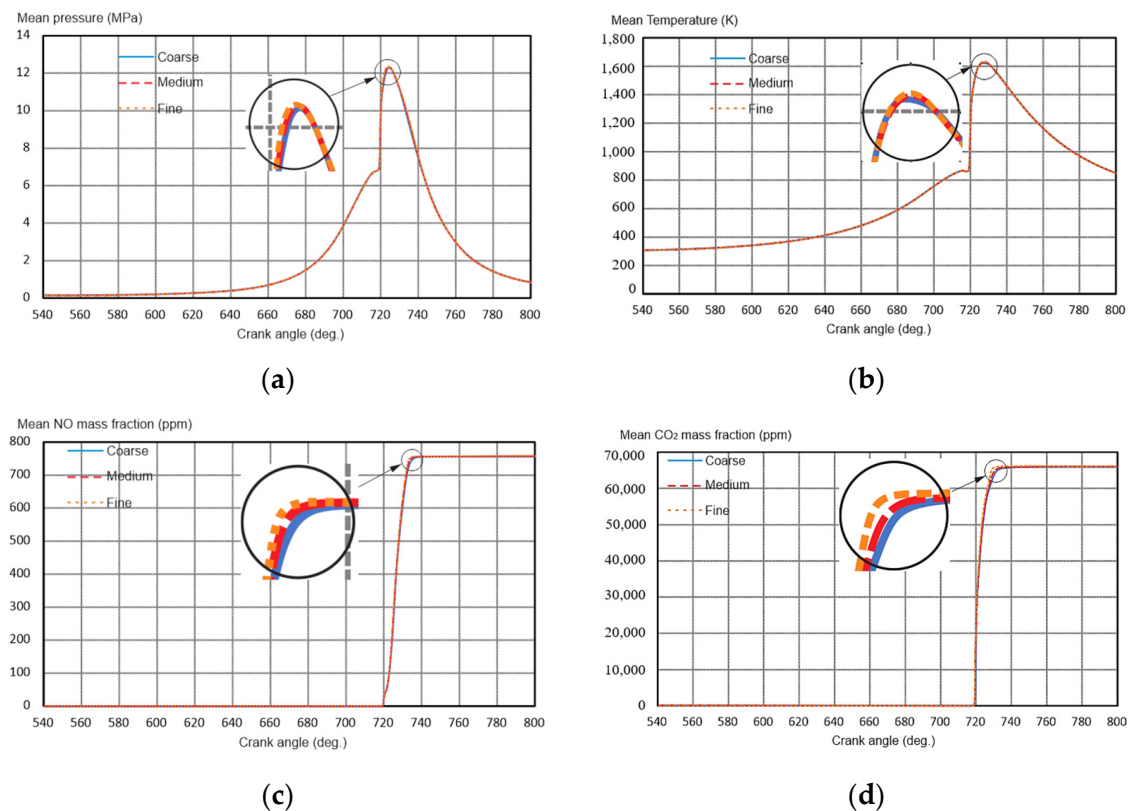


Figure 4. The CFD results of the simulations with three different meshes: (a) mean pressure; (b) mean temperature; (c) mean NO mass fraction; (d) mean CO₂ mass fraction.

It can be seen that the simulation results gave higher accuracy with the fine mesh, especially at the position where the graph is bent, but the calculation time was extended for each case, while the results were relatively accurate and showed a reduced calculation time with the medium mesh. Therefore, the median mesh was selected in this study. The details of the chosen mesh for each of the combustion chambers is given in Table 6.

Table 6. Mesh parameters used for combustion model studies when the piston was at the bottom dead center (BDC).

Mesh Parameters	$D_b = 67 \text{ mm}$	$D_b = 74 \text{ mm}$	$D_b = 80 \text{ mm}$	$D_b = 90 \text{ mm}$	$D_b = 100 \text{ mm}$
Average cell size	0.5 mm	0.5 mm	0.5 mm	0.5 mm	0.5 mm
Number of faces	47,395	47,004	31,747	27,885	29,135
Number of triangle faces	0	0	0	0	0
Number of boundary faces	885	913	719	679	679
Total number of cells	540,220	551,564	378,036	353,316	363,452

Figure 5 shows the meshing of the model at the TDC position in the case of 67 mm and 100 mm diameters.

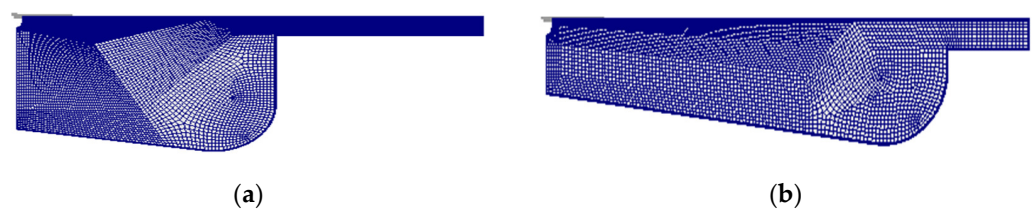


Figure 5. Meshing in the calculated model in the case of 67 mm and 100 mm diameters: (a) $D_b = 67 \text{ mm}$; (b) $D_b = 100 \text{ mm}$.

2.4. Check the Accuracy of the 3D Model in AVL Fire

To validate the combustion model of the 1CH 12/13 engine established in AVL Fire, the in-cylinder pressure and the rate of heat release (RoHR) values were selected as the value basis for comparison with the those values in an experiment under the same conditions. The original piston bowl shape with $D_b = 100 \text{ (mm)}$ and $T = 10 \text{ (mm)}$ was used in experiments with the original engine.

A schematic diagram of the experimental setup is shown in Figure 6. The devices are installed on the experiment engine and the measurement errors of the test equipment are shown in Table 7.

Table 7. The devices on the experiment engine.

Measuring Devices	Name of Measuring Devices	Measurement Range	Measurement Error
Dynamometer	Electric dynamometer made in Germany, typ. GPFc 13 h	0–6000 rpm	-
Pressure in cylinder chamber	A Piezo sensor of MADI design is connected to the AVL A03 amplifier	0–20 MPa	$\pm 5\%$
Crankshaft speed	Electronic tachometer	50–9999 rpm	$\pm 0.2 \text{ rpm}$
Effective torque	Elastically deformable beam with electrical resistance strain gauge		
Fuel pressure in the rail	DMP 304—BD sensor	0–400 MPa	$\pm 0.5\%$
Mass of fuel consumption per hour	mass fuel flow meter ART-2	10–99.99 kg/h	$\pm 0.3\%$

Table 7. Cont.

Measuring Devices	Name of Measuring Devices	Measurement Range	Measurement Error
Volume of air per hour	Volumetric flowmeter RG 600-1	0–600 m ³ /h	±2%
Intake pressure	Pressure transducer NPP “Elemer” AIR-10N-DIV (model 1360) which was connected to the technological meter-regulator NPP “Elemer” IRT5320N	0–2.4 MPa	±0.5%
Intake temperature	Thermal converter NPP “Elemer” TS-1088L/1 which was connected to the technological meter-regulator NPP “Elemer” IRT5323N	−50–+200 °C	±0.3 °C
NO _x in exhaust gases	Gas analyzer NPO “Eco-Intech” Infralight-11P	0–1000 rpm 1001–2000 rpm	±5%
Smoke of exhaust gases	Opacimeter NPO “Eco-Intech” Infralight-11D	0–100%	±2%

