ABSTRACT: In this study, two different piston structures (bowl-shaped pit and trapezoidal pit) are constructed; the mixture formation, combustion, and emissions of a gasoline direct injection engine with the two piston types are compared and analyzed by computational fluid dynamics simulation. The results show that piston A (bowl-shaped) can form a combustible mixture near the spark plug at the ignition time and has higher pressure peaks and accumulated heat release than piston B (trapezoidal), which helps improve the combustion efficiency of the internal combustion engine. Furthermore, three pistons with different bowl-shaped pit depths (pistons A1, A2, and A3) are designed based on piston A. The results show that piston A2 (7.7 mm) has advantages in terms of strengthening the turbulence in the cylinders, promoting fuel evaporation, increasing the in-cylinder turbulent kinetic energy and the velocity of the airflow near the spark plug at the ignition time, and accelerating the rapid diffusion of combustion and the rapid increase in in-cylinder temperature and pressure. Also, piston A2 can reduce the CO and soot emissions.

1. INTRODUCTION

So far, the internal combustion engine is still the main power source of the automobile, even in the hybrid power system and range extender. It is also inseparable from the internal combustion engine. Therefore, the development of an efficient and clean internal combustion engine is necessary to achieve energy conservation and emission reduction.1−3 Much research shows that the direct-injection technology is the inevitable trend for the development of internal combustion engines.4,5

The gasoline direct injection (GDI) technology is different from the conventional port fuel injection (PFI)6 technology, which directly injects fuel into the cylinder and mixes it with intake air. Compared with PFI,7 GDI can avoid the fuel wet wall at the valve, achieve precise fuel supply, and overcome fuel injection delay. Consequently, the engine can be started within two working cycles, effectively reducing the emission of unburned hydrocarbons (HCs) during cold start. Moreover, GDI can not only achieve the stratification of the mixture but also provide conditions for lean burn and help improve the fuel economy of a gasoline engine. Last but not least, the latent heat of vaporization of the fuel that is injected into the cylinder can effectively reduce the temperature and volume of the mixture, leading to improved charging efficiency, reducing the tendency of deflagration, and creating conditions for increasing the compression ratio. This is beneficial to increasing the power of the engine.

Although the GDI engine has a broad prospect for application, some problems and challenges still need to be solved. First, the GDI engine has stricter requirements for fuel evaporation and mixture formation, which means increasing the atomization and evaporation rate of fuel.6 Second, the mixture formation time in the cylinder is short, resulting in the stratification of the mixture. Reasonable control of mixture stratification is also a key problem, which makes the mixture suitable for better spark and...
ignition. Finally, the stratification of the in-cylinder temperature distribution becomes extreme due to the non-uniform concentration distribution of the mixture, and there is a local higher-temperature region, which raises the risk of NOx emission. The thinner half of the stratified mixture’s outer edge, on the other hand, is prone to flame extinguishment. Simultaneously, the wet wall phenomena of fuel injection in the cylinder would worsen combustion where the mixture is overly rich on the top of the piston and the cylinder wall, resulting in comparatively large HC emissions at low load. Under low load, transition conditions, and cold start, locally concentrated mixtures or unevaporated liquid droplets in the cylinder contribute to incomplete particle oxidation, resulting in more particle emissions than a PFI gasoline engine. To remedy all of the concerns listed above, improved mixture preparation is required to achieve better mixture stratification, which is favorable to spark, ignition, and combustion, using a higher injection pressure system and better air motion. As a result, fuel injection parameter optimization and correct turbulence intensity rise become increasingly crucial.

In addition to reasonably organizing air movement through the design of the intake system, the design and optimization of the combustion chamber are also key methods for achieving better air movement. On the one hand, the piston structure affects the flow field distribution in the cylinder, which in turn affects the formation of the mixture. On the other hand, it has a guiding effect on fuel spray that plays an important role in combustion in the cylinder. Therefore, the optimization configuration of the top structure of the piston has the potential to improve the homogeneous combustion mode of the engine at high load and the stratified lean combustion mode at medium load and low load.

The piston shape has always been the focus of researchers around the world. Kurniawan et al. focused on the study of the effect of the piston shape on the fluid flow characteristics in a four-stroke GDI by computational fluid dynamics (CFD). They find that the shape of the piston crown does play a significant component in mixing parameters during the whole process of the engine. Two experimental techniques (laser Doppler anemometry and particle image velocimetry) were used to study the in-cylinder tumbling flow structure on a GDI engine by Krishnaiah et al., and they report that the flow structure largely depends on the piston bowl shape. Kang and Kim investigated the stratification of fuel vapor with different pistons in a GDI engine. Experimental results indicate that most of the fuel vapor is transported to the lower region for the flat piston (at 270°BTDC), the concentration of the mixture in the center region was the highest in the bowl piston, and the re-entrance piston was better for stratification due to a relatively smaller bowl diameter than the others (at 60°BTDC). All the research studies mentioned above show that the structure of the top of the piston is a key factor affecting the flow field condition and combustion process of the internal combustion engine.

