Research Article

Regular Analysis of Aero-Diesel Piston Engine between Combustion Chamber Size and Emission

Zhongjian Pan¹, Yu Deng² and Lizhi Cheng¹

¹School of Mechanical and Electrical Engineering of Changsha University, Changsha 410022, China

²Sunward Intelligent Equipment Co. Ltd, Changsha 410100, China

Correspondence should be addressed to Zhongjian Pan; pzjian1005@163.com

Received 9 January 2019; Revised 16 February 2019; Accepted 21 February 2019; Published 5 May 2019

Guest Editor: Abdallah Bouabidi

Copyright © 2019 Zhongjian Pan et al. This is an open access article distributed under the Creative Commons Attribution License, which permits unrestricted use, distribution, and reproduction in any medium, provided the original work is properly cited.

The emission of aero-engines has been a focused issue, studying the regular of combustion chamber size on engine emission performance, with an aviation diesel piston engine as the object of study; the numerical model of diesel combustion spray and emission model are analyzed; and the dynamics grid of the combustion chamber is meshed by FIRE software, analyzing the relationship between the reentrant diameter, the maximum depth of the combustion chamber, and the emission generation, comparing the NO_x and soot emissions under different combustion chamber sizes. The results show that reducing appropriately the reentrant-max diameter ratio and max diameter-max depth ratio of the combustion chamber can reduce emissions when maintaining the same compression ratio by adjusting the mid-depth. Modifying the geometry parameters of the combustion chamber to verify regularity, it was found that engine NO_x emission decreased by 28% and soot emission decreased by 3.6% when changing the size, which verified the correctness of regular analysis.

1. Introduction

Piston engine is the main power unit for ground transportation as well as general aviation equipment [1]. The emission problem of diesel engine has obtained more and more attention for its better fuel economy [2], especially at the time that the general aviation market is open to the whole world. General aviation aircraft have grown quite rapidly in recent years; the number of general aircraft in the United States has exceeded 300,000 by 2017. Some aero-gasoline engines have been replaced by aviation diesel engines, for example, the EPS engine and WMA-100 engine; however, pollution problems still exist. Currently, much research has been done on the emission of engine, and many emission control methods have been raised. However, most of them are based on engine reprocessing and injection control strategy [3–6]. Chmielewski and Gieras [7] have adopted a variable turbo charging and exhaust gas circulation to improve emission control, and Bogarra et al. [8] have designed an electronic EGR system to ensure the EGR rate under different working conditions to reduce the emission of NO_x . Besides, there are also some

scholars [9–11] who have done a lot of research on emission from oxidation sensors and other research methods. These emission-related studies are mainly for automotive engines; some scholars have also made research on the emissions of aero-engines. For example, Liu et al. [12] provided a comprehensive review of low-emission combustion technologies for modern aero-gas turbines. Besides, Innocenti et al. [13] presented a numerical study where a hybrid CFD-chemical reactor network (CRN) approach is used to predict pollutant emissions in a tubular combustor for aero-gas engine applications. And in Xisto et al. [14], the synergistic combination of intercooling with pulsed detonation combustion is analyzed concerning its contribution to NO_x and CO_2 emissions. There is very little research on emissions of aviation heavy oil piston engine. So far, the manufacturers who have obtained type certifications are Thielert in Germany, Continental in United States, Austro in Germany, SMA in France [15], the developing DeltaHawk in the U.S., and Wilksch in Britain [16–19]. Emission control indicators have been formed on aviation turbine engines; Federal Aviation Administration (FAA) and European Aviation Safety Agency (EASA) have

not made requirements for emissions when collecting airworthiness certificates of aviation piston engines. However, with the continuous advancement of environmental protection policies, the emission problem of aviation diesel piston engine will be brought to the agenda as soon as possible, and the research and development of emission control of aviation piston engine will also learn from the experience of automotive diesel engine. In this paper, engine emission under an instantaneous state is studied and the influence of different structural parameters on the emission is summarized starting from the basic parameters of the combustion chamber and by changing the structure parameters to provide the reference for the research of aero-diesel engine emission.

