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On the Design and Optimization of a Clean and Efficient Combustion Mode for Internal Combustion Engines through a Computer NSGA-II Algorithm

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Abstract: In order to address typical problems due to the huge demand of oil for consumption in traditional internal combustion engines, a new more efficient combustion mode is proposed and studied in the framework of Computational Fluid Dynamics (CFD). Moreover, a Non-dominated Sorting Genetic Algorithm (NSGA-II) is applied to optimize the related parameters, namely, the engine methanol ratio, the fuel injection time, the initial temperature, the Exhaust Gas Re-Circulation (EGR) rate, and the initial pressure. The so-called Conventional Diesel Combustion (CDC), Homogeneous Charge Compression Ignition (HCCI) and the Reactivity Controlled Compression Ignition (RCCI) combustion modes are compared. The results show that RCCI has a higher methanol ratio and an earlier injection timing with moderate EGR rate and higher initial pressure. The initial temperature increases as the methanol ratio increases. In comparison, CDC has the lowest hydrocarbon and CO emissions and the highest combustion efficiency. At different crankshaft rotation angles corresponding to 50% of the combustion amount (CA50), the combustion temperature and boundary layer temperature of HCCI change significantly, while those of RCCI undergo limited variations. At the same CA50, the exergy losses of HCCI and RCCI are lower than that of the CDC. On the basis of these findings, it can be concluded that the methanol/diesel RCCI engine can be used to obtain a clean and efficient combustion process, which should be regarded as a promising combustion mode.

Keywords: Computer-optimized NSGA-II algorithm; novel clean and efficient combustion mode; thermodynamics; combustion engine; methanol

1 Introduction

The major fuel of traditional internal combustion engines is petroleum. With the rapid development of the automobile industry, a large number of petroleum resources have been exploited, and the urban environment has been seriously threatened. In addition, exhaust gases emitted by internal combustion engines, such as carbon monoxide, sulfur dioxide, hydrocarbons and nitrogen oxides (NOx), will not only endanger people's health but also cause acid rain and photochemical smog, damaging the natural ecosystems [1]. Given the multiple pressures of resources, health, and environment, it has become a research hot spot in the industry to propose a clean and efficient combustion mode. At present, researchers have proposed a variety of advanced and efficient combustion modes, such as Homogeneous Charge Compression Ignition (HCCI), Premixed Charge Compression Ignition (PCCI), and Reactivity Controlled Compression Ignition (RCCI) [2]. Studies have shown that compared with HCCI and PCCI, PCCI has more flexible and effective control methods. It can adjust the injection timing of diesel to control the ignition time and combustion process and adjust the proportion of premixed fuel to control the combustion rate [3].To fundamentally resolve the contradiction between the supply and demand of petroleum, in addition to improving the combustion process in the cylinder of the internal combustion engine, clean and efficient fuels, such as alcohol, bio-diesel, and dimethyl ether, should be promoted to replace petroleum. Among them, bio-diesel and dimethyl ether have limitations in production and fuel viscosity, which cannot be used on a large scale. Methanol has a wide range of sources, which can be produced from coal, oil, natural gas, and biomass; therefore, it can be mass-produced. It has unique physical and chemical properties, which is conducive to the full combustion of fuel and reduces emissions [4]. Traditional diesel is the most commonly used highly active fuel in RCCI engines. Zhang et al. [5] studied the performance of methanol/diesel RCCI engines and proposed that compared with gasoline/diesel RCCI, methanol/diesel RCCI has higher thermal efficiency and lower emissions of soot particulates. Several studies have discussed the superiority of methanol/diesel RCCI engines, which will be the research object of this study.

To obtain excellent performance of the engine, it is necessary to comprehensively consider the influence between operating parameters for optimization. In recent years, genetic algorithms have been widely used in the optimization design of internal combustion engines. Among them, (Non-dominated Sorting Genetic Algorithm (NSGA)-II has obvious advantages in convergence speed and optimization results [6]. Jin et al. [7] optimized the NSGA-II so that the gasoline/diesel RCCI can cover a larger operating load with higher fuel economy and lower exhaust emissions, but the engine is sensitive to initial conditions and boundary conditions, and corresponding measures need to be taken to guarantee the stability of its operation.