For the GDI engine or direct injection spark ignition (DISI) engine, many researchers have studied the pitted piston on top and compared it with a flat-top piston. Okada et al. focused on the stratified charge combustion process of the GDI engine. The results indicate that the deep bowl is prone to produce unstable air inlets in the combustion process of homogeneous charge, which can inhibit the formation of uniform mixture and reduce power. The shallow bowl design can promote the formation of an excellent mixture in homogeneous charge combustion, achieving high output and stable combustion. Using CFD simulation, Harshavardhan and Mallikarjuna evaluated the sensitivity of the geometrical parameters to the variation of the in-cylinder flow and the air-fuel interaction in a DISI engine. So, they analyzed in-cylinder flow and air–fuel interaction created by different piston shapes (e.g., flat, flat-with-center-bowl, inclined, and inclined-with-center-bowl pistons) in a four-stroke DISI engine at an engine speed of 1500 rev/min. They find that the flat-with-center-bowl piston yields approximately 51% higher tumble ratio and about 21% higher turbulence kinetic energy (TKE) of in-cylinder flows, leading to better mixture stratification along with a 33% higher evaporation rate and a 33% higher percentage of fuel evaporation than flat pistons. Qian et al. performed numerical analysis on a GDI engine, proposing that the piston pit with a shallow depth adjacent to the center of the cylinder could intensify the scavenging motion and increase the velocity inside the pit, and more residual gas and carbon particles could therefore be cleared.

Meanwhile, researchers have found that improving the bowl-shaped piston can optimize combustion and emission, especially the tumble ratio and TKE. Krishna and Mallikarjuna investigated the in-cylinder tumble flows in a single-cylinder GDI engine with five different piston crown shapes at an engine speed of 1000 rev/min during intake and compression strokes under motoring conditions using particle image velocimetry. They found that the pentroof-offset-bowl piston showed approximately 41 and 103% improvement in tumble ratio and mean TKE, respectively, compared to those of a flat piston at the end of compression. Wang et al. performed numerical analysis on the GDI engine to study the effect of piston bowl shapes and direct injection strategies on the in-cylinder conditions. The results show that the shallow bowl piston shape could improve the local velocities and TKE around the spark plugs compared with the deep piston bowl, and the evaporation ratios of the bowl piston are comparable with those of the original flat piston. Yin et al. investigated the combined effects of piston bowl geometry and a charge motion control valve on the tumble flow and combustion features in a GDI engine. The results indicate that the bowl on the top of the piston is beneficial for the formation and development of tumble flow. However, with the increase in engine speed and load, the dual offset bowl piston has a low tumble ratio and TKE at the end of the compression stroke because of the projection in the middle of the piston’s top surface, which leads to a low pressure increase rate and a reduced flame propagation speed at high load. Wang et al. used CFD to analyze the effect of the stratified flame formed by different piston shapes on the stoichiometric stratified flame ignition hybrid combustion and heat release process in PFI/GDI engines. They found that the original stratified mixture pattern was formed by the newly designed bowl piston and larger diameter bowl piston, which were characterized by the central rich mixture around the spark plug and the stratified lean mixture in the peripheral region. This leads to an effective decrease in the maximum pressure rise rate with a slight deterioration of the indicated mean effective pressure. Shafei and Said observed the effects of different piston bowl structures on the in-cylinder flow characteristics of a GDI engine. They found that the geometry of the piston bowl has almost no effect on the in-cylinder flow during the intake stroke, and the piston bowl structure significantly affects the flow, especially near the top dead center (TDC) during the compression stroke. In addition, the swirl ratio was increased by 34.8%, and the tumble ratio was increased by 7% compared to the original piston bowl shape at
the end of the compression stroke. Zheng et al.\textsuperscript{8} performed a numerical analysis on the GDI engine using four piston shapes. The results show that the flat top surface of the piston is more conducive to the formation of a combustible mixture for stable ignition near the spark plug at the moment of ignition. A smoother piston top surface strengthens the TKE on the spark timing, thereby accelerating the combustion process. Adhishan et al.\textsuperscript{19} investigated the influence of squish on the piston bowl in a single-cylinder compressed ignition engine fueled with jojoba biodiesel and its effect on performance, emission, and combustion characteristics. They compared the combustion chamber geometry of the optimized octagonal bowl combustion chamber with the standard bowl piston and found that a significant increase in brake thermal efficiency was noticed for the octagonal piston bowl. There is a reduction in HC and CO, which is seen while using an octagonal bowl piston compared with the standard piston.