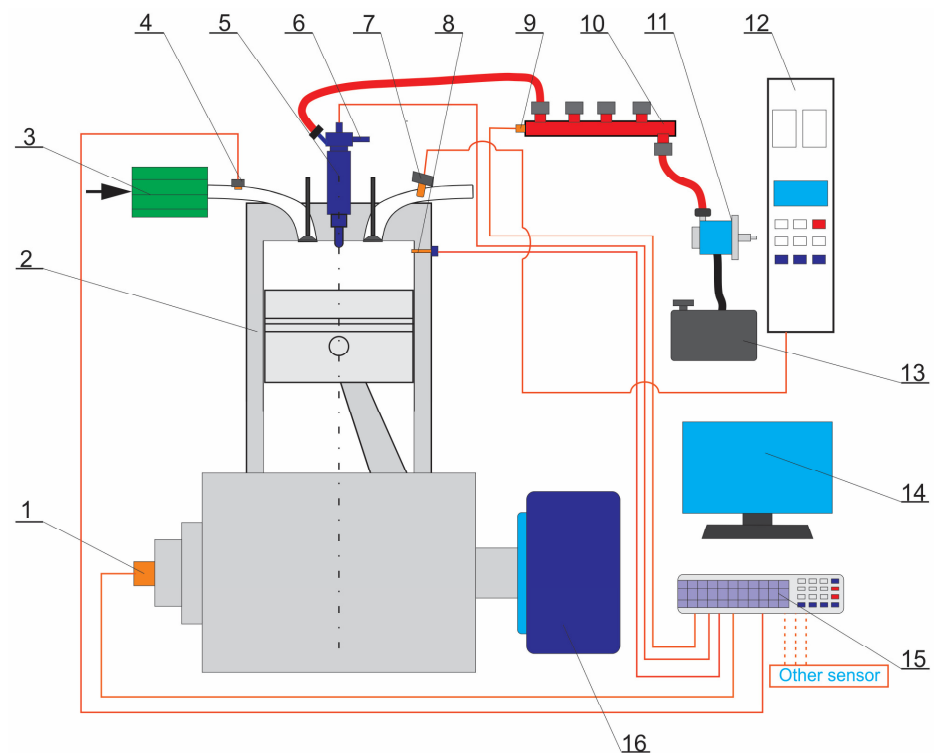


Figure 6. Schematic diagram of the experimental setup. 1—encoder; 2—diesel engine; 3—air filter; 4—air flow sensor; 5—fuel injector; 6—line of back fuel; 7—exhaust gas collection; 8—cylinder pressure sensor; 9—fuel pressure sensor; 10—rail; 11—height pressure pump; 12—exhaust analyzer; 13—fuel tank; 14—computer for control and analyzer; 15—adapter; 16—dynamometer.

The simulation and experimental results were selected to compare the reliability of the model as follows.

First, for the engine operating mode in which the engine speed reached 1400 rpm, the fuel injection pressure was 150 MPa at 23 deg. of crank angle BTDC, and the average effective pressure was limited to 0.9 MPa. With this mode, the maximum pressure value in the combustion chamber was 10.2 MPa and the maximum RoHC value was 48 J/deg. CA.

When the injection pressure increased to 300 MPa, the fuel mass per cycle was 45 mg, the intake air pressure was 0.07 MPa, and a RoHR value of 225 J/degree was obtained.

Additionally, with an injection pressure of 300 MPa, the fuel mass per cycle increased by $Q = 60$ mg and $Q = 120$ mg, intake air pressure increased by 0.1 bar and 0.15 MPa, respectively, and we also obtained the pressure in the combustion chamber and the corresponding RoHR, as shown in Figure 7. The RoHR value obtained in these experiments was calculated using Labview software version 2018 (single zone + heat transfer).

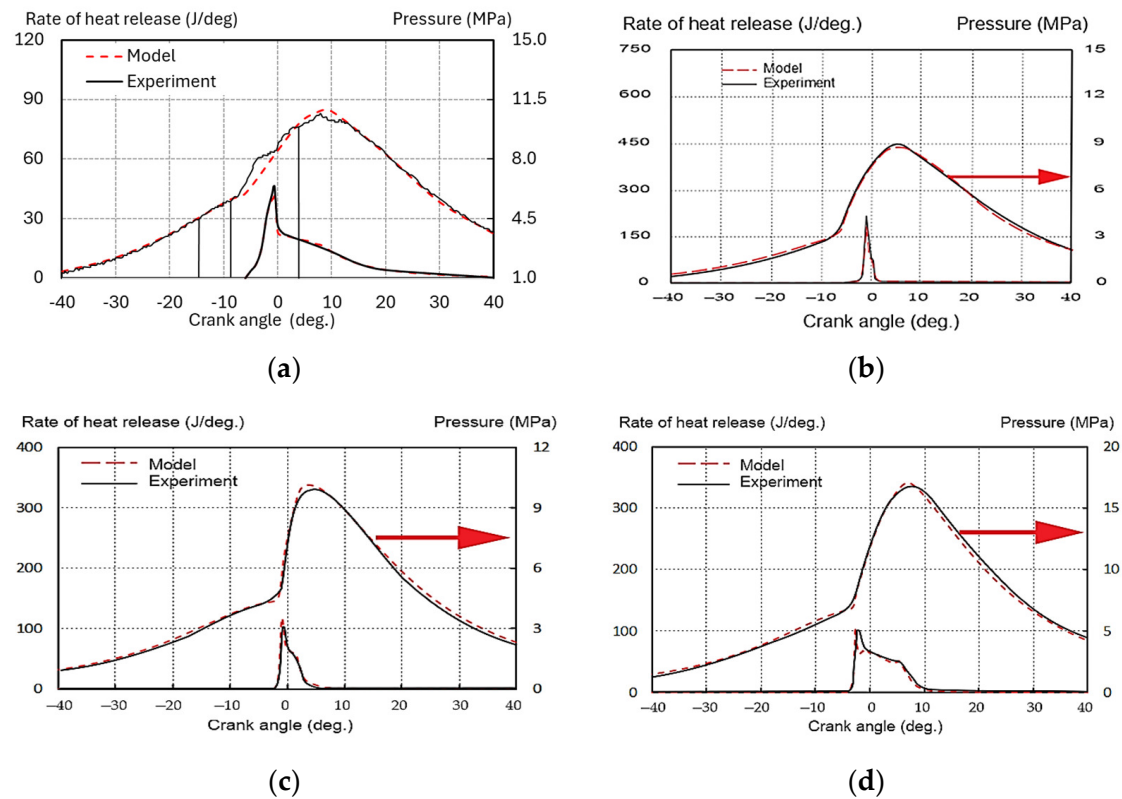


Figure 7. Checking the accuracy of the 3D model in the AVL Fire with experimental results: (a) 150 MPa of injection pressure and 0.1 MPa of the inlet pressure; (b) 300 MPa of injection pressure, $Q = 45$ mg and 0.07 MPa of the inlet pressure; (c) 300 MPa of injection pressure, $Q = 60$ mg and 0.1 MPa of the inlet pressure; (d) 300 MPa of injection pressure, $Q = 120$ mg and 0.15 MPa of the inlet pressure.

With the above results, it was found that the error value is less than 5%. Therefore, the 3D models in AVL Fire were reliable and suitable to be used for the purpose of this study.

3. The Research Results and Discussion

This study was carried out at an engine speed of 1400 rpm, fuel injection pressure of 300 MPa, and with the boost pressure varying from 0.15 to 0.45 MPa with a change step of 0.1 MPa. The simulation was performed when the D_b/T ratio varied from 3.4 to 10 at $\varepsilon = 15.4$, which corresponded to a change in D_b from 67 to 100 mm.

During the simulation, the start of injection (SoI) with each speed mode needed to be optimized to achieve the lowest indicated fuel consumption (g_i) and the biggest indicated power (N_i). Figure 8 illustrates the result of the SoI optimal with a fuel injection pressure of 300 MPa and engine speed of 1400 rpm. From the results, the SoI optimal for this case was chosen as 5 degrees of crank angle BTDC. At this point, the N_i obtained was the largest, and the g_i was the smallest.