The innovation of this study is to utilize a genetic algorithm for optimizing multiple parameters of methanol/diesel RCCI engine synchronously and apply the first and second laws of thermodynamics in comparing and analyzing the influence of combustion mode and fuel type on the energy and exergy distribution in the engine cylinder. This study has provided guidance for reducing engine exergy losses and improving energy efficiency. At present, there have been studies applying computer-optimized NSGA-II algorithms to natural gas/diesel RCCI and gasoline/diesel RCCI engines, but few studies have optimized the performance of methanol/diesel RCCI engines. Therefore, this study uses NSGA-II to optimize the engine performance of RCCI with novel clean and effective combustion mode, providing a reliable reference for optimizing the combustion process of internal combustion engines and controlling strategies for reducing harmful exhaust emissions.

2 Experiments and Methods

2.1 CFD Numerical Model

The Generalized Re-Normalization Group (gRNG) k-ε model can accurately capture complex transient and anisotropic turbulent flows in the cylinder and has good prediction accuracy [8]. Therefore, this study adopts the gRNG k-ε model as the turbulence model in CFD. The turbulent kinetic energy transport equation of the gRNG k-ε model is shown in Eq. (1).

$$\frac{D(\rho k)}{Dt} = P - \rho\varepsilon + D_k \tag{1}$$

where D/Dt represents the mass derivative, k represents the turbulent kinetic energy, P represents the turbulent kinetic energy generation term, ρ is the density, ε represents the turbulent energy dissipation rate, and D_k represents the turbulent diffusion term.

The turbulent dissipation rate equation is shown in Eq. (2).

$$\frac{D(\rho\varepsilon)}{Dt} = C_1 \frac{P\varepsilon}{k} + C_1' \rho v_t \frac{\varepsilon}{k} (\nabla \cdot \overline{u})^2 - C_{2n} \rho \frac{\varepsilon^2}{k} - \rho R + C_3 \rho \varepsilon (\nabla \cdot \overline{u}) + D_\varepsilon \tag{2}$$

where C_1, C_1', C_{2n}, and C_3 are experience constants, ∇ is Hamiltonian, v_t is the eddy viscosity coefficient, $\overline{u}$ is the average velocity vector, and D_ε represents a turbulent diffusion term.

In this study, the spray crushing model in CFD adopts KH-RT (Kelvin-Helmholtz/Rayleigh-Taylor) crushing model [9]. After the fuel leaves the nozzle, there will be two crushing processes. The first crushing occurs near the nozzle, generating sub-liquids, and the second crushing occurs downstream of the fuel jet. It is assumed that the initial particle diameter of the fuel is the effective diameter of the nozzle, and the radius of the sub-droplets generated in the first crushing is as shown in Eq. (3).

$$r_c = B_0 \Lambda_{KH} \tag{3}$$

where B_0 is an empirical constant, Λ_{KH} indicates the wavelength at which a rapidly generated wave is formed on the droplet surface.

The crushing length calculation equation is shown in Eq. (4).

$$L_d = C_d d_0 \sqrt{\rho_{fuel}/\rho_{air}} \tag{4}$$

Among them, C_d is the empirical constant, and d_0 represents the droplet diameter of the parent.

In the wall heat transfer model, it is assumed that the gas is ideal, the pressure in the space is evenly distributed, and the velocity of the fluid is parallel to the wall surface. Only the density gradient and temperature gradient perpendicular to the wall surface is considered, while the heat transfer caused by viscous dissipation and radiation transmission is ignored. In the combustion model, chemkin-II is coupled with kiva-3v to solve the chemical reaction process. Within each time step, chemkin-II reads the component concentration and thermal parameters of each unit from kiva-3v. Each unit is regarded as a homogeneous stirred reactor, and chemkin-II chemical reaction calculation is performed. Then, the result is returned to kiva-3v for the in-cylinder flow field solution. At the next time step, the above process is repeated until the calculation ends. During the combustion process, the components of NO_x emissions are mainly Nitric Oxide (NO) and Nitrogen Dioxide (NO_2). The production of NO is higher than that of NO_2, and the production of NO_2 mostly comes from NO. Therefore, the emphasis is on the generation of NO. In the engine, NO comes from heat generation. Therefore, in the NO_x model, the focus is on the thermodynamic pathway of NO. The carbon soot model in this study uses a semi-empirical soot model. The reaction mechanism includes a six-step reaction path of precursor generation, carbon particle formation, surface growth, carbon particle condensation, surface oxidation, and precursor oxidation [10].