Other studies on the shape of the piston are also being recorded. A joint numerical and experimental investigation was conducted by Xu et al.\textsuperscript{20} in a heavy-duty compression ignition engine, investigating the effects of injection timing, piston geometry, and compression ratio on the fuel/air mixing and combustion, covering different regimes of low-temperature combustion engines. The result shows that in the transition regime, the stepped-lip piston is favorable toward the balance of the fuel fraction inside and outside the piston bowl and promotes the fuel–air mixing process in the squish region, which simultaneously improves the combustion in the squish region and reducing the maximum pressure rise rate and the emissions, especially for NO\textsubscript{x}.

From the research studies mentioned above, it can be seen that only the proper setting of the shape of the combustion chamber, especially the structure of the piston top with pits, is beneficial to the formation of the mixture. So, improving the shape of the piston can enhance the strength of the vortex and turbulent flow in the cylinder. However, the previous research only studied the shape of the same type of pit and did not study the possibility of possible problems. It is also possible to conduct a detailed study on the design of the piston shape, such as the depth of the piston pit. Moreover, the design of each piston shape is not universal and cannot be applied to all types of engines.

Based on the previous studies, this study designed two pistons with a large difference in pit shape such as bowl-shaped pit and trapezoidal pit, and the pits are deepened in three different depths. CFD simulation has strong advantages\textsuperscript{21} for the incipient combustion chamber design because it can not only obtain the detailed information of various fields (such as flow field, temperature field, pressure field, etc.) completely and predict the performance of the internal combustion engine comprehensively but also reduce the research cost, shorten the development cycle, and save a lot of manpower and material resource. Therefore, CFD simulation was used in this study to analyze the effect of piston crown geometry on the in-cylinder working process (including flow, spray, mixture formation, pollutant emissions, etc.) of GDI so as to further optimize the piston shape of the GDI combustion chamber.

2. METHODOLOGY

2.1. Geometric Model and Calculation Model of the Direct Injection Gasoline Engine. 2.1.1. Geometric Model. The basic parameters and setting of the crank angle of the GDI engine are shown in Tables 1 and 2, respectively, and the valve and piston lift are shown in Figure 1. Meanwhile, the geometric model and meshing of the engine are shown in Figure 2.

| Table 1. Basic Parameters and Setting of the GDI Engine |
|-----------------------------------------------|
| parameter | number |
| number of cylinders | 4 |
| cylinder bore (mm) | 86 |
| piston stroke (mm) | 86 |
| compression ratio | 10.3 |
| link length (mm) | 142.8 |
| single-cylinder displacement (L) | 0.5 |

| Table 2. Setting of Crank Angle |
|--------------------------------|
| location | angle (°CA) |
| ignition timing | 700 |
| intake TDC | 360 |
| compression TDC | 720 |
| intake valve open timing | 368 |
| intake valve close timing | 591 |
| exhaust valve open timing | 172 |
| exhaust valve close timing | 388 |

2.1.2. Selection of the Numerical Model Submodel. In the numerical simulation of the engine, the used submodels are shown in Table 3. In terms of internal combustion engines, turbulence calculations often use the k-\epsilon double equation to close the Reynolds stress term in the momentum equation. k and \epsilon represent the turbulent kinetic energy and the turbulent energy dissipation rate, respectively. The k-zeta-f model selected in this study is developed on the basis of k-\epsilon, which has better calculation accuracy and stability, but the calculation time is slightly longer.\textsuperscript{22}

The spray model in this article starts from the theory that the fuel spray has a gas–liquid two-phase structure and focuses on simulating the interaction that occurs at the gas–liquid interface and the mass, momentum, and energy exchange process between the two phases. Considering the simplicity of the calculation process, the spray model in this study is based on the DDM\textsuperscript{23,24} method, which includes submodels such as fragmentation, evaporation,\textsuperscript{25} collision aggregation, wall film model,\textsuperscript{26} and wall interaction.\textsuperscript{27}

The combustion model used in this paper is an extended coherent flame model (ECFM)\textsuperscript{28} based on the coherent flame model. This model uses a two-step chemical reaction mechanism and considers the formation of CO and H\textsubscript{2} in the stoichiometric ratio and fuel-rich state. At the same time, the model assumes that there is no fuel in the burned gas under high temperature conditions, but chemical reactions can occur. In addition, it is coupled with a spray module, which is more suitable for describing the combustion phenomenon of direct injection internal combustion engines. This article also includes NO\textsubscript{x},\textsuperscript{29,30} and soot emission models.\textsuperscript{31} The selection of submodels is shown in Table 3 below.

