INTRODUCTION

Gasoline compression ignition combustion has demonstrated the potential of getting high fuel efficiency. By using high octane fuel with a longer ignition delay, GCI can get the start of combustion and the injection basically separated compared with conventional diesel combustion. Therefore, GCI mode can achieve relatively low...
soot emission. However, at high load, the soot emission still deteriorates because of the fuel-stratification control combustion.

The low-temperature combustion mode of GCI has the potential to achieve clean and efficient combustion in a wider range of engine operating conditions. However, there are still some problems to be solved urgently, including the problem of stable combustion at low load, the problem of over-high pressure caused by excessive heat release rate at high load, and the deterioration of soot emission caused by incomplete separation of fuel injection and combustion processes.

Researchers have carried out a series of experimental and computational studies on GCI. Goyal et al. studied the ignition process of GCI combustion in a small-bore optical engine. The results show that the ignition process of single injection comprises multiple auto-ignition kernel development from which the isolated flame growth occurs. Jiang et al. found that high efficiency under all loads could be achieved indeed by choosing proper intake pressure, intake temperature, and fuels. Wei et al. studied the knock phenomenon of GCI combustion and found that an increasing trend of both brake mean effective pressure (BMEP) and knock intensity as injection timing advances. Kim et al. compared double-injection with single injection and studied the effects of the pilot-injection timing and the pilot-injection quantity on GCI combustion under low load condition. Kalghatgi et al. carried out GCI experiments on a single-cylinder engine with a compression ratio of 14 and a displacement of 2 L. The results show that the indicated fuel consumption rate (ISFC) can reach 178 g/kW·h, and the soot is less than 0.4 FSN under the single injection of gasoline (RON 94.7) near the top dead center and the IMEP of 14.86 bar. Hildingsson et al. studied the effects of fuel octane number on the combustion characteristics of GCI. Results reveal that at low load, higher octane number is conducive to reducing NOx emission; at high load, a relatively low octane number fuel should be considered to promote combustion, and exhaust gas recirculation (EGR) should also be used to reduce excessive pressure rise rate. It is also pointed out that the reasonable range of octane number for partial premixed compression ignition of gasoline is 75-85.

In order to reveal the combustion process and emission mechanism in depth, numerical simulation has been more and more widely applied in GCI research, as it can reveal the microscopic information which cannot be obtained in the test process and provide guidance for the experimental research. Liu et al. simulated the effects of different premixing ratios on the ignition delay of GCI. It was considered that higher premixed ratios and advanced injection timing can not only help to prolong the ignition delay, but also lead to high combustion rate. In order to control the pressure rise rate, the injection timing should be maintained not earlier than—8 °CA ATDC, and the premixed ratio should be less than 50%. Kim et al. used a detailed soot model to study the effects of injection pressure on the mixing and evaporation process of fuel. The results show that increasing injection pressure can promote atomization, the number density, and mass of soot decrease, and the soot particle size presents a trend of single peak small particle size.

Previous studies reveal that mixture stratification in the cylinder can significantly affect soot emission and pressure rise rate, and compound injection strategy is an effective way to control mixture stratification, pressure rise rate at high load operation, and soot emission. According to different injection strategies, Adam et al. divided the mixture into three categories: partial fuel stratification (PFS), moderate fuel stratification (MFS), and heavy fuel stratification (HFS). Through the simulation analysis of different mixing modes at the same CA50 time, it is considered that there is a compromise between the mixture stratification and combustion efficiency. Although great progress has been made in the research of GCI in recent years, the effects of mixture stratification on the combustion process of GCI and the mechanism of fuel injection strategy on soot formation still need to be further studied, especially at high load. Therefore, in this work, a numerical study was conducted to explore the combustion characteristics and soot formation evolution processes within GCI combustion at high load. Firstly, a simulation model is established based on the diagnosis results of optical engine, and the effect of mixture stratification on GCI combustion and emission was investigated. Secondly, according to an actual engine, we establish an engine model to explore the influence of injection strategy on GCI combustion and soot emissions under high load condition.