To verify the reliability of the CFD numerical model proposed in this study, a simulation verification is performed on the RCCI engine. Considering the huge costs of the simulation calculation, only the process from the moment when the intake valve closes to the moment when the exhaust valve opens is considered, while the intake and exhaust processes of the engine are not considered for the time being. Also, it is assumed that at the moment when the intake valve is closed, all the mixture in the cylinder is evenly distributed, and the initial pressure and initial temperature of the mixture in the cylinder are given at the beginning of the calculation.

2.2 RCCI Design Based on a Computer-Optimized NSGA-II Algorithm

NSGA-II is one of the most popular multi-objective genetic algorithms [11]. It reduces the complexity of non-inferior ranking genetic algorithms and has the advantages of fast running speed and good convergence of solution set [12]. It uses a fast non-dominated sorting algorithm to reduce complexity; in addition, it uses congestion and congestion comparison operators and serves as the winning criterion in the same-level comparison after quick sorting, so that individuals in the quasi-Pareto domain can be evenly distributed throughout the Pareto domain to maintain the diversity of the population [13]. The introduction of an elite strategy expands the sampling space, prevents the loss of the best individual, and improves the speed and robustness of the algorithm [14]. The calculation flowchart of NSGA-II is shown in Fig. 1. First, the ranking level is determined according to the number of individuals dominating between individuals, the crowding distance is calculated, and the individual's final fitness is obtained from it. The Pareto optimal solution set is selected from all the individuals in the population. The individuals with the lower ranking are selected first. If the ranking of the individuals is the same, the individuals with the larger crowding distance are preferred.