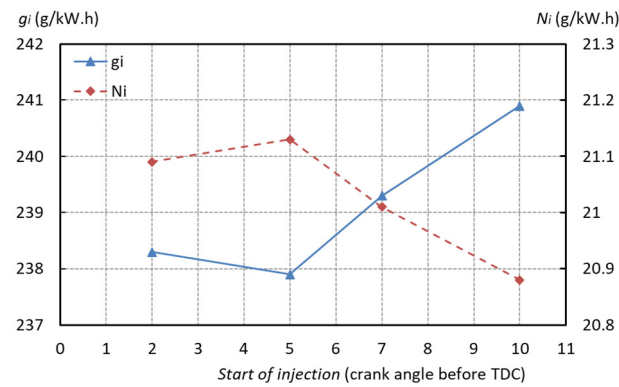


Figure 8. The optimal SoI at engine speed 1400 rpm and fuel injection pressure of 300 MPa.

3.1. Influence of Geometric Parameters of the Combustion Chamber and Boost Pressure on the Characteristics of the Injection Fuel Spray

Figures 9 and 10 demonstrate the characteristics of the development of the fuel spray as a function of time.

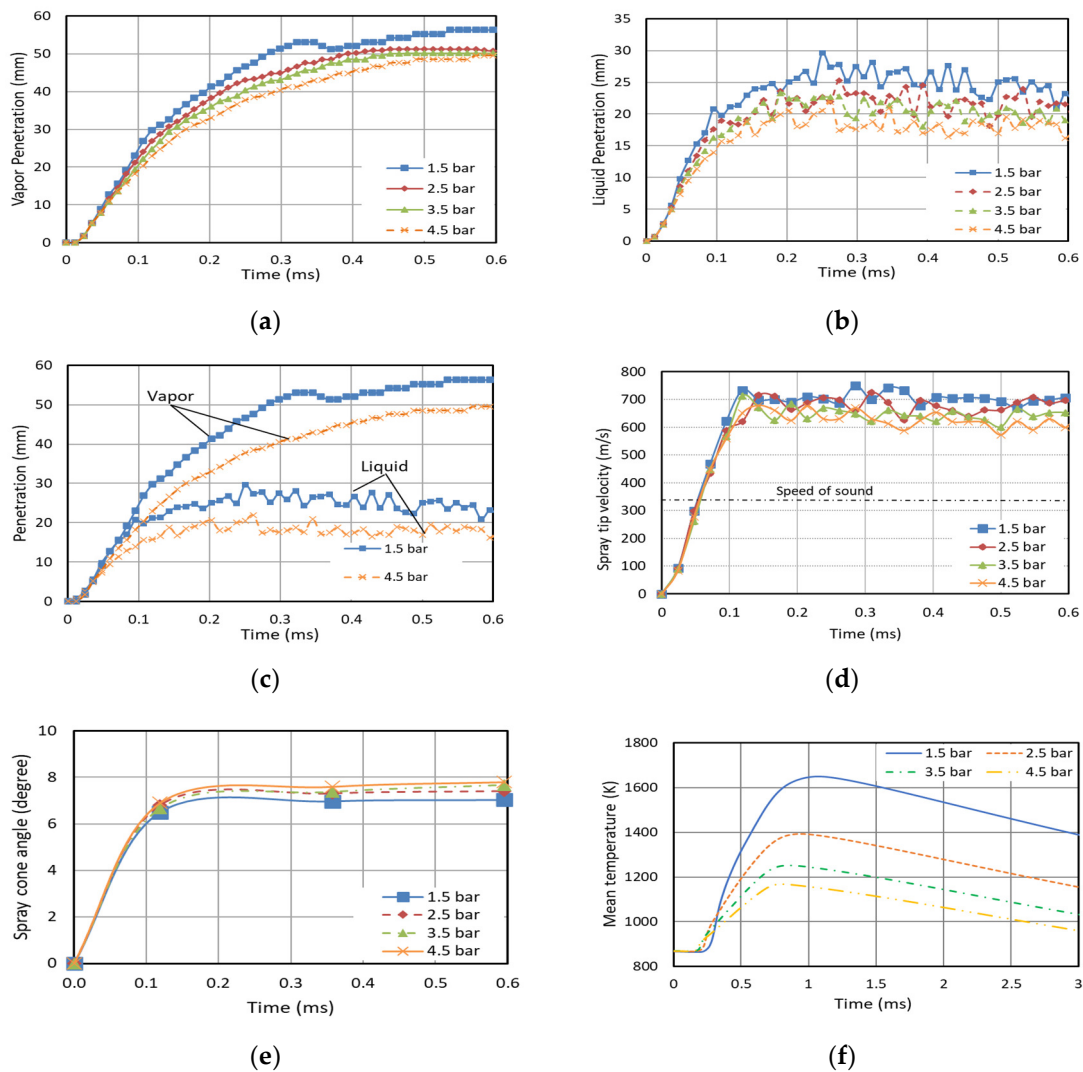


Figure 9. Effect of the boost pressure on the spray characteristics and the mean temperature at an engine speed of 1400 rpm, $p_{ij} = 300$ MPa, $Q = 60$ mg and $D_b = 100$ mm: (a) vapor penetration; (b) liquid penetration; (c) penetration; (d) spray tip velocity; (e) spray cones angle; (f) mean temperature.