2.1.3. Initial and Boundary Conditions. In this study, the piston and valve are defined as moving wall surfaces, and the others are considered fixed wall surfaces. Temperature boundaries are used, and their values are determined empirically. The boundary conditions of the inlet and exhaust are taken from one-dimensional (1D) simulation results and are set as the mass flow inlet and pressure outlet, respectively. The initial conditions used in the calculation are also assigned to the inlet,
exhaust, and inner areas of the cylinder based on the 1D simulation results. Table 4 contains the specific definitions. The coordinate position and spray direction of the nozzle are measured in the computer-aided design model. Moreover, spray parameters such as the jet rate, half-cone angle, initial droplet velocity, and size distribution are defined according to the test results.

2.2. Grid and Numerical Simulation Model Verification. 2.2.1. Grid Establishment and Verification. In this study, grid sizes of 4, 2, and 1 mm are selected for grid independence verification. Figure 3 is a comparison chart of the mean pressure and mean temperature in the cylinder under the three grid sizes in the cold flow state. Two millimeters is used as the grid size calculated in this study. For one thing, it can be seen from the figure that the pressure and temperature of the 2 mm grid size are the same as those of the 1 mm grid size. For another, the larger grid needs to be selected for calculation in order to improve the calculation accuracy and save the calculation time.

2.2.2. Model Validation. After the fuel enters the combustion chamber, a series of physical changes occur, such as crushing turbulence disturbance, deformation, collision polymerization, and wall collision. The constant volume projectile was used to calibrate the spray model, and the calculated conditions were consistent with the spray test conditions in the literature. The simulation conditions are shown in Table 5. The experimental conditions are shown in Table 6. Figure 4 shows the verification of spray and combustion models, and the experimental data comes from previous research. The simulation and test in Figure 4a showed similar liquid fuel spray structures and penetrations, confirming the validity of the free spray model. The correctness of the ECFM combustion model is verified in Figure 4b. It can be seen that the numerical model of this study can simulate the real working conditions very well.

3. RESULT AND DISCUSSION

3.1. Design of Piston Types. In this study, ProE software was used to build two different piston types as shown in Figure 5, namely, A-type and B-type (hereinafter referred to as piston A and piston B, respectively). The combustion chamber of the piston crown with pits has been proven to be beneficial to mixture formation. So, piston A and piston B designed in this study also have pits. The difference is that piston A has a bowl-shaped pit in the center of the piston crown to control the development of fuel mist, and piston B is designed with a trapezoidal recess near the intake side. In the following paragraphs, the mixture formation, combustion, and emissions of the GDI engine with the two piston types were compared and analyzed.

The calculation conditions include the engine compression ratio at 10.3. The fuel injection amount per cycle is 21.692 mg, injection is divided into two times, and the fuel injection ratio is 25:75. The fuel injection times are 440 and 635 °CA, and the ignition time is 700 °CA.

3.1.1. Mixture Preparation. Figure 6 shows the changes in the mass flow rate of the cylinder intake air and the total intake air volume caused by the two pistons. It can be seen that there is a little difference in the mass flow rate of the intake air, which is caused by the different atomization effects of the fuel after the
first injection. The total intake volume of the two is almost the same. Therefore, the difference of air intake caused by piston shape can be ignored, and the overall air—fuel ratio of the two combustion chambers can be considered consistent.

Figure 7 shows the fuel evaporation mass and fuel film mass with crank angle. It can be seen that different airflow movements lead to varying fuel evaporation in the cylinder (Figure 7a). Before the second injection, the fuel evaporation amount of piston B is greater than that of piston A. The larger the fuel evaporation amount, the correspondingly less oil film is formed, which is consistent with the rule in Figure 7b. However, although piston B’s fuel evaporation was larger, its fuel film quality increased significantly over piston A after the second

Table 5. Simulation Condition

| parameter                | value  |
|--------------------------|--------|
| ambient temperature (°C) | 25     |
| ambient pressure (kPa)   | 140    |
| fuel temperature (°C)    | 80     |
| injection quality (mg)   | 76.38  |
| injection pressure (MPa) | 15     |
| injection duration (ms)  | 4.25   |
| initial velocity (m/s)   | 152    |
| initial particle size (mm)| 0.1    |
| computing time (ms)      | 6      |
| time step (ms)           | 0.05   |
| maximum iteration step length | 100    |