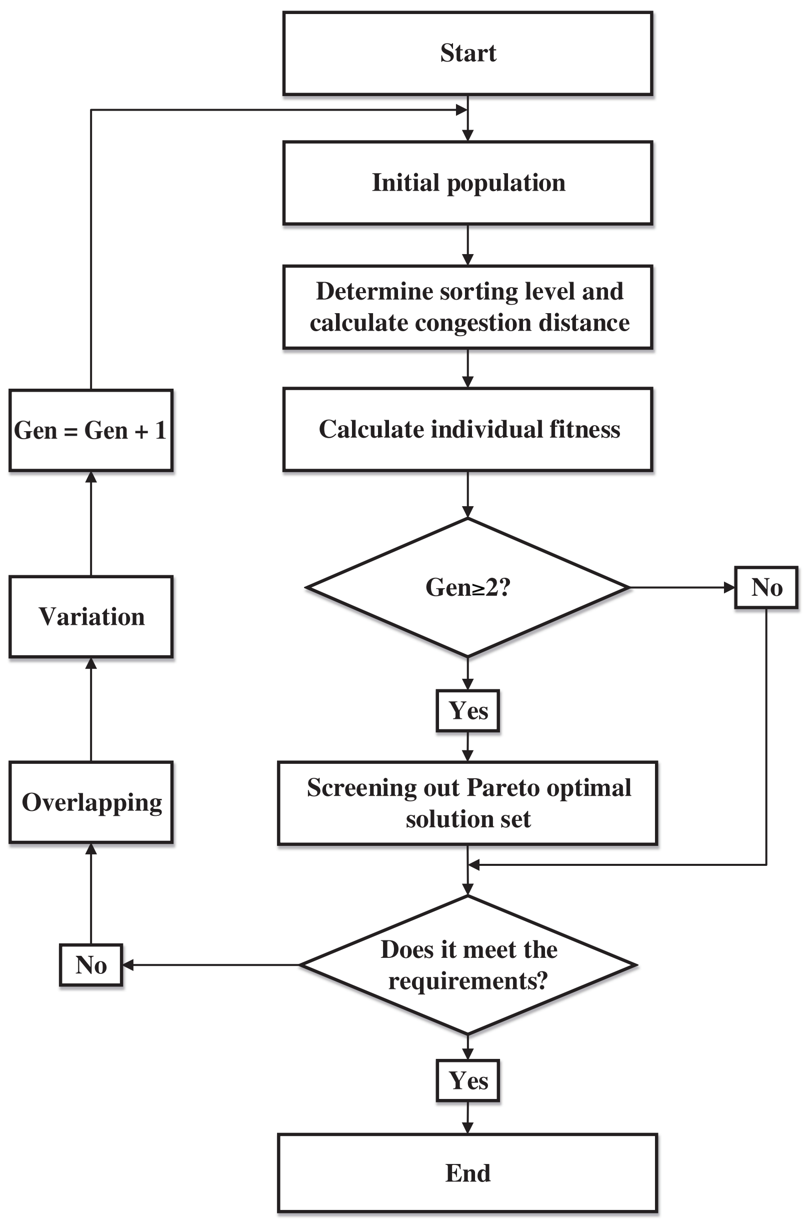

Figure 1: Calculation flowchart of NSGA-II

The NSGA-II algorithm is coupled with the three-dimensional CFD open-source program to coordinately optimize the operating parameters of the engine, thereby optimizing the performance of the RCCI. The NSGA-II algorithm is used to randomly generate the primary population, and each individual is brought into the program to solve by master-slave parallel computing. The target value is returned to NSGA-II to determine the fitness of each individual, select the contemporary population from a collection of optimal individuals, and generate the next generation of individuals through operations such as crossover and mutation. The parameters of the new individual are brought into the program again for the solution. Starting from the second generation, each generation of the population must be merged with the previous generation into a mixed population, and the optimal individual is selected from the mixed population according to the non-dominated ranking and crowded distance. The crossover and mutation operations are performed to generate next-generation populations and bring them into the program for calculation. The above steps are repeated until the calculation is finished.

This study uses the computer to optimize the NSGA-II algorithm to coordinate the effects of various parameters on the engine, thereby comprehensively improving the performance of the engine. In a methanol/diesel RCCI engine, both the methanol ratio and the injection timing can significantly affect the concentration and activity distribution of the fuel in the cylinder. Increasing the methanol ratio can reduce the overall activity of the fuel in the cylinder, delay the ignition time, and make the transition of the active gradient in the cylinder gentler. Therefore, at a higher methanol ratio, the oxygen in the cylinder can be used more comprehensively during combustion, which completely oxidizes the hydrocarbons, carbon monoxide, and soot, as well as shortens the duration of high-temperature reactions and reduces NO_x emissions. At the earlier injection time, the ignition time is delayed, which makes the diesel and air mix more evenly. The premixed combustion ratio increases and the combustion rate accelerates, resulting in increased peaks in sound intensity and heat release rate. By increasing the initial temperature of the intake air, it can be ensured that the ignition timing will not be too late, and the Exhaust Gas Re-circulation (EGR) rate can also help control the ignition timing of the engine and effectively suppress the generation of NO_x. In addition, the initial pressure is of great significance for the high load expansion of the engine and the overall equivalent ratio in the cylinder. Therefore, in this study, engine X is optimized for five parameters, i.e., methanol ratio, injection timing, initial temperature, EGR rate, and initial pressure. The optimized target values are set to NOx, Equivalent Indicated Specific Fuel Consumption (EISFC), and soot. To exclude calculation examples of excessive combustion, insufficient combustion, and misfire, the limiting conditions are set to the minimum temperature in the cylinder above 1000 K, the maximum pressure in the cylinder below 13 MPa, and the maximum pressure increase rate above 20 bar/°CA. At the same time, the pre-mixed energy fraction is used to represent the pre-mixed ratio of the fuel. The definition of the pre-mixed ratio is as shown in Eq. (5).

$$R_a = \frac{M_a \cdot H_a}{M_a \cdot H_a + M_b \cdot H_b} \tag{5}$$

where M_a represents the quality of the premixed fuel, H_a represents the low heating value of the premixed fuel, M_b represents the quality of the premixed diesel, and H_b represents the low heating value of the premixed diesel. Despite how the energy ratio of premixed methanol changes, the total energy of diesel and methanol contains the same fuel. The range of optimization parameters is as follows. The methanol energy ratio is 0–100%; the injection time is −50–0° CA After Top Dead Center (ATDC); the initial temperature is 320–420 K; the EGR rate is 0–100%, and the initial pressure is 1.0–2.1 bar.