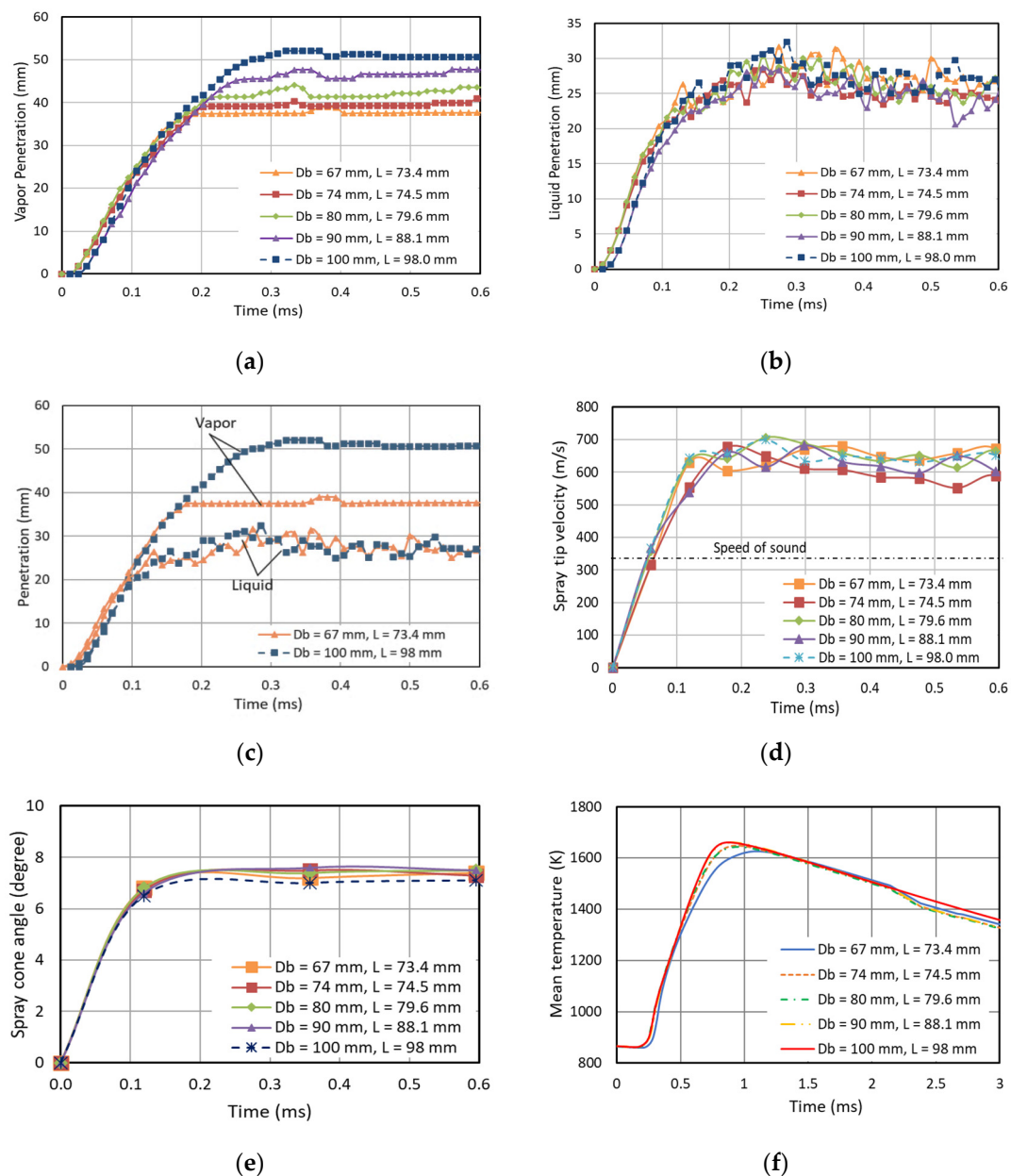


Figure 10. Effect of the combustion chamber geometry on the spray characteristics and the mean temperature at an engine speed of 1400 rpm, $p_{ij} = 300$ MPa, $Q = 60$ mg and 0.15 MPa of the boost pressure: (a) vapor penetration; (b) liquid penetration; (c) penetration; (d) spray tip velocity; (e) spray cone angle; (f) mean temperature.

At the initial stage of the injection (up to 0.05 ms), the boost pressure and the D_b/T ratio had a weak effect on the injection process (Figures 9 and 10). The oscillation of both the liquid and the vapor phases' penetration and the spray tip velocity started after 0.1 (ms) time from the SoI. The length of penetration was limited by fuel evaporation and its movement in the gaseous state. As the boost pressure increased, the maximum values of the penetration length and the spray tip velocity decreased due to greater air resistance (Figure 9). The boost pressure defines the pressure in the engine cylinder; therefore, the higher the boost pressure, the higher the spray deceleration. In this case, the number of fuel droplets at the periphery and on the shell of the spray increased. As a result, the width of the tip edge increased—in other words, the spray cone angle increased.

The values of the spray penetration and the spray cone angle changed significantly when the boost pressure increased from 0.15 to 0.25 MPa, while those values changed slightly with an increase in the boost pressure from 0.25 MPa to 0.35 MPa.

With an increase in D_b (D_b/T ratio), the spray parameters (liquid penetration) varied a little (Figure 10). This behavior can be explained by the fact that the pressure in the combustion chamber and the fuel injection pressure do not depend on D_b/T (because the values of all the fuel injection pressure, the boost pressure, and the compression ratio ε are constant). As can be seen in Figure 8, the spray cone angle increases, but both the fuel spray tip velocity and the penetration depth decrease simultaneously with the start of combustion (SoC), taking place earlier in the cycle owing to the higher air–fuel vapor mixing rate at the high boost pressure of 0.45 MPa and high temperature inside the cylinder. The D_b/T value affects the oscillation results of the spray penetration and the spray tip velocity. When increasing the combustion chamber diameter, their maximum value also grows, while their values decrease when the boost pressure decreases (Figure 9).

In fact, the angle between the axis of the spray hole and the axis of the fuel injection nozzle (φ) changes as the D_b/T ratio varies. This angle should be in accordance with the diameter of the combustion chamber to ensure that fuel is injected into the space in the combustion chamber. The amount of fuel injected should not be too close to the cylinder head or too much on the surface of the combustion chamber. To achieve that, it is necessary to optimize this angle with the maximum value of the indicated power. In the study, this angle was also optimized for each case of the diameter of the combustion chamber. It can be seen that the φ angle changes according to the change in the diameter of the combustion chamber. Furthermore, the length of the nozzle holes increases too, while the wall thickness of the nozzle tip does not change (Figure 11). This causes disturbances in the fuel flow and the spray parameters. The maximum value of the spray tip velocity in all cases (Figures 9 and 10) exceeds the speed of sound.

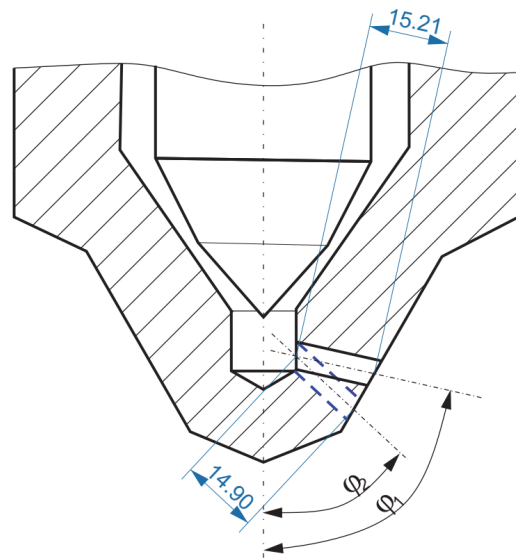


Figure 11. The dependence of the length on the nozzle angle.

3.2. Effect of the Geometric Parameters of the Combustion Chamber and the Boost Pressure on Combustion Characteristics

Figures 12 and 13 illustrate the dependencies of the excess air ratio α , the maximum pressure value $p_{\max}$, the maximum temperature value $T_{\max}$, as well as the maximum pressure rise rate $(dp/d\varphi)_{\max}$ on the value of D_b/T ratio and the boost pressure.

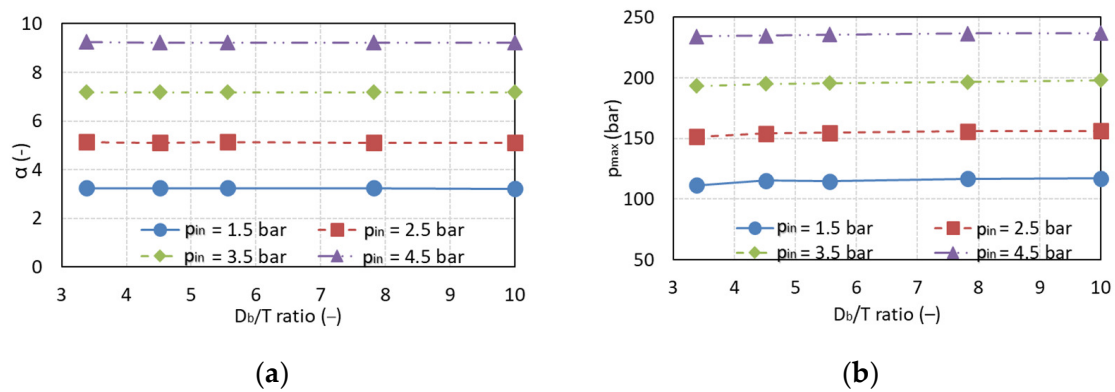


Figure 12. Dependence of the air–fuel ratio α and p_{max} on the D_b/T ratio ($p_{ij} = 300$ MPa): (a) dependence of the air–fuel ratio α ; (b) dependence of p_{max} .