2. The Establishment of a Spray Numerical Model

Inside the cylinder of a direct injection engine, a series of physical changes occur when injecting diesel under high pressure, such as evaporation, break-up, interaction, and dispersion of diesel droplets. In order to better study diesel characteristics under combustion process, the numerical model is established for these processes.

2.1. The Diesel Droplet Evaporation Model. The Dukowicz model has been used to establish a model, and the following assumptions have been made: ① the temperature of the whole diesel droplet is the same, ② droplets are spherical, ③ the droplets are surrounded by a steady gas-phase flow field, ④ the droplets' surface is quasi-static, and ⑤ the inter-section of gas and liquid is in thermodynamic equilibrium.

According to the above assumptions, diesel droplets are injected into the cylinder. The energy absorbed by diesel droplets is mainly used for heating and evaporating itself as shown in

$$m_d c_{pd} \frac{dT_d}{dt} = L \frac{dm_d}{dt} + \alpha A_s (T_\infty - T_s). \quad (1)$$

In this equation, m_d is the mass of droplets, c_{pd} is the diesel-specific heat capacity under constant pressure, L is the latent heat of droplet evaporation, α is the convective heat-transfer coefficient, and A_s is the surface area of diesel droplets.

According to the thermodynamic equilibrium conditions, the expression equation of mass flow is shown in

$$\frac{dm_d}{dt} = Q \frac{f_{vs}}{q_s}. \quad (2)$$

In this equation, f_{vs} is the evaporative mass flow, q_s is the surface heat flux density of diesel droplets, and Q is the heat from in-cylinder air to the droplets.

$$\frac{f_{vs}}{q_s} = \frac{\rho \beta}{\kappa} \left(\frac{1}{1 - \mu_{vs}} \right) \frac{Le}{c_p} \left[\frac{h_{sv} - h_s}{\mu_{v,sv} - \mu_{vs}} - h_{sv} + h_{gs} \right]. \quad (3)$$

We set $Le = 1$, and the mass transfer coefficient is

$$B_y = \frac{\mu_{vco} - \mu_{vs}}{1 - \mu_{vs}}. \quad (4)$$

The Nusselt value is derived from the following formula according to the single droplet correlation theory:

$$N_\mu = 2 + 0.6 R_e^{1/2} P_r^{1/3}. \quad (5)$$

R_e and P_r can be acquired through the experiment.

From equations (1), (2), (3), (4), and (5), the Nusselt value is used to replace the convective heat transfer coefficient, and the heat absorbed by the diesel droplets is shown in

$$Q = D_d \pi \lambda N_\mu (T_\infty - T_s). \quad (6)$$

D_d is the droplet diameter, λ is the thermal conductivity of gas mixture, T_∞ is the gas mixture temperature, and T_s is the droplet temperature [20].

2.2. The Diesel Break-Up Model. After diesel is injected into the cylinder, the primary break-up occurs in the vicinity of the high-pressure nozzle, and the secondary break-up zone occurs in the farther area. The droplet radius can be estimated as shown in equation (7) by using the wave model:

$$r = \frac{2\pi\sigma B}{\rho_2 U^2}. \quad (7)$$

B is the magnitude constant 1, U is the velocity difference between gas-liquid two-phase motion, and σ is the surface tension of droplets.

The actual equation is

$$r = \begin{cases} B\Lambda, & \text{when } B\Lambda < a, \\ \min \left\{ \left(\frac{3\pi a^2 U}{2\Omega} \right)^{0.33}, \left(3a^2 \left(\frac{\Lambda}{4} \right) \right)^{0.33} \right\}, & \text{when } B\Lambda > a. \end{cases} \quad (8)$$

Jet surface wave formed during diesel injection. Λ is the corresponding wavelength, a is the amplitude critical value, and Ω is the maximum perturbation wave growth rate which causes break-up of the diesel jet. When the amplitude of the unstable wave is greater than the critical value, the diesel oil droplets break apart.

2.3. The Wall Interaction Model. Because of the small volume of the reciprocating engine, the diameter of the working cylinder is also not big. During the diesel injection process, the interaction with the cylinder wall is inevitably occurred. Then, a variety of movement trails are produced after interaction, such as adherent cylinder wall, rebound, and movement on the wall.