Table 6. Engine Experimental Parameters

| parameter                | value  |
|--------------------------|--------|
| rotating speed (r/min)   | 2000   |
| torque (N m)             | 88.9   |
| load                     | 35%    |
| equivalence air—fuel ratio | 1.0    |
| injection start time (°CA)| 57.9   |
| end of injection (°CA)   | 83.4   |
| injection quantity (mg)  | 21.10  |
| ignition timing (°CA)    | 341    |
| intake pressure (bar)    | 1.138  |
| intake temperature (K)   | 293    |
injection. As mentioned previously, the quality of the fuel film should be reduced as much as possible because it is harmful to the combustion and emissions from the cylinder of the engine. From this picture, it can be seen that when it is about to reach the top dead center at 700° CA, the fuel film mass of piston A only has 1.3 mg, which is much lower than 1.7 mg of piston B. So, in this regard, piston A has a remarkable effect, while piston B is worse.

Figure 8 shows the comparison of fuel film thicknesses at the spark time. It can be seen that the piston pit is the main area where the fuel film exists, and the fuel film in the piston pit is mainly caused by the collision of the fuel beam with the piston. The fuel deposited on the top of the piston will adsorb the next injected fuel and form a vicious cycle, thereby increasing HC emissions. In the piston pit, the thickness of the wall fuel film produced by piston B is thick, whereas that of piston A is thin. Figure 9 shows the change in the fuel–air equivalence ratio at the spark plug after the second injection and before ignition. At the moment of ignition, the spark plugs of both combustion chambers can achieve ignition. The fuel–air equivalence ratio of piston A lies between 1.29 and 1.3, whereas that of piston B is only between 0.75 and 0.76. The ratios are not in the range of 1.1−1.25, which is not suitable for stable ignition.

Figure 10 presents the distribution of equivalence ratios at the spark time. According to the distribution of the mixture concentration in the cylinder at the time of ignition, the rich
mixture of piston B is biased toward the intake side, and that of piston A is roughly in the middle of the cylinder, which is determined by the position and shape of the piston pit of piston A. Piston A has the best effect in terms of the uniformity of the mixture distribution in the cylinder.

The formation and propagation of the initial flame are not only related to the local concentration distribution but also largely affected by the current local air movement. Under the premise of ensuring that the flame is not blown out, the greater the air intensity is, the more conducive the flame is to the transfer of air masses and the more conducive to the diffusion of the combustion flame. The mean TKE in the cylinder reflects the intensity of airflow movement in the cylinder. Figure 11 gives the curves of in-cylinder mean TKE evolution. From the whole process, the mean TKE of piston A is greater than that of piston B after the second injection and ignition, and the mean TKE of the two conditions thereafter is approximately the same. The entrained airflow during high-pressure fuel injection has an important contribution to the in-cylinder TKE and has a positive effect on the formation of the mixture and combustion process.

The airflow movement streamline in Figure 12 shows that a large part of air enters the cylinder and reaches the top of the piston via the exhaust side. Under the action of the high back of the pit, the air “stripping” on the piston tops and wall surfaces of piston A and piston B develops obliquely upward. Thus, the tumble center is biased to the exhaust side. Figure 11 shows that during the intake stroke, the areas with high TKE are often found on the locations where the airflow collided and the airflow hit the wall. With the loss of TKE, the TKE in the cylinder with piston A is greater than that with piston B before and after the ignition at the end of compression. The airflow direction also changes from the intake side to the exhaust side through the spark plug (Figure 12). Therefore, in the preparation of the mixture, regardless of the concentration or velocity distribution, the overall effect of piston A is better than that of piston B.

3.1.2. Combustion Performance. Different mixture preparations determine various combustion rates. Figure 13 shows the
comparison of flame contour evolution. In terms of the intensity of the airflow movement at the spark plug, piston B’s flame spreads faster than that of piston A. At 730 °CA, piston A’s flame has a large unburned area on the intake side, and the rest has basically been burned. Piston B still has a small unburned area on the inlet and exhaust side edges of the combustion chamber. Also, Figure 13 shows that the main development trend after flame formation is from the intake side to the exhaust side, which is determined by the airflow direction in the cylinder. At the end of the compression stroke, the large-scale positive tumble flow was deformed and broken because of the compression of the piston and the cylinder head. Then, an air flow movement from the intake side blows toward the exhaust side near the spark plug.