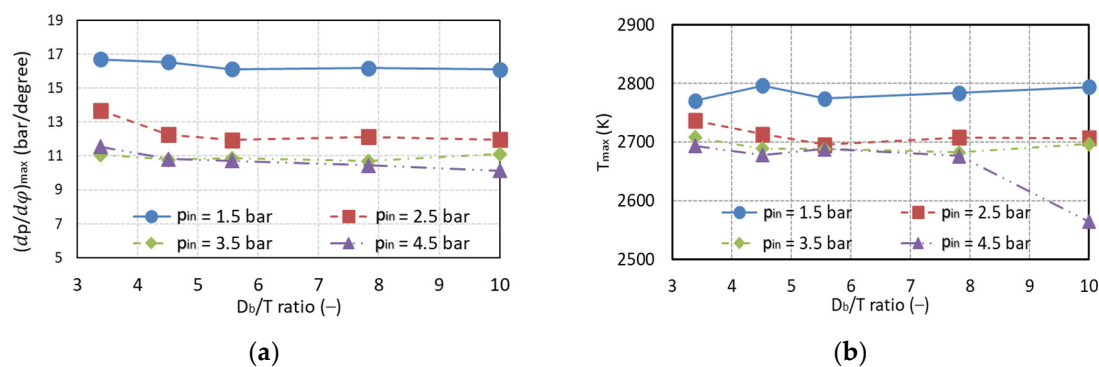


Figure 13. Dependence of $(dp/d\phi)_{max}$ and T_{max} on the D_b/T ratio ($p_{ij} = 300$ MPa): (a) dependence of $(dp/d\phi)_{max}$; (b) dependence of T_{max} .

The excess air ratio does not depend on the D_b/T ratio, but it has a weak effect on p_{max} (Figure 12) at the ϵ value unchanged condition.

When the value of the fuel injection in a cycle is unchanged ($Q = 60$ mg), the excess air ratio α and the maximum pressure in the cylinder p_{max} increase due to the significant rise in the boost pressure (Figure 12b). The excess air ratio α rises because the mass of the air entering the cylinder increases. Those variations are determined by the increase in the in-cylinder pressure at the end of the compression process. The α and p_{max} values grow by 2.84 and 2.1 times, respectively, when the boost pressure increases from 0.15 to 0.45 MPa (Figure 12a).

The increase in the boost pressure has a positive effect on the mixing rate (the maximum value of the spray penetration decreased but the cone angles of the spray increased, Figure 9); therefore, the maximum increment rates in the in-cylinder pressure $(dp/d\phi)_{max}$ and T_{max} decrease as well (Figure 13). Thus, at the value of $D_b/T = 10$, the boost pressure variation from 0.15 to 0.45 MPa leads to a decrease of 38% in $(dp/d\phi)_{max}$ and of 9% in T_{max} . The decrease in the $(dp/d\phi)_{max}$ value has a positive effect on the production of NO_x emissions and noise in the operating process of a diesel engine.

The growth in the boost pressure has a higher effect on the $(dp/d\phi)_{max}$ than that on the T_{max} value. As the boost pressure increases, especially above 0.35 MPa, its role in reducing the values of both $(dp/d\phi)_{max}$ and T_{max} decreases, especially significant in T_{max} .

The highest values of $(dp/d\phi)_{max}$ and T_{max} were achieved at the smallest diameter of the combustion chamber (Figure 13) and the biggest depth value of the combustion chamber ($D_b/T = 3.4$), which is associated with more fuel striking against the combustion chamber wall until the autoignition occurs; however, the only exception for T_{max} is at $p_{in} = 1.5$ bar.

As the diameter of the combustion chamber grows, the $(dp/d\phi)_{\max}$ value reasonably or slightly decreases. At $p_{in} = 0.45$ MPa, there is a gradual decrease in $(dp/d\phi)_{\max}$ with an increase in D_b . Actually, the maximum pressure increment rate decreases by 13% when the D_b/T ratio rises from 3.4 to 10. The $(dp/d\phi)_{\max}$ value also goes down significantly by 12.5% at $p_{in} = 0.25$ MPa and within the variation range of the D_b/T ratio from 3.4 to 5.6.

To explain the results presented in Figure 13 in more detail, the characteristics of the RoHR (Figure 14) and the in-cylinder temperature distribution (Figures 15 and A1 in Appendix B) at different diameters of the combustion chamber are displayed.

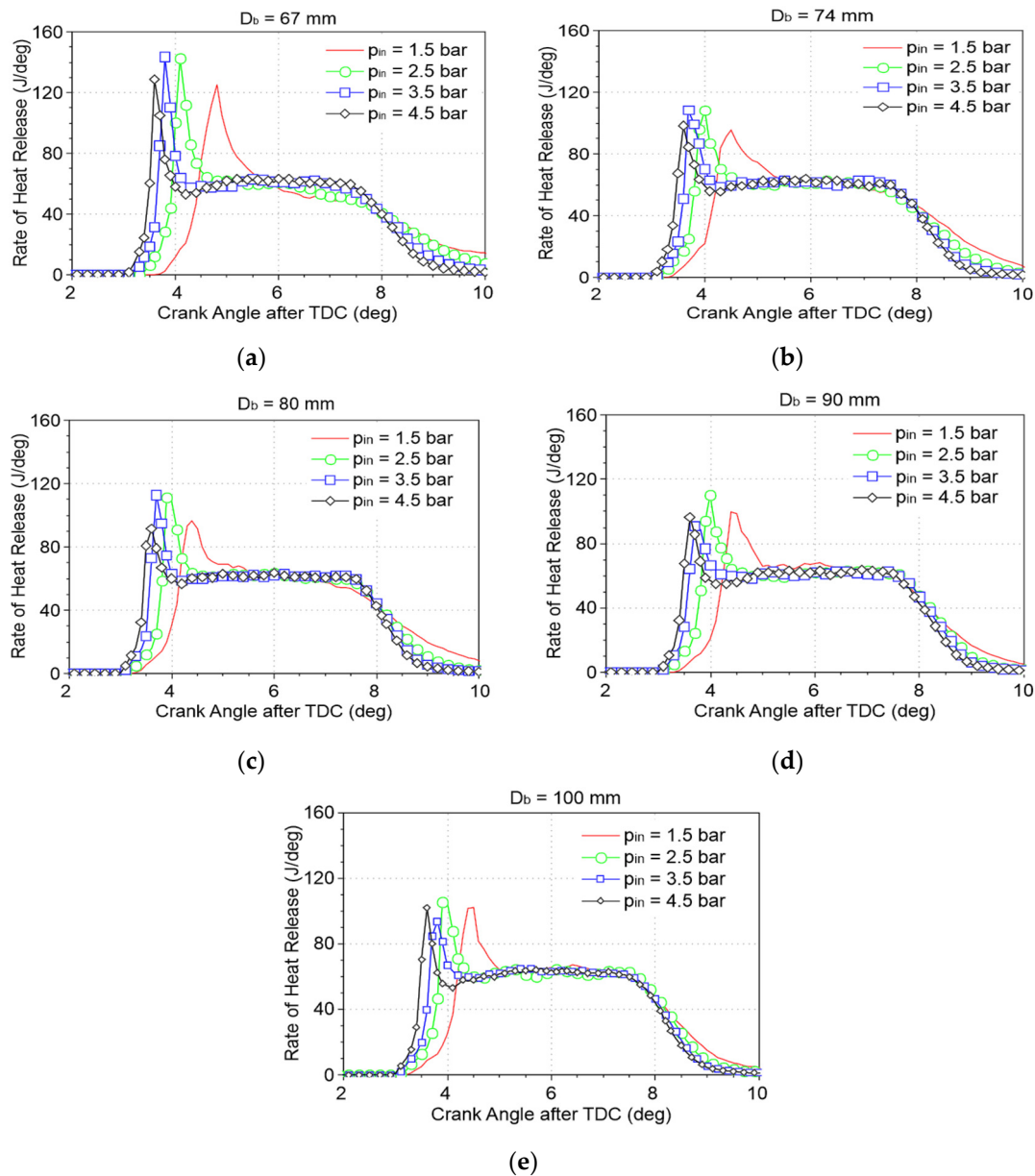


Figure 14. Variation of heat release rate versus crank angle degrees (CADs) for various boost pressure values p_{in} and the combustion chamber diameters D_b : (a) $D_b = 67$ mm; (b) $D_b = 74$ mm; (c) $D_b = 80$ mm; (d) $D_b = 90$ mm; (e) $D_b = 100$ mm.