The wall-jet model is used. It is supposed that the wall liquid film does not influence the droplet movement after the

wall interaction. The movement forms of the interaction are judged according to the parameter We , and the particle size is associated with the Weber value after interaction. The equation is

$$\begin{cases} d_1 = d_0, & We < 50, \\ d_1 = d_0 \cdot f(We_{\perp, in}), & 50 \leq We \leq 300, \\ d_1 = 0.2d_0, & We > 300. \end{cases} \quad (9)$$

d_1 is the droplet diameter before wall interaction, and d_0 is the droplet diameter after wall interaction. When the We value is less than 50, the reflection interaction model is used. When the value is greater than 50, the jet interaction model is adopted.

2.4. The Turbulent Dispersion Model. Diesel is affected by the gas turbulence when injected into the cylinder. Then the random movement of the turbulent vortex causes a disturbance force to diesel droplets, and the turbulent dispersion model is mainly used to analyze the random disturbance. The smaller the droplet is, the more obvious the turbulent affect is, and the trail of the droplet motion is not smooth. The Gosman model is used, and u'_i is attached to the average velocity of gas u_g to describe the disturbance effect of turbulent dispersion on droplets. The equation is

$$u'_i = \left(\frac{2}{3}k\right)^{1/2} \cdot \text{sign}(2Rn_i - 1) \cdot \text{erf}^{-1}(2Rn_i - 1). \quad (10)$$

Rn_i is the random number of the velocity component (0~1), k is the turbulent energy of the droplet, and erf^{-1} is the inverse Gaussian function.

The time t_{turb} of interaction between diesel droplets and gas turbulence is shown in

$$t_{\text{turb}} = \min \left\{ C_\tau \frac{k}{\varepsilon}, C_r \frac{k^{3/2}}{\varepsilon} \frac{1}{|u_g + u'_i - u_d|} \right\}. \quad (11)$$

C_τ and C_r are both empirical coefficients, k is the turbulence pulsation energy, ε is the turbulence energy dissipation rate, u'_i is the gas pulsation speed, u_g is the gas speed, and u_d is the movement speed of diesel droplets.

When the action time of diesel droplets in the turbulent cluster $t > t_{\text{turb}}$, the droplet enters the next turbulent vortex cluster [21].

3. The Establishment of an Emission Numerical Model

3.1. NO_x Model. NO and NO_2 are the main forms of nitrogen oxide emissions of aviation diesel engines, with NO being the major element. The Zeldovich model in FIRE software can be used together with all combustion models, and the equation is shown in

At high temperature, the following reaction occurs among N_2 , O_2 , and NO , and the equation is as shown in

k_3 is a constant value, and k_1 , k_2 , k_4 , and k_5 are positive reaction rates related to temperature. $\text{m}^3/(\text{kmol}\cdot\text{s})$.

3.2. Soot Model. Soot is a major component of diesel particulate emissions. The Kennedy/Hiroyasu/Magnussen model is used to describe the formation of soot. The equation is

$$s_{\emptyset_s} = s_n + s_g + s_{o_2}. \quad (14)$$

$s_{\emptyset_s}$ is the soot formation rate, and s_n is the soot nucleus formation rate which is irrelevant to temperature. Partial oxidation and thermal cracking of hydrocarbon molecules take place under the condition of high temperature and oxygen deficiency to form various unsaturated hydrocarbons, which continuously dehydrogenate and polymerize into carbon-based soot nucleus. s_g is the surface generation rate of soot, which is proportional to temperature. s_{o_2} is the soot oxidation rate, which is proportional to temperature [22].

4. The Establishment and Analysis of the Combustion Chamber Model

The FIRE software is one of the professional software for engine combustion simulation which is launched by AVL Company in the USA and used by most of the engine R&D companies for simulating engine combustion [23]. We can put the engine parameters provided by the engine company like nozzle diameter and the top dead center gap into the corresponding module of FIRE. But the shape of the combustion chamber can only be required when the engine company provides the corresponding detailed drawings. In AVL FIRE, the simulated combustion process begins from the opening of the intake valve to the closing of the exhaust valve. The original data of the engine provided is that the closing angle of the intake valve is 36° behind the bottom dead center and the opening angle of the exhaust valve is 48° in front of the bottom dead center. According to general design principle, the top dead center defined is 720°CA and the bottom dead center is 540°CA , while the combustion simulation process is from 576°CA to 848°CA .