2.3 Thermodynamic Analysis

The first law of thermodynamics (energy analysis) is usually used to measure the thermal efficiency of the engine. The numerical differences in energy can be compared. Based on the first law of thermodynamics, the second law of thermodynamics (exergy analysis) is added to analyze the engine's thermal conversion and loss of energy during cycling. This study uses thermodynamic analysis to analyze the relationship between Equivalent Indicated Specific Fuel Consumption (EISFC) and Ringing Intensity (RI) in three combustion modes, i.e., Conventional Diesel Combustion (CDC), HCCI, and RCCI. Also, the energy utilization of the three modes is compared to find out the main factors affecting energy loss, thereby proposing an optimization scheme to improve energy utilization. The operating conditions of the engine are as follows: the average effective pressure is 5.8 bar, and the speed is 2000 rev/min. In view of the significant impact of CA50 on engine performance, each combustion mode is successively run at four different CA50 moments: 3°CA ATDC, 5°CA ATDC, 7°CA ATDC, and 9°CA ATDC, with a total of 12 calculation examples [15]. CA50 represents the crankshaft rotation angle corresponding to 50% of the combustion amount, which can indirectly reflect the phase of the combustion heat release curve, that is, "early" and "late" in the combustion heat release process [16].

3 Results and Discussion

3.1 Validation Results of CFD Numerical Model in RCCI Engine

When the intake port is pre-mixed with methanol and the diesel is directly injected into the cylinder, under different operating loads, the comparison results of simulated prediction and experimental measurement of the in-cylinder pressure and heat release rate are shown in Fig. 2. It can be found that the pressure and heat release rate in the cylinder calculated by the CFD numerical model are basically consistent with the experimental measurement results no matter what operating condition the RCCI engine is running under. The simulation prediction can capture the change characteristics of the pressure and heat release rate in the cylinder under different operating loads.

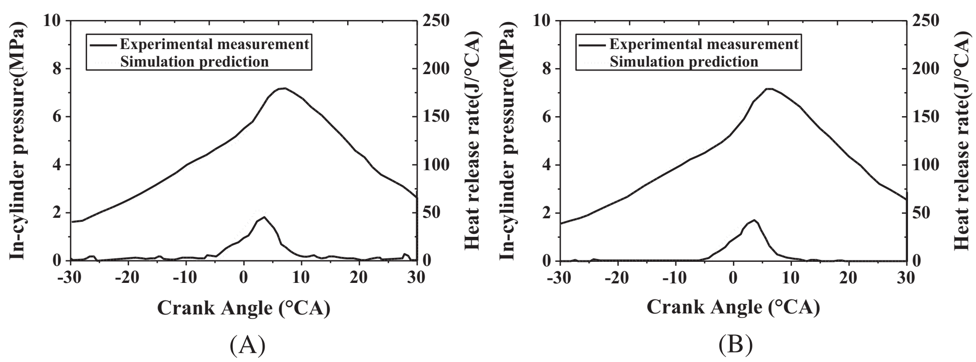

Figure 2: The Comparison of in-cylinder pressure and heat release rate between simulated prediction and experimental measurement (A: injection pressure of 3.7 bar and an engine speed of 2000 rpm; B: injection pressure of 4.0 bar and an engine speed of 1500 rpm)

As shown in the results, the numerical models used can simulate the combustion process of the fuel in the cylinder of the engine well, and reproduce the change rule of the emission products well [17]. There is a small deviation between the predicted emissions and the experimental values. This is due to the use of several assumptions in the model, as well as the existence of the mechanism of engine performance and multiple factors. Although there is a deviation between the predicted emissions and the experimental values, the deviation is small, and the emission products predicted by the calculation model can accurately capture the changing trend of the emissions. Therefore, using CFD numerical model to conduct qualitative research can ensure the accuracy and reliability of simulation results.