Figure 14 shows the comparison of in-cylinder mean pressure and temperature profiles with the two combustion chambers. It can be seen that from the moment of ignition (700 °CA), the mean pressures of piston A and piston B both increase first and then decrease, and piston A reaches its peak earlier than piston B. After TDC (720 °CA), the mean pressure in the cylinder of piston A is 3.2 MPa, which is significantly greater than that of piston B of 2.7 MPa. Since the increase in cylinder pressure is beneficial to the improvement of engine power performance, from the perspective of improving engine power, piston A is better than piston B. Furthermore, the optimal position of the cylinder pressure peak value of the direct injection gasoline engine to minimize the negative compression work and improve the engine efficiency is 10−15 °CA after TDC (730–735 °CA). Piston B is slightly outside this range. Figure 14b shows that the mean temperature in the cylinder of piston A is lower than that of piston B, the temperature rise rate of the two pistons is slower before reaching TDC, and the mixture pressure and temperature in the cylinder rise rapidly after TDC. So, the temperature rise rate is faster.

Figure 15 shows the in-cylinder temperature distributions. It can be seen that the mixture temperature rapidly increased after the flame front sweeps. The spatial distribution and time duration of piston B in the high-temperature region at 720 °CA are greater than those of piston A because the uniformity of the mixture of piston B is better than that of piston A. Figure 16 shows that the changing trends of piston A and piston B are almost the same, and the heat release of piston A and piston B is very concentrated. After the TDC, the piston descends and the heat dissipation area increases. At this time, the efficiency of converting the heat generated by the combustion of the mixture into work decreases, which will affect the power and economy of the engine, requiring the engine to quickly release heat. It can be seen from the figure that the maximum heat release rate of piston B is 35 J/°CA, but that of piston A is only 30 J/°CA. Therefore, the effect of piston B is slightly better than that of piston A. It can be seen from Figure 16b that the cumulative heat release with piston A is 720 J, which is higher
than that with piston B of 680 J. This is because piston B produces a substantial oil film after the second injection and the flame propagation condition of the combustion chamber is poor after the occurrence of ignition. Consequently, the fuel forming a fuel film cannot participate in combustion completely, which leads to the finding that cumulative heat release with piston A is higher than that with piston B. Due to the higher fuel utilization rate of piston A, the effect of piston A is better than that of piston B.

3.2. Comparison of the Shape of the Piston Top Surface. From the mixture formation, combustion performance, and emission performance, the comprehensive performance of piston A is better than that of piston B. Also, on this basis, this study made a specific design for the pit depth of piston A as follows.

Pit depth is defined as the height difference between the deepest point of the pit and the highest point of the piston top surface, and pit width is defined as the maximum width difference between the pits perpendicular to the valve centerline. According to General Motors Corporation, a GDI engine with a piston dimple has considerably better combustion effects than the one with flat-topped pistons. Therefore, the structure of piston A is changed on the basis of the four valve dimples and one piston dimple in this study. In this paper, the engine compression ratio is fixed, the piston pit depth is changed correspondingly, and then the influence of piston pit depth on the engine working process is studied and an optimization scheme is found. The pit depths are 3.7, 7.7, and 6.7 mm, and the corresponding pistons are named piston A1, piston A2, and piston A3 (Figure 17).

3.2.1. Comparison of the Combustion Performance of Pistons with Different Pit Depths. The difference in in-cylinder fuel evaporation is shown in Figure 18a. Before ignition, the pistons that correspond to the fuel evaporation amount in descending order are piston A3, piston A2, and piston A1. The larger the fuel evaporation amount is, the less the oil film is formed, which is consistent with the rule in Figure 18b. The quality of the oil film should be reduced as much as possible because it is harmful to the combustion and emissions in the engine cylinder. In this regard, piston A3 has the best effect in the three pistons, followed by piston A2 and piston A1.

Figure 19 shows the distribution of equivalence ratios at the spark time. From the perspective of the mixture concentration distribution in the cylinder at the ignition time, the rich mixtures of piston A2 and piston A3 are biased toward the intake side,
whereas the rich mixture of piston A1 is roughly in the middle of the cylinder. The latter is decided by the position and shape of the pit of piston A1. Because the deeper pits make the fuel concentrated in the pits and not easy to fall off, the fuel–air equivalence ratios of piston A2 and piston A3 are wider. From the perspective of the uniformity of the mixture distribution in the cylinder, the effect of piston A2 is the best. In the picture, the equivalence of piston A2 and piston A3 at $y = 2$ mm slice at the piston pit is determined by the slice position.

The airflow movement streamline in Figure 20 shows that a large part of air enters the cylinder and reaches the piston top through the exhaust side. Under the action of the high back of the pit, the air “stripping” on the piston top and the wall surfaces of piston A2 and piston A3 develops obliquely upward, thereby making the center tumble to the exhaust side. With the loss of TKE, the order of the TKE in the cylinder from intense to weak before and after ignition at the end of compression changes into piston A3, piston A2, and piston A1. The airflow direction through the spark plug also changes from the intake side to the exhaust side.