Before the accurate combustion simulation, the combustion chamber should be mesh-generated. Three dynamic meshes which are divided by FIRE software under different crank angles are shown in Figures 1 and 2.

FIGURE 1: The meshes of the combustion chamber under 740°C.

FIGURE 2: The meshes of the combustion chamber under 850°C.

In order to optimize the design of the combustion chamber, new programs are provided on the basis of the original one. Specific data is shown as Table 1, and the geometry and size of the combustion chamber are shown as Figure 3.

The optimization of the combustion chamber geometry shape can increase the turbulent motion properly, improve the mixed effects of fuel and air, speed up the combustion speed, and prolong the vortex duration. The reentrant-max diameter ratio and max diameter-max depth ratio of the combustion chamber are important parameters for optimization, where the reentrant-max diameter ratio refers to the ratio of the reentrant diameter to the maximum diameter,

which affects the squish effect in the combustion chamber. The max diameter-max depth ratio refers to the ratio between the maximum diameter and the maximum depth, which affects the mixture of fuel and air. These two ratios both affect the turbulence and squish flow effects in the cylinder during combustion, thereby affecting the combustion and emission of the engine.

Two programs are selected in order to study the influence of the basic size of the dumbbell-type combustion chamber on emission. On the basis of the original, one program changes the maximum diameter of the combustion chamber and its reentrant diameter to achieve the effect of changing

TABLE 1: Basic size of the combustion chamber.

Program	Max diameter (D), mm	Reentrant diameter (d), mm	Mid-depth (g), mm	Max depth (H), mm	Reentrant max diameter ratio (d/D)	Max diameter max depth ratio (D/H)
Original	51.3	42.4	8.0	17	0.83	3.01
Program 1	49.1	38	9.7	17	0.78	2.88
Program 2	49.1	38	9.4	19.5	0.78	2.51

FIGURE 3: Geometry and size of the combustion chamber.

6_12:AN_730.0:flow: temperature (K)

FIGURE 4: The temperature field under 730°C of the original program.

the reentrant-max diameter ratio, which is 0.83 in the original scheme and 0.77 after modification. On the basis of program 1, program 2 keeps the reentrant-max diameter ratio unchanged and changes the maximum depth to achieve the effect of changing the max diameter-max depth ratio. The max diameter-max depth ratio of the original is 3.0, and that of program 2 is 2.51. In the design of two programs, the mid-depth is constantly adjusted to achieve a constant compression ratio of the engine. The combustion numerical simulation analysis of two designing programs is carried out.

Three programs are analyzed and compared. Since the combustion is mainly carried out near the top dead center, the maximum pressure and maximum temperature appear under the piston downward at 10°C. Since the generation of NO_x and soot has a direct relationship with temperature,

730°C is thus selected for the analysis under transient state when the NO_x and soot emission directly reflects ultimate emissions. Therefore, the angle is analyzed and the temperature inside the cylinder is as shown in Figures 4–6.

It is concluded through analyzing the temperature fields under three programs that the temperature inside the cylinder will change together with the change in the chamber size, while the flame transmission rule has not changed and the maximum temperature appears in the same position as the flame transmission path. The temperature drops at 16.4 K in program 1 and at 42.4 K in program 2 compared with the original program, and the drop in temperature has a certain effect on the generation of NO_x and soot. This is mainly because the combustion chamber in program 1 adopts a smaller reentrant-max diameter ratio, and a larger squish

6_12:AN_730.0:flow: temperature (K)

FIGURE 5: The temperature field under 730°CA of program 1.