3.2 Optimization Results of Computer-Optimized NSGA-II Algorithm for RCCI

The distribution of methanol ratio, injection timing, initial temperature, EGR rate, and initial pressure for all optimization examples are shown in Fig. 3. From the distribution of the methanol ratio and injection timing of the optimization example (Fig. 3A), it can be seen that there are three types of all optimization examples, i.e., higher methanol ratio and earlier injection timing, medium methanol ratio and top dead center injection time, and lower methanol ratio and earlier injection time. In the RCCI engine, a higher methanol ratio and an earlier injection timing can simultaneously achieve better fuel economy and lower NO_x emissions. From the distribution of methanol ratio and injection time of all optimization examples (Fig. 3B), it can be seen that the initial temperature of the optimization example increases as the methanol ratio increases and the fuel activity of methanol is lower. After being raised, the combustion time will be delayed. At this time, the initial temperature needs to be increased to make the engine run at the optimal combustion phase time. From the distribution of methanol ratio and EGR rate in all optimization examples (Fig. 3C), it can be seen that a higher fuel economy can be achieved when the methanol ratio is greater than 12%. If the methanol ratio is too low, the methanol distribution concentration in the cylinder is low, and the premixed methanol in the low-temperature region near the wall surface cannot be fully ignited, thereby increasing the emissions of NO_x. In addition, the proportion of direct-injection diesel is too high, and the overall activity of the fuel in the cylinder is high. Therefore, it will burn ahead of time, increasing heat transfer loss and negative work. At the same time, a large amount of direct-injection diesel will gather in the cylinder to form a local rich area to make combustion deteriorated, which is not conducive to fuel economy. Therefore, the proportion of methanol should be appropriately increased to improve fuel economy. When the methanol ratio is 12% to 30%, the EGR rate of the optimal example is 49% to 37%. When the methanol ratio is greater than 30%, the EGR rate of the optimal example is between 28% and 39%. Increasing the proportion of methanol will significantly prolong the ignition time, resulting in incomplete combustion, thereby misfire. After increasing the EGR rate, the combustion temperature will decrease, which will also significantly extend the duration of combustion and reduce fuel economy. Therefore, to reduce NO_x emissions to the greatest extent, the value of the EGR rate should be higher than 28%. Also, when a higher methanol ratio is selected, the value of the EGR rate should not be higher than 39%. From the distribution of methanol ratio and initial pressure in all optimization examples (Fig. 3D), it can be seen that the initial pressure and the methanol ratio of the optimization examples are not significantly related. Generally, higher initial pressure is used to provide sufficient oxygen for the fuel to ensure comprehensive combustion.

With NSGA-II as the optimization tool, the optimization targets are NOx, fuel economy, and soot emissions. Comprehensive system optimization is performed for five important parameters of the methanol/diesel RCCI engine. After successive generations of gradual evolution, the three optimization goals have been significantly improved. The methanol/diesel RCCI engine has achieved an efficient, clean, and stable combustion process, which has improved fuel economy.

3.3 Thermodynamic Analysis Results

The relationship between EISFC and RI in the three combustion modes of CDC, HCCI, and RCCI is shown in Fig. 4. Comparing the EISFC of the three combustion modes, the CDC is the highest, followed by RCCI and HCCI.

Of the three combustion modes, when the combustion phase of HCCI is delayed to sufficiently late time, the limit requirement of RI can be reached. However, CDC lacks the assistance of EGR, and its sound intensity is high. Therefore, no matter how the combustion phase is delayed, it cannot meet the RI limit. In comparison, RCCI has the slowest combustion process; therefore, its IR is the lowest.

The comparison results of the energy distribution of the three combustion modes of CDC, HCCI, and RCCI are shown in Tab. 1. The energy distribution includes four parts, i.e., incomplete combustion, wall heat transfer, exhaust emission, and external work. It can be seen that the combustion efficiency of the CDC, HCCI, and RCCI all gradually decreases with the delay of the combustion phase. In comparison, the CDC has the lowest hydrocarbon and CO emissions and the highest combustion efficiency. On the contrary, HCCI and RCCI have lower fuel concentration near the wall and in the gaps. The incomplete oxidation leads to higher hydrocarbon and CO emissions and lower combustion efficiency. Besides, because of the equivalent ratio and fuel active stratification in the RCCI cylinder, its combustion efficiency is lower than HCCI.