Figure 21a shows the airflow velocity at the spark plug gap at the ignition time. As shown in a previous study, a stable ignition requires the airflow velocity of the spark plug gap to be less than 15 m/s. The figure illustrates that three combustion chambers meet the spark plug gap airflow velocity of less than 15

![Figure 19. Distributions of equivalence ratio at the spark time.](image19.png)

![Figure 20. In-cylinder TKE distributions (with streamlines) ($x = 0$ mm).](image20.png)

![Figure 21. Air velocity near the spark plug (a) and fuel–air equivalence ratio (b) at the spark time.](image21.png)
The speed of the airflow of piston A2 and piston A3 is higher than that of piston A1, and the combustion promotion effect is better. This result is consistent with the distribution law of the average TKE in the cylinder at the ignition time.

Figure 21b shows the fuel–air equivalence ratio at spark plugs at the ignition time. The spark plugs of the three combustion chambers can achieve ignition. However, the fuel–air equivalence ratio of piston A2 is higher than 1.4, whereas those of piston A1 and piston A3 are between 1.29 and 1.3, thereby exceeding the range of 1.1 to 1.25.

Therefore, in the preparation stage of the mixture, the overall effect of piston A2 is the best regardless of the concentration or velocity distribution. The velocity distribution of piston A2 is equivalent to that of piston A3, but the concentration distribution and air velocity at the spark plug of piston A2 are slightly lower than those of piston A3. Piston A1 is the worst among the three pistons.

The intensity of the airflow movement at the spark plug implies that the flame diffusion speed of piston A2 and piston A3 is faster than that of piston A1 (Figure 22). At 730 °CA, the piston A2 and piston A3 flames have basically burned through the entire combustion chamber. Piston A1 still has a larger unburned area on the intake side edge of the combustion chamber compared with piston A2 and piston A3. Piston A1 still has a larger unburned area on the intake side edge of the combustion chamber compared with piston A2 and piston A3. The main development trend after the flame formation is to pass from the intake side to the exhaust side. At the end of the compression stroke, an airflow movement from the intake side to the exhaust side is formed near the spark plug. Although the mixture formed by piston A2 and piston A3 on the intake side is suitable for the rapid propagation of the flame, it is not the mixture concentration that dominates the propagation direction of the flame after the flame formation but the local airflow movement.

Figure 23 shows the in-cylinder mean pressure. It can be seen that piston A1 has the largest pressure peak, followed by piston A2 and piston A3. Since the increase in in-cylinder pressure is beneficial to the improvement of engine performance, piston A1 and piston A2 are better than piston A3. Figure 23b shows that the in-cylinder mean temperature of piston A2 is the highest, and the other two are relatively lower. This is because the shallow pits are not conducive to uniform mixing of the mixture. The flame spread rate is small, the temperature in the cylinder rises slowly, and the peak is low when the lean mixture is not conducive to flame spread. Therefore, in terms of temperature, piston A2 is better than piston A1 and piston A3.

The change in the temperature distribution in the cylinders (Figure 24) demonstrates that the high-temperature region of piston A3 at 720 °CA is larger in terms of space distribution and duration than those of piston A1 and piston A2, but such a disadvantage increases NOx emissions.

3.2.2. Comparison of the Emission Performance of Pistons with Different Pit Depths. Figure 25 shows in-cylinder NO distributions and fractions. It can be seen that the high-temperature areas in the three combustion chambers are close to one another. As a result, the locations with high NO concentrations in the three combustion chambers are also close and are basically on the exhaust side (Figure 25a). In the combustion chamber of piston A2, the amount of NO produced is large due to the fact that the cylinder temperature is high during the main combustion, and the peak time appears earlier than that of piston A1 and slightly later than that of piston A3. Figure 25b shows the in-cylinder NO mass fractions at exhaust valve open. Figure 25b shows that the NO emissions of the three pistons are different, with the mass fraction of piston A1 being $9 \times 10^{-6}$. After that, it is $1.6 \times 10^{-5}$ for piston A3 and $2.1 \times 10^{-5}$ for piston A2.

Figure 26 shows in-cylinder soot mass distributions and fractions. From Figure 26a, it can be seen that soot is mainly concentrated on the intake side and limited on the exhaust side. The time sequence of soot peaks shows the same trend as that of NO, that is, piston A2 and piston A3 are earlier, followed by piston A1. Figure 26b shows that the mass of soot produced by piston A1 is greater than that of piston A2 as the high temperature of the combustion chamber is conducive to the later
oxidation of soot. Although piston A3 has the highest temperature and the poorest fuel film quality compared to pistons A1 and A2, its mixture is richer and the concentration gradient is larger due to the relatively concentrated fuel.