Case_2:AN_730.0:flow: temperature (K)

FIGURE 6: The temperature field under 730°CA of program 2.

flow is formed in the piston compression process, which improves the mixing quality of fuel and air and is beneficial to combustion. Also, the flame spreads more evenly, as shown in Figure 5. Therefore, the combustion temperature value in program 1 drops more than that in the original program. Program 2 reduces the max diameter-max depth ratio based on program 1, which improves the turbulent flow of the intake vortex formed in the combustion chamber. Meanwhile, with the max diameter-max depth ratio decreasing, the distance between the nozzle and the combustion chamber wall is reduced, which can strengthen the turbulence intensity near the chamber wall. Besides, a better quality of the mixture of fuel and air is obtained and the flame spreads more evenly; thus, the combustion efficiency is improved, as shown in Figure 6. The temperature of the combustion chamber drops.

The distributions of NO_x under 730°CA are, respectively, shown in Figures 7–9, which are similar to the temperature fields inside the cylinder. Besides, the temperature is the main reason for NO_x generation.

By comparing the new program with the original one, the maximum generation of NO_x was found to decrease. It decreases by 11.7% in program 1 and 28% in program 2, and there is a general downward trend. NO_x is related to the temperature in the combustion chamber. When the temperature in the combustion chamber drops, the concentration of NO_x will decrease. As indicated in equations 12 and 13, the positive reaction rate decreases when the temperature lowers, and the generated NO_x decreases. With the decrease in the reentrant-max diameter ratio in program 1, the temperature field in the combustion chamber drops, and the production of NO_x decreases accordingly. Meanwhile,

FIGURE 7: The distribution of NO_x under 730°CA of the original program. FIGURE 8: The distribution of NO_x under 730°CA of program 1.

the velocity of the flow field in the combustion chamber will accelerate, resulting in a more even distribution of NO_x than the original. On the basis of program 1, program 2 continues to reduce the max diameter-max depth ratio, and the temperature of the combustion chamber is lower than that of program 1, so the generation of NO_x is more reduced. At the same time, the velocity of the flow field in the combustion chamber increases more and the turbulence is strengthened. Under the action of a strong flow field, the distribution of NO_x in the combustion chamber is more even.

As shown in Figures 10–12, soot generation analysis of original and new programs is done. Soot is generated mainly in the diffusion combustion phase, mainly formed in the bottom of the combustion chamber, so the distributions and temperature fields are slightly different. It is found that, through comparison, the maximum generation of soot has

risen by 5.8% in program 1 and reduced by 3.6% in program 2 based on that of the original. Soot is associated with the combustion temperature, and the combustion high efficiency in the cylinder can reduce soot generation. As shown in equation 14, the rate of formation of soot is related to temperature. The surface generation rate of soot and the soot oxidation rate decrease when the temperature in the cylinder lowers so that soot decreases. Program 1 reduces the reentrant-max diameter ratio. However, the maximum depth remains the same. So it is easy to touch the bottom of the combustion chamber in the process of jet fuel, and at a certain moment the combustion gets affected and soot is generated easily. We can clearly see, in Figure 11, that soot on the bottom left side of the combustion chamber apparently increases. In the meantime, the period under 730°CA belongs to the diffusion stage of combustion in the cylinder. At this

Case_2:AN_730.0:emission:NO_mass_fraction(-)

FIGURE 9: The distribution of NO_x under 730°CA of program 2.

6_12:AN_730.0:emission:soot_mass_fraction(-)

FIGURE 10: The distribution of soot under 730°CA of the original program.

stage, the quality of a local mixture is poor, resulting in high temperature and oxygen deficiency, and the production of carbon smoke increases in a short time. Program 2 reduces the max diameter-max depth ratio based on program 1. Thus, the max depth of the combustion chamber increases and enlarges the fuel atomization space. Meanwhile, the flow field of the combustion chamber speeds up due to squish flow effects. Under the action of these favorable factors, soot reduces.

After analyzing and comparing modified programs for the combustion chamber, it is clearly shown that NO_x emission decreased significantly in program 2, while soot generation has a slight reduction. Generally speaking, emission still tends to decrease.