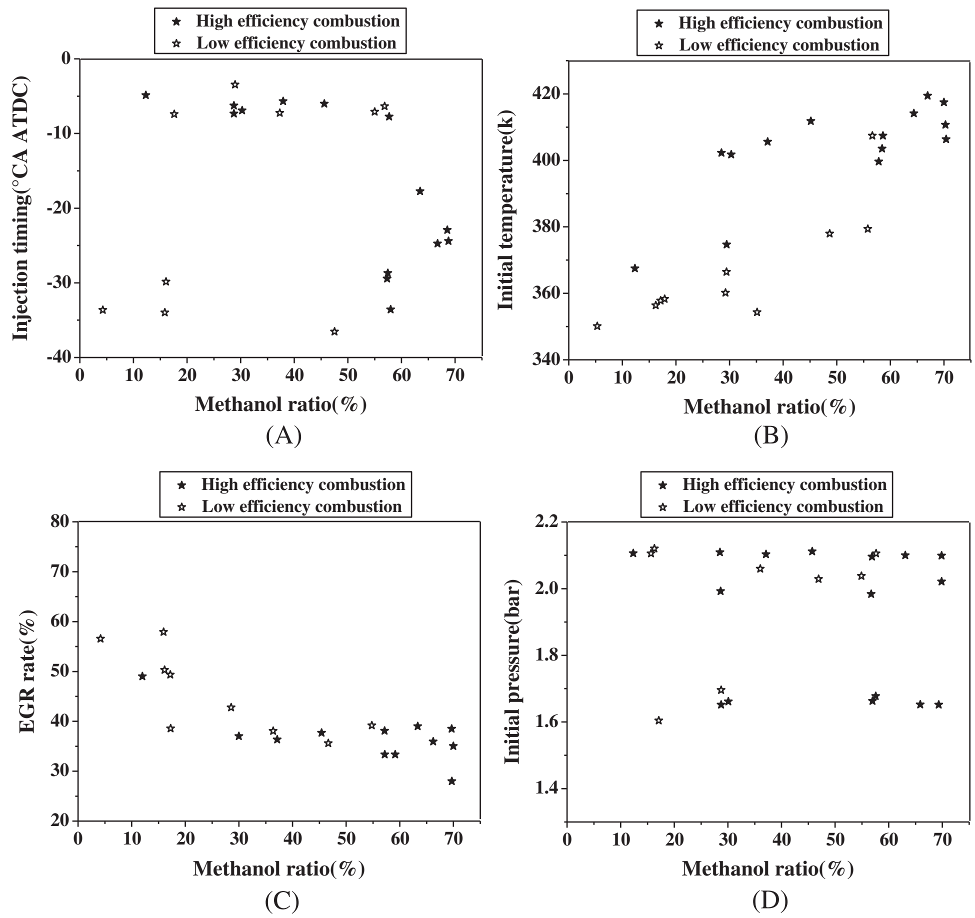

Figure 3: Distribution of methanol ratio, injection timing, initial temperature, EGR rate, and initial pressure for all optimization studies (A: Distribution of methanol ratio and injection timing for all optimization examples; B: Distribution of methanol ratio and initial temperature for all optimization examples; C: Distribution of methanol ratio and EGR rate of all optimization examples; D: Distribution of methanol ratio and initial pressure of all optimization examples)

The comparison results of the exergy distribution of the three combustion modes of CDC, HCCI, and RCCI are shown in Tab. 2. It can be seen that compared with the energy distribution comparison results; the exergy distribution also includes exergy losses. Among them, the proportion of incomplete combustion, wall heat transfer, exhaust emissions and external work in the exergy distribution is lower than that in the energy distribution. The exergy value in fuel is higher than the low heating value, and all parts of the energy will be lost during the transfer process. Therefore, the exergy values of incomplete combustion, wall heat transfer, exhaust emissions, and external work are all lower than the energy value. In other words, the proportion of each part in the second law of thermodynamics is lower than the proportion of the corresponding part in the first law of thermodynamics, but the change law of each part is consistent in the second law of thermodynamics and the first law of thermodynamics. The exergy/energy of the heat transfer on the wall of the CDC and HCCI is very close, which is higher than that of RCCI, while the exergy/energy of HCCI and RCCI exhaust emissions is very close and is lower than that of CDC. At different CA50 moments, the combustion temperature and boundary layer temperature of HCCI both change significantly, while the combustion temperature and boundary layer temperature of RCCI do not change much. At the same CA50 moment, the exergy losses of HCCI and RCCI are lower than those of the CDC. Therefore, HCCI and RCCI have greater potential in improving energy utilization efficiency, and the advantages of RCCI are more obvious.