Consequently, the influence of concentration is greater than that of a high temperature, resulting in a remarkably increased amount of soot generated and a higher amount of soot produced by piston A3 than piston A1. Also, the soot mass fraction of piston 2 is $4 \times 10^{-9}$, which is the lowest among the three pistons.

Figure 25 shows in-cylinder CO mass distributions and fractions. From Figure 27a, it can be seen that peaks of CO concentration in the three combustion chambers appear in the
order of piston A2, piston A3, and piston A1. The CO concentration depends on the mixture concentration. Piston A1, piston A2, and piston A3 have a large amount of fuel that does not leave the piston pit but accumulates at the pit edge. The concentration of the mixture is consequently high. Thus, the CO concentration at the edge of the piston pit is higher than that at the center of the pit. The order of the mass fraction of CO in the cylinders is 0.03 for piston A1, 0.021 for piston A3, and 0.015 for piston A2. The reason is that the oxidation of CO is inversely related to the concentration distribution (Figure 27b).

Overall, piston A2 is the most advantageous in reducing the original emissions of soot and CO because of its complete combustion process, but NO\textsubscript{x} emissions are correspondingly high. However, for the GDI combustion near the equivalence ratio, the problem of high NO\textsubscript{x} emissions can be solved via a three-way catalytic converter.

4. CONCLUSIONS

Two different piston types were constructed in this study, and the mixture formation, combustion, and emissions of the GDI engine with the two piston types were compared and optimized. The specific conclusions are as follows:

1) Piston A (bowl) has a faster mixture formation speed and combustion diffusion speed than piston B (trapezoid). On the one hand, the mass of the fuel film of piston A is only 1.3 mg after ignition, while the mass of the fuel film of piston B is 1.5 mg. On the other hand, the equivalence ratio of piston A is 1.29–1.3, and the equivalence ratio of piston B is 0.75–0.76. Also, considering the combustion in the cylinder, the peak pressure and cumulative heat release of piston A reached 3.3 MPa and 720 J, respectively, while those of piston B were only 2.7 MPa and 680 J. As a result, piston A can effectively improve the combustion efficiency of the internal combustion engine.

2) Compared to shallow piston pits, piston A2 (7.7 mm pits) is beneficial to the strengthening of the turbulence in the cylinders, promoting fuel evaporation, reducing the mass of the piston oil film, and increasing the in-cylinder TKE and the velocity of the airflow near the spark plug at the ignition time. Also, the proper design of deepening the pit depth (piston A2) is beneficial to the rapid diffusion of combustion and the rapid increase in the in-cylinder temperature and pressure. The peak temperature and pressure are also high. In terms of emissions, the mass fractions of CO and soot emitted by piston A2 are only 0.015 and $4 \times 10^{-9}$, respectively, which are the lowest of the three piston types. It shows that a suitable deep pit piston top helps disperse CO and promote the secondary oxidation of CO, thereby reducing CO emissions.

3) All the three bowl-shaped piston shape combustion chambers based on the design of piston A can form a better combustible mixture near the spark plug at the ignition time, but piston A2 has the best effect and highest stability.

5. THE DEFICIENCIES OF THE RESEARCH IN THIS ARTICLE AND FUTURE PROSPECTS

1) This paper does not consider the influence of the fuel injection strategy on combustion and emissions in the
cylinder and does not consider the influence of different fuel injection strategies on different piston shapes. The follow-up will combine the influence of the fuel injection strategy and then select the best fuel injection strategy and piston shape.

(2) In addition to the fuel injection strategy and the shape of the piston, the intake port has a great influence on the formation of the mixture. Due to time constraints, this article has not launched a study on the influence of different intake port types on the formation of the mixture. The follow-up work should be considered comprehensively: matching and optimization of multiple parameters such as intake port, fuel injection strategy, combustion chamber shape, and ignition timing.

Author Contributions

X.D. and Z.Z. contributed equally to this work. The manuscript was written through contributions of all authors. All authors have given approval to the final version of the manuscript.

Notes

The authors declare no competing financial interest.

ACKNOWLEDGMENTS

This research is supported by the National Natural Science Foundation of China (grant no. 51776024) and Special Key Project of Chongqing Technology Innovation and Application Development, China (grant no. cstc2020jscx-dxwtBX0024).

ABBREVIATIONS

GDI, gasoline direction injection; CFD, computational fluid dynamics; TKE, turbulent kinetic energy; PFI, port fuel injection; HC, unburned hydrocarbon; BDTC, before dead top center; DISI, direct injection spark ignition; ECFM, extended coherent flame model; CFM, coherent flame model; 1D, one-dimensional; SOI, spark of ignition; EVO, exhaust valve open

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