It is beneficial to decrease the reentrant-max diameter and max diameter-max depth ratio appropriately, but the combustion of the engine will be affected if the size of the

combustion chamber is greatly modified or the reentrant-max diameter and max diameter-max depth ratio are greatly reduced. Too much decrease in the max diameter-max depth ratio will lead to fuel directly spraying on the inner wall of the combustion chamber. In this way, fuel attached to the inner wall promotes the generation of soot and emissions worsen. Too much reduction in the reentrant-max diameter ratio makes it difficult to form an orderly flow field in the combustion chamber, which will affect the mixture of fuel and air then bring down the combustion quality and worsen the emission.

5. The Experiment

According to the size of the combustion chamber of program 2, the bench test is carried out after the assembly, and the AVL and opaque smoke meter instrument is connected.

6_12:AN_730.0:emission:soot_mass_fraction(-)

FIGURE 11: The distribution of soot under 730°C of program 1.

Case_2:AN_730.0:emission:soot_mass_fraction(-)

FIGURE 12: The distribution of soot under 730°C of program 2.

TABLE 2: AVL DITEST Gas 1000 technical parameter.

Measured	Measuring	Resolution	Accuracy
CO	0-15% vol.	0.01% vol.	<10.0% vol., $\pm 0.02\%$ vol., $\pm 3\%$ o.M. $\geq 10.0\%$ vol., $\pm 5\%$ o.M.
NO _x	0-5000 ppm vol.	1 ppm vol.	± 5 ppm vol. $\pm 1\%$ o.M.
HC	0-30000 ppm vol.	$\leq 2.000 : 1$ ppm vol.	<2000 ppm vol., ± 4 ppm vol., $\pm 3\%$ o.M. ≥ 5000 ppm vol., $\pm 5\%$ o.M. ≥ 10000 ppm vol., $\pm 10\%$ o.M.
CO ₂	0-20% vol.	0.01% vol.	<16.0% vol., $\pm 0.3\%$ vol., $\pm 3\%$ o.M. $\geq 16.0\%$ vol., $\pm 5\%$ o.M.
O ₂	0-25% vol.	0.01% vol.	$\pm 0.02\%$ vol. $\pm 1\%$ o.M.
λ (excess air factor)	0-9.999	0.001	Calculated from CO, CO ₂ , HC, O ₂

FIGURE 13: The measurement unit of AVL DITEST Gas 1000.

The main equipment used in the experiment is the AVL DITEST Gas 1000 emission analyzer to detect NO_x in the exhaust gas. The size of the equipment is small around 344 mm* 252 mm* 85 mm, and the weight is light. The parameters in the measurement process can be read directly, and exhaust gas can be diagnosed by itself. Technical parameters are shown in Table 2.

The measurement unit of AVL DITEST Gas 1000 is shown in Figure 13. The function of each connector is marked in the figure.

The AVL439 opaque smoke meter can be used to detect the soot of exhaust gas. This meter is designed on the basis of Beer-Lamber's rule as shown in equation 15. The device is 650 mm* 420 mm* 450 mm in size and is connected to the engine exhaust pipe according to the designed connector.

$$K = \frac{1}{L} \cdot \ln \left(1 - \frac{N}{100} \right). \quad (15)$$

N is the opacity, in %; K is the absorption, in m^{-1} ; and L is the length, in m.

The main technical parameters of the equipment are shown in Table 3.

The above equipment on the bench is installed, as shown in Figure 14, and the engine is tested. The test steps are as follows:

- (1) Install it on the engine test bench according to the AVL analyzer and smoke meter instructions

TABLE 3: AVL439 main technical parameter.

Item	Parameter
Absorption range	0-10 m^{-1}
Resolution ratio	0.0025 m^{-1}
Sample frequency	50 Hz
Response time	0.1 s
Allowable exhaust pressure	-100 mbar-400 mbar

- (2) Start the engine and wait for oil temperature to reach 70°C, and start measuring emissions data under different RPM
- (3) Insert the sampling tube of AVL DITEST Gas 1000 into the engine exhaust pipe and read the data
- (4) Read the opaque data on the smoke meter
- (5) After the measurement, the air pump shall keep working for 10 min to clean the internal pipeline of AVL DITEST Gas 1000

The engine was tested and the engine basic parameters and emissions data are recorded. The recorded data is shown in Table 4. The engine power torque does not change substantially, but the emissions were reduced compared to the original combustion chamber.