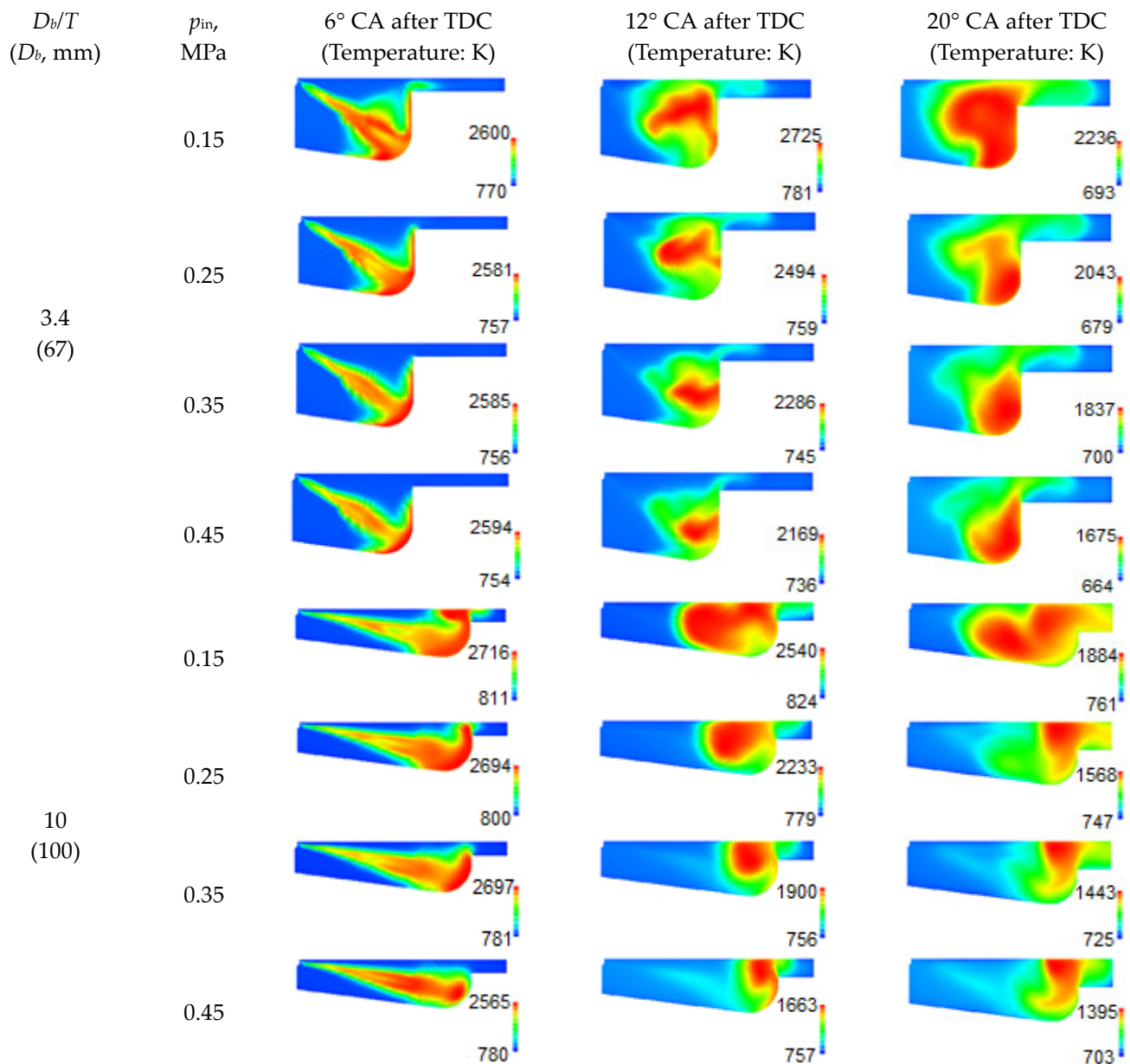


Figure 15. In-cylinder temperature distribution with $D_b/T = 3.4$ and $D_b/T = 10$.

It is known that the fuel spray parameters are affected by the following factors. (1) The more that fuel injection pressure increases, the greater the fuel injection speed and the energy of the spray movement. In this study, the fuel injection pressure is kept constant; therefore, it does not affect the spray rate. (2) Increasing the boost pressure and compression ratio leads to deceleration of the spray (an increase in the width of the tip edge) and creates more fuel droplets; they are stalled on its periphery and the shell. The fuel injection pressure determines the energy of movement, while the boost pressure determines the energy of the spray deceleration. (3) Temperature of fuel flow in the cylinder and the boost pressure rise, which makes the fuel spray warm up and accelerates the ignition. (4) The bigger the diameter of the combustion chamber is, the longer the spray of the fuel, which requires more time for its warmup, ignition, and combustion in the volume [39,40].

Figure 13 shows that the spray tips move and quickly reach the wall of the combustion chamber. The smaller the diameter of the combustion chamber is, the greater the amount of the fuel portion which is injected spreads along the cylinder wall. This fuel warms up slowly (for the beginning of the heat release rate characteristics, see Figure 14), evaporates

from the hot wall surface, ignites, and burns near the parietal zone. The heat release process is delayed, which can be seen at the end of the rate of heat release.

When the boost pressure is 0.25 MPa, more fuel ignites. The fuel droplets warm up faster; therefore, they lose more energy when the spray moves. As a result, less fuel (compared to $p_{in} = 0.15$ MPa) falls on the cylinder wall, which results in more heat by the fuel spray from the burned fuel bring into the combustion chamber. Usually, the fuel spray absorbs more heat when it spends more time in the combustion chamber volume due to a higher total surface area of the spray fuel droplets directly exposed to the hot in-cylinder compressed air charge. However, the heat transfers from the gases to the fuel spray depend upon the injection pressure, combustion chamber design, pressure, temperature, and many variable parameters to be considered in each specific case. The fuel that falls on the cylinder wall evaporates and ignites quickly (a more rapid onset of RoHR, Figure 14). The combustion duration is still quite long, although it is less than with the value $p_{in} = 0.15$ MPa.

With a further increase in the boost pressure from 0.35 to 0.45 MPa, the more fuel warms up and ignites in volume, and the little fuel drops on the wall of the combustion chamber (Figure 15). As a consequence, the combustion process passes faster. In the beginning, the RoHR is higher, and the combustion process is shorter (Figure 14).

Thus, the boost pressure plays a role as a distributor to separate the amount of the combustion fuel in the spray (volumetric mixture formation process) and near the wall of the combustion chamber (near-wall mixture formation process). At the volumetric mixture formation condition, the amount of burned fuel at the start of the combustion process rises with increasing boost pressure, and the fuel concentration is higher near the wall of the combustion chamber at the end of the combustion process.

As the diameter of the combustion chamber grows, the length of the fuel spray also increases, which takes longer than time for warming up, ignition and burning. The less fuel that falls on the combustion chamber wall, the more fuel that is burned under the volumetric mixture formation conditions, especially at a lower boost pressure value of $p_{in} = 0.15$ MPa, compared with the RoHR curves of $D_b = 67$ and $D_b = 100$ mm (Figure 13b). The reciprocal change of D_b and T strongly affects both the shape and the movement of the burning cloud in the middle and at the end of the combustion process.

3.3. Economic and Technical Indicators

The indicated power increases with increases in the boost pressure for all cases of the D_b/T ratio, while the indicated specific fuel consumption (ISFC) decreases. The possible reason is that when the boost pressure increases, the maximum pressure of the cycle also goes up if the same quantity of fuel is injected, and, therefore, the work carried out in the cycle during the expansion process is greater under the condition that the displacement and the volume are unchanged (Figures 16 and 17).