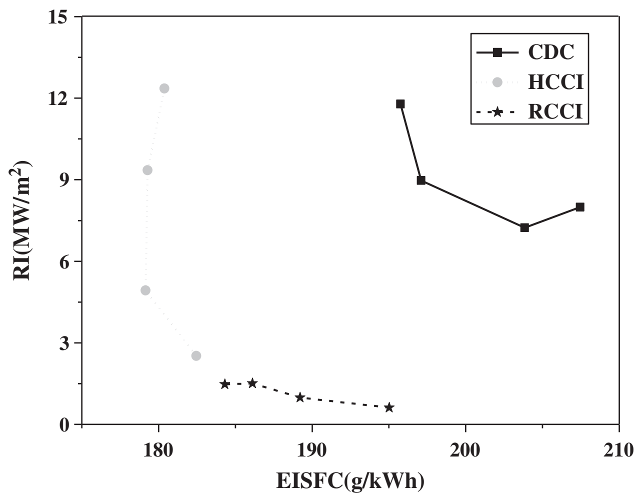

Figure 4: Relationship between EISFC and RI in three combustion modes of CDC, HCCI, and RCCI

Table 1: Comparison of the energy distribution of three combustion modes of CDC, HCCI, and RCCI (incomplete combustion; wall heat transfer; exhaust emissions; external work)

Combustion modes	CA50 = 3	CA50 = 5	CA50 = 7	CA50 = 9
CDC	5: 17: 36: 42	5: 16: 37: 42	6: 15: 38: 41	6: 15: 39: 40
HCCI	7: 19: 29: 45	7: 18: 30: 45	8: 17: 30: 45	9: 16: 31: 44
RCCI	11: 13: 32: 44	12: 11: 33: 44	13: 10: 33: 44	15: 10: 33: 42

Table 2: Comparison results of exergy distribution in three combustion modes of CDC, HCCI, and RCCI (incomplete combustion; wall heat transfer; exhaust emissions; external work; exergy loss)

Combustion modes	CA50 = 3	CA50 = 5	CA50 = 7	CA50 = 9
CDC	5:11:19: 39: 26	5:10 AM: 20: 39: 26	6:10 AM: 20: 38: 26	6:10 AM: 20: 38: 26
HCCI	7:11:15 AM: 42: 25	7:12:14 AM: 42: 25	7:11:14 AM: 42: 26	7:12:13 AM: 42: 26
RCCI	11:8:16: 41: 24	11:7:17 AM: 41: 24	12:7:17 AM: 40: 24	14:6:17 AM: 39: 24

The first and second laws of thermodynamics are used to study the energy and exergy distribution of the three combustion modes of CDC, HCCI, and RCCI. According to the above numerical calculation results, the influence of the combustion mode and fuel type on the fuel economy of the engine, as well as the key factors affecting the crowd loss, are well understood. The results show that although the exergy proportion score of each part of the three combustion modes is lower than the energy fraction, the change rules of exergy and energy parts remain the same. When different fuels are used on RCCI engines, different operating parameters need to be matched, which results in different combustion temperatures and fuel distribution in the cylinder, causing different proportions of congestion losses. To effectively reduce the crowding loss, a higher proportion of premixed fuel and earlier injection timing should be used, while the equivalent ratio in the cylinder should be controlled within the stoichiometric ratio, the combustion temperature of the fuel should be increased, and the fuel with lower thorium loss should be selected.

4 Conclusions

To solve the contradiction between supply and demand caused by a large amount of oil used in traditional internal combustion engines, a CFD numerical model is established. Based on the computer-optimized NSGA-II algorithm, a novel clean and efficient combustion mode is proposed. Five important parameters of the RCCI engine are optimized, including methanol ratio, injection time, initial temperature, EGR rate, and initial pressure. Meanwhile, the CDC, HCCI, and RCCI combustion modes are analyzed thermodynamically. Through research, the methanol/diesel RCCI engine has great potential in improving energy utilization efficiency, which realizes a clean and efficient combustion process and improves fuel economy while meeting strict emission regulations. Therefore, methanol/diesel RCCI is a very promising clean and efficient combustion mode, which will be widely used in the near future. However, there are still some shortcomings in the research. Although the load reached by RCCI has been greatly improved, it cannot achieve full load operation. Therefore, in the subsequent study, the load will be widened and its practicality will be enhanced.

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