It can be seen from the experimental data that the emissions corresponding to program 2 and the experimental data of the original have a downward trend, and the data of the decline is basically consistent with the data calculated by

Bench test

AVL monitoring systems

FIGURE 14: Engine bench test.

TABLE 4: Emission data under different operating conditions.

RPM	kW	N.m	Opacimeters (m^{-1})		NO_x (ppm)	
			Original	Program 2	Original	Program 2
2700	71	250	0.25	0.23	863	684
2500	62	235	0.17	0.15	671	527
2300	43	180	0.07	0.06	478	373
2100	30	136	0.02	0.02	298	230

the simulation, which indicates that the simulation calculation method is feasible.

6. Conclusion

- (1) Reducing the maximum diameter and reentrant diameter of the combustion chamber can reduce the combustion temperature in the engine cylinder, which can reduce the generation of NO_x , but the amount of soot is increased. The maximum diameter size is reduced by 4.3%, the reentrant diameter size was reduced by 10.4%, the temperature was reduced by 16.4K, the NO_x was reduced by 11.7%, and the soot was increased by 5.8%
- (2) The maximum depth and middle depth of the combustion chamber can also affect the temperature and emissions in the cylinder. The maximum depth increases by 14.7%, NO_x decreases by 28%, and the soot maximum production decreases by 3.6%
- (3) In the bench test of the modified combustion chamber, the engine power and torque are basically unchanged, but the emission generation is indeed reduced, which proves that the optimization of the combustion chamber and analysis is feasible

Data Availability

The simulation and experimental data used to support the findings of this study are included within the article. The size data of the combustion chamber and the combustion simulation data have been recorded in the manuscript, and the bench test data is also recorded and analyzed in the manuscript and analyzed. The figures and tables in the manuscript record the data in detail, such as Table 1 which shows the basic size of the combustion chamber, Table 2 shows the emission data, and Figures 4–12 record the simulation data.

Disclosure

We declare that we have no financial and personal relationships with other people or organizations that can inappropriately influence our work.

Conflicts of Interest

There is no conflict of interest regarding the publication of this article.

Acknowledgments

I highly appreciate the funding under project number K1705041 provided by Changsha Science and Technology Bureau and the great support of our team members. Last but not the least, the authors are thankful to the director for the permission to publish this work.

References

- [1] Z. Yu, D. S. Liscinsky, E. C. Fortner et al., "Evaluation of PM emissions from two in-service gas turbine general aviation aircraft engines," *Atmospheric Environment*, vol. 160, pp. 9–18, 2017.
- [2] M. Wise, M. Muratori, and P. Kyle, "Biojet fuels and emissions mitigation in aviation: an integrated assessment modeling