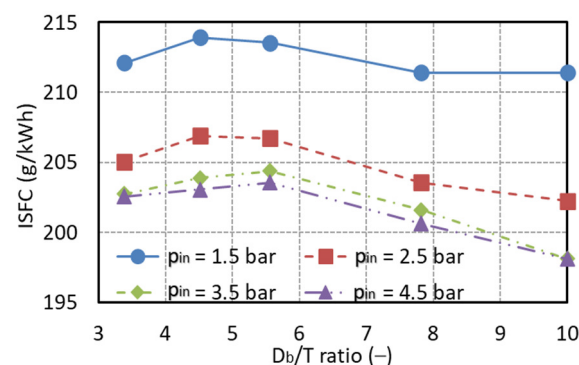


Figure 16. Dependence of ISFC on the change in p_{in} and D_b/T ratio.

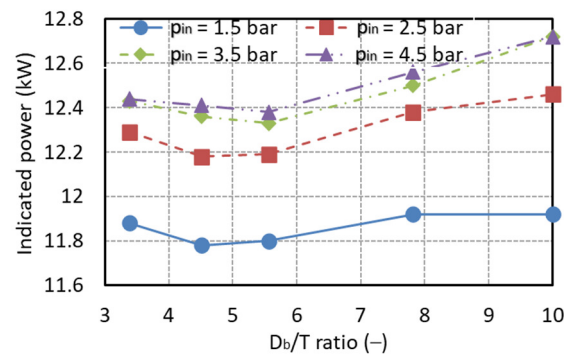


Figure 17. Dependence of the indicated power on the change in p_{in} and D_b/T ratio.

The ignition delay decreases (due to a rise in the ratio of the volumetric mixture formation process), while the value of the heat transfer into the wall of the combustion chamber grows with an increase in the D_b/T ratio. Additionally, the combustion process is faster and produces more effective work by the expansion process. These two factors determine the best values of the indicated power and the ISFC, which depends mainly on the D_b/T ratio (the presence of the optimal value of the D_b/T ratio).

When the boost pressure increases, the heat release process begins earlier (Figure 12), and more fuel gets burned in the combustion chamber volume. As a result, the optimal value of the D_b/T ratio increases, and the range of variation of the indicated power and ISFC becomes larger (Figures 16 and 17). At the same time, the nature of the changes in the indicated power and the ISFC for the boost pressures of 0.35 and 0.45 MPa is different from those values obtained at the boost pressures of 0.15 and 0.25 MPa.

3.4. Emission Characteristics

Variation of toxic substances and soot emission in exhaust gases with a change in the boost pressure and the D_b/T ratio is illustrated in Figure 18.

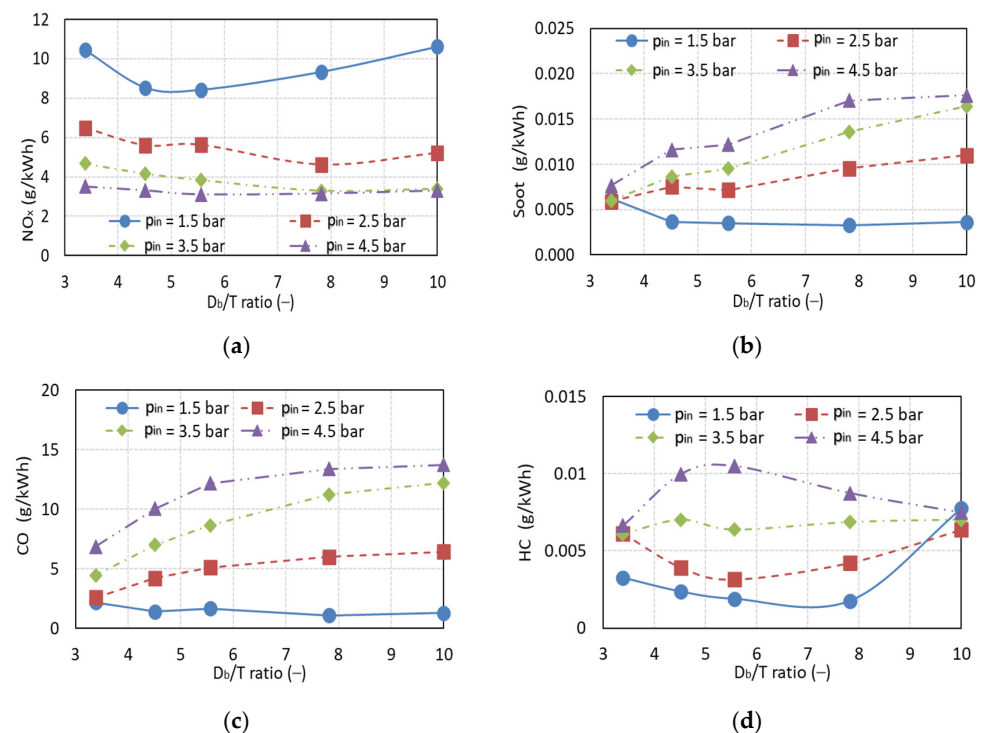


Figure 18. Variation of toxic substances and soot content in diesel engine exhaust gases with the boost pressure and D_b/T ratio change: (a) NO_x ; (b) soot; (c) CO; (d) HC.

The reasons for the decrease in temperature-related NO_x emissions include the growth in the boost pressure and the decrease in the time of ignition delay. The boost pressure value rises, which leads to air mass growth; therefore, α goes up, and as a result, $T_{\max}$ (Figure 13b) and NO_x (Figure 16) decrease. At the high boost pressure, the geometry of the combustion chamber has a slight effect on the production of NO_x , while at the low boost pressure, the effect of the combustion chamber geometrical parameters on the variation of NO_x is significant (at $p_{in} = 0.15$ MPa). The NO_x production depends on the D_b/T ratio and reaches the minimum value of NO_x at a D_b/T ratio of about 5.0. The NO_x value decreases when the ratio $D_b/T = (3.2 \div 4.5)$ and is almost unchanged with the ratio $D_b/T = (4.5 \div 5.6)$ and continues to decrease when the ratio $D_b/T = (7.8 \div 10)$. Variation of soot is opposite to the NO_x change. Except for $p_{in} = 0.15$ MPa, the NO_x value increases with the ratio $D_b/T = (7.8 \div 10)$, and soot and CO are almost unchanged when the ratio $D_b/T = (4.5 \div 10)$.

As the boost pressure goes up, more fuel concentrates in a smaller volume, and so the fuel density in the volume of the spray increases, which increases the resistance time of the spray. As a result, less air can be available to burn the injected fuel completely with a cleaner exhaust, with the cloud of burning fuel (Figure 15) smaller at the end of combustion, especially when $D_b = 90$ mm. This, in turn, leads to an increase in CO, HC, and soot (Figure 18).

4. Conclusions

The research was performed with a four-stroke diesel engine at a speed of 1400 rpm, boost pressure values from 0.15 to 0.45 MPa, and a constant CR injection pressure value of 300 MPa. The value of the injection fuel was kept constant at 60 mg per cycle, and the diameter of the combustion chamber increased from 60 to 100 mm on the condition that the compression ratio was unchanged. Based on the results received, we have the following conclusions:

- The boost pressure has a great influence on the fuel injection characteristics. As the boost pressure increases, the penetration and the spray tip velocity decrease, but the spray cone angle of the fuel spray increases.
- The increase in the boost pressure has a positive effect on the mixing rate due to an increase in the value of the pressure and the temperature at the end of the compression stroke. Therefore, at the value of $D_b/T = 10$, the boost pressure variation from 0.15 to 0.45 MPa leads to a decrease of 38% in $(dp/d\varphi)_{\max}$ and of 9% in $T_{\max}$.
- When the diameter values of the combustion chamber increase, the penetration and the value of the spray tip velocity also increase, but it slightly affects the spray cone angle. Additionally, the variation in the diameter of the combustion chamber has an influence on the fluctuation of the spray tip velocity and penetration.
- With an increase in the D_b/T ratio, the time of ignition delay decreases and the amount of heat transfer to the wall of the combustion chamber increases. Additionally, the combustion process is faster and produces more effective work by the expansion process.
- The maximum values of the indicated power and the ISFC depend mainly on the D_b/T ratio. The excess air ratio does not depend on the D_b/T ratio, but it has a weak effect on $p_{\max}$ at the ε value unchanged condition.
- At a high boost pressure, the geometry of the combustion chamber has a slight effect on the production of NO_x , while at a low boost pressure, the effect of the combustion chamber geometrical parameters on the variation of NO_x is significant (at $p_{in} = 0.15$ MPa).
- The NO_x production depends on the D_b/T ratio and reaches the minimum value of NO_x at a D_b/T ratio of about 5.0. The NO_x value decreases when the ratio $D_b/T = (3.2 \div 4.5)$ and is almost unchanged with the ratio $D_b/T = (4.5 \div 5.6)$; it continues to decrease when the ratio $D_b/T = (7.8 \div 10)$. Variation of soot is opposite to the NO_x change. Except for $p_{in} = 0.15$ MPa, the NO_x value increases with the ratio $D_b/T = (7.8 \div 10)$, and soot and CO are almost unchanged when the ratio $D_b/T = (4.5 \div 10)$.

- As the boost pressure goes up, carbon monoxide (CO), hydrocarbons (HC), and soot increase.

Author Contributions: Conceptualization, A.Y.D.; methodology, A.Y.D.; software, T.Q.N.; validation, T.Q.N. and A.Y.D.; formal analysis, A.Y.D.; investigation, T.Q.N.; resources, T.Q.N.; data curation, T.Q.N.; writing—original draft preparation, T.Q.N.; writing—review and editing, A.Y.D.; visualization, A.Y.D.; supervision, A.Y.D.; project administration, T.Q.N.; funding acquisition, T.Q.N. All authors have read and agreed to the published version of the manuscript.

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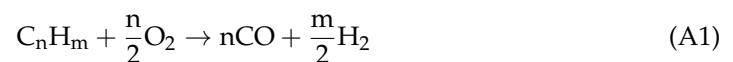
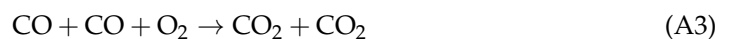
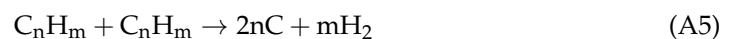
Data Availability Statement: The data presented in this study are available on request from the corresponding author. The data are not publicly available due to privacy.

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Conflicts of Interest: The authors declare no conflicts of interest.

Appendix A

The reaction mechanism model of soot formation is applied in the model:



Appendix B

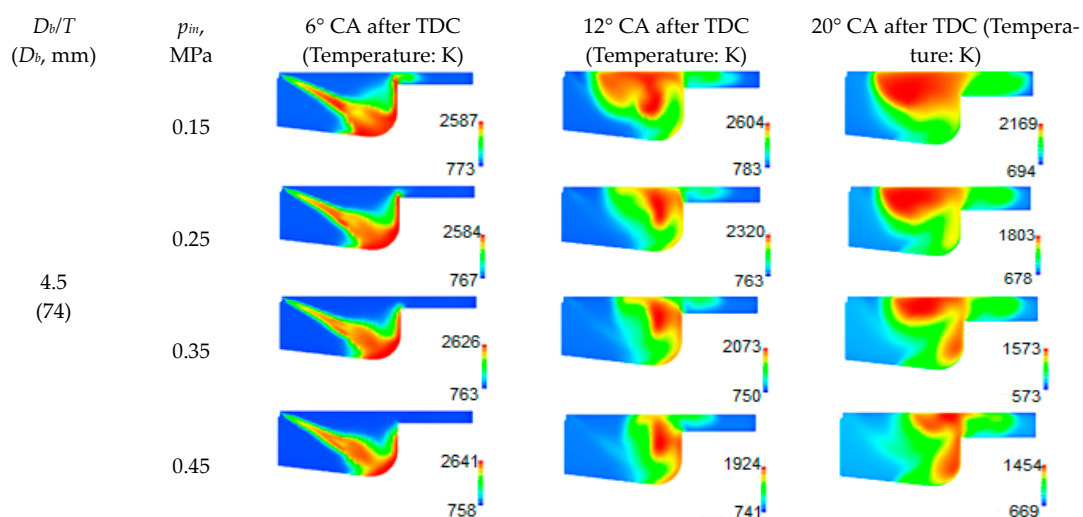


Figure A1. Cont.

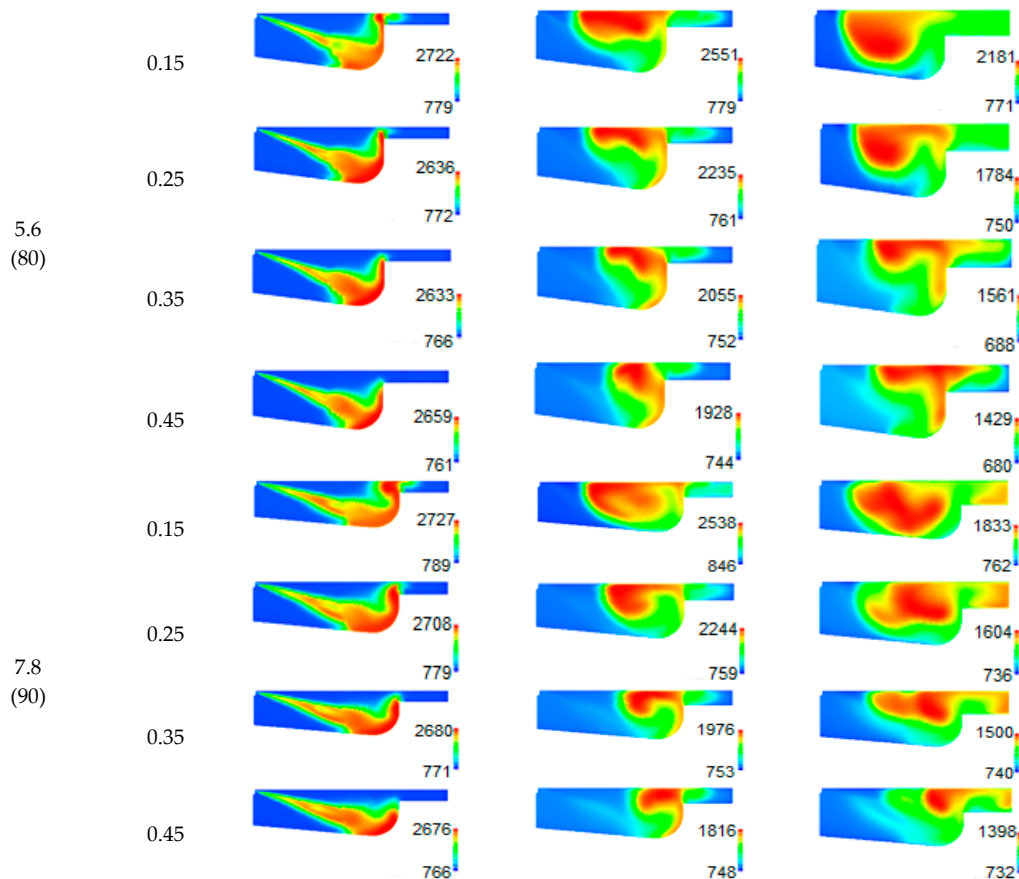


Figure A1. In-cylinder temperature distribution with $D_b/T = 4.5$, $D_b/T = 5.6$ and $D_b/T = 7.8$.

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