- analysis," *Transportation Research Part D: Transport and Environment*, vol. 52, pp. 244–253, 2017.
- [3] Z. Pan, Q. He, X. Zhang, and D. Zhang, "Numerical simulation of 2-stroke diesel engine for light aircraft," *IEEE Aerospace and Electronic Systems Magazine*, vol. 30, no. 3, pp. 4–10, 2015.
 - [4] L. Chen, S. Ding, H. Liu, Y. Lu, Y. Li, and A. P. Roskilly, "Comparative study of combustion and emissions of kerosene (RP-3), kerosene-pentanol blends and diesel in a compression ignition engine," *Applied Energy*, vol. 203, pp. 91–100, 2017.
 - [5] W. Balachandran, N. Manivannan, R. Beleca et al., "Nonthermal plasma system for marine diesel engine emission control," *IEEE Transactions on Industry Applications*, vol. 52, no. 3, pp. 2496–2505, 2016.
 - [6] T. Tang, D. Cao, J. Zhang, Y.-G. Zhao, and S.-J. Shuai, "Experimental study of catalyzed diesel particulate filter with exhaust fuel injection system for heavy-duty diesel engines," SAE Technical Paper 2014-01-1496, 2014.
 - [7] M. Chmielewski and M. Gieras, "Impact of variable geometry combustor on performance and emissions from miniature gas turbine engine," *Journal of the Energy Institute*, vol. 90, no. 2, pp. 257–264, 2017.
 - [8] M. Bogarra, J. M. Herreros, A. Tsolakis, A. P. E. York, P. J. Millington, and F. J. Martos, "Impact of exhaust gas fuel reforming and exhaust gas recirculation on particulate matter morphology in gasoline direct injection engine," *Journal of Aerosol Science*, vol. 103, pp. 1–14, 2017.
 - [9] M. Capobianco and S. Marelli, "Experimental analysis of unsteady flow performance in an automotive turbocharger turbine fitted with a waste-gate valve," *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*, vol. 225, no. 8, pp. 1087–1097, 2011.
 - [10] N. Stamoudis, C. Chrysosakis, and L. Kaiktsis, "A two-component heavy fuel oil evaporation model for CFD studies in marine diesel engines," *Fuel*, vol. 115, pp. 145–153, 2014.
 - [11] A. K. Wamankar and S. Murugan, "Combustion performance and emission characteristics of a diesel engine with internal jet piston using carbon black- water- diesel emulsion," *Energy*, vol. 91, pp. 1030–1037, 2015.
 - [12] Y. Liu, X. Sun, V. Sethi, D. Nalianda, Y. G. Li, and L. Wang, "Review of modern low emissions combustion technologies for aero gas turbine engines," *Progress in Aerospace Sciences*, vol. 94, pp. 12–45, 2017.
 - [13] A. Innocenti, A. Andreini, D. Bertini, B. Facchini, and M. Motta, "Turbulent flow-field effects in a hybrid CFD-CRN model for the prediction of NO_x and CO emissions in aero-engine combustors," *Fuel*, vol. 215, pp. 853–864, 2018.
 - [14] C. Xisto, O. Petit, T. Grönstedt, and A. Lundblad, "Assessment of CO₂ and NO_x emissions in intercooled pulsed detonation turbofan engines," *Journal of Engineering for Gas Turbines and Power*, vol. 141, no. 1, article 011016, 2019.
 - [15] Z. Pan and Q. He, "High cycle fatigue analysis for oil pan of piston aviation kerosene engine," *Engineering Failure Analysis*, vol. 49, pp. 104–112, 2015.
 - [16] A. P. Carlucci, A. Ficarella, D. Laforgia, and A. Renna, "Supercharging system behavior for high altitude operation of an aircraft 2-stroke diesel engine," *Energy Conversion and Management*, vol. 101, pp. 470–480, 2015.
 - [17] C. A. Rinaldini, E. Mattarelli, and V. Golovitchev, "CFD analyses on 2-stroke high speed diesel engines," *SAE International Journal of Engines*, vol. 4, no. 2, pp. 2240–2256, 2011.
 - [18] E. Mattarelli, C. A. Rinaldini, G. Cantore, and P. Baldini, "2-Stroke externally scavenged engines for range extender applications," SAE Technical Paper 2012-01-1022, 2012.
 - [19] E. Mattarelli, F. Paltrinieri, F. Perini, C. A. Rinaldini, and M. Wilksch, "2-Stroke diesel engine for light aircraft: IDI vs. DI combustion systems," SAE Technical Paper 2010-01-2147, 2010.
 - [20] V. Ebrahimian and C. Habchi, "Towards a predictive evaporation model for multi-component hydrocarbon droplets at all pressure conditions," *International Journal of Heat and Mass Transfer*, vol. 54, no. 15-16, pp. 3552–3565, 2011.
 - [21] W. Qi, P. Ming, W. Zhang, and Y. Peng, "A simulation method and the effect of turbulence perturbation on droplet breakup," *Journal of Harbin Engineering University*, vol. 39, no. 4, pp. 709–715, 2018.
 - [22] J. Y. Tan, F. Bonatesta, H. K. Ng, and S. Gan, "Developments in computational fluid dynamics modelling of gasoline direct injection engine combustion and soot emission with chemical kinetic modelling," *Applied Thermal Engineering*, vol. 107, no. 25, pp. 936–959, 2016.
 - [23] D. K. Soni and R. Gupta, "Numerical analysis of flow dynamics for two piston bowl designs at different spray angles," *Journal of Cleaner Production*, vol. 149, pp. 723–734, 2017.