2 MODELING AND VALIDATION

2.1 Optical engine modeling and validation

Accurate model is a prerequisite for providing fundamental knowledge of soot formation of GCI combustion. Therefore, an optical engine model is established in commercial software Converge 2.3 based on an optical engine to explore the influence of mixture stratification on soot formation of GCI combustion. The optical engine is modified from a four-cylinder diesel engine. Details of the engine can be obtained from reference Figure 1 and Table 1. Considering the structural strength of quartz glass, the compression ratio is modified to 11. In order to reduce the pollution of continuous fuel injection to the
quartz glass window, the fuel injection process is carried out every 20 engine cycles in the test process, and the single test is usually within half an hour with timely cleaning. The experiment of optical engine is kept at a constant operation, that is, rotating speed 1200 r min⁻¹ and circulating oil 26 mg cycle⁻¹ (IMEP 5.99 bar, −25 °CA ATDC). EGR is not used in the test, so the EGR rate is 0.

Figure 2 shows the established simulation model. According to the results of mesh sensitivity analysis, 2 mm basic mesh, twice adaptive mesh encryption, and fixed mesh encryption at injector position are used. In addition, the simplified kinetic mechanism of TRF/PAH developed by Wang et al²² is adopted in this simulation, which consists of 109 species and 543 reactions and has been fully validated. Other models used in this work are listed in Table 2.

Figure 3 shows the cylinder pressure and heat release rate between experiment and simulation. The simulation results can reasonably reproduce the combustion heat release process. The peak value of heat release rate is slightly higher than the experimental value, but it is still within the acceptable range. In addition, Figure 4 compares the experimental and simulated spray processes. A reasonable prediction for spray development process in the cylinder was achieved at different crank angle. At −17.1 °CA ATDC, there is a phenomenon of spray wall impingement, which

TABLE 1 Optical engine specifications and operation conditions

| Specifications          | Values |
|------------------------|--------|
| Stroke                 | 100 mm |
| Displacement           | 0.664 L|
| Connecting rod length  | 155 mm |
| Compression ratio      | 11     |
| Combustion chamber diameter | 63 mm |
| Common rail pressure   | 600 bar|
| Holes number of injector | 6     |
| Spray included angle   | 150°   |
| Injector hole diameter | 0.15 mm|
| Speed/(r min⁻¹)        | 1200   |
| Injection mode         | Single injection |
| EGR/%                  | 0%     |
| Fuel                   | PRF70  |
| Injection advance angle/(°CA ATDC) | −25/−60/−90 |
| Fuel/(mg cycle⁻¹)      | 26     |
| Injection pressure/MPa | 60     |
| Intake pressure/KPa    | 100    |
| Intake temperature/K   | 398    |

TABLE 2 Models and mechanism in simulation

| Models and mechanism                | Values           |
|-------------------------------------|------------------|
| Turbulence                          | RNG k-ε²⁵        |
| Spray breakup                       | KH-R²⁶,²⁷        |
| Turbulent diffusion model           | O’Rourke²⁸       |
| Drop/wall interaction               | Wall-film²⁷      |
| Chemistry modeling                  | SAGE             |
| Chemical mechanism                  | TRF/PAH²²        |
is also reflected in the simulation. Figure 5 is an image of combustion process taken from the experiment in a cycle in the cylinder. In the simulation, the development of temperature in cylinder is compared with it. The temperature distribution in the simulated image shows that there are several self-ignition points on one side of the fuel wall, and then, the flame spreads to the whole combustion chamber quickly, which is basically consistent with the combustion process in the experiment, as shown in Figure 5A. In general, the model can reasonably capture the combustion process in the cylinder of GCI, which indicates it can be used to explore the effect of physics and chemistry caused by mixture stratification on GCI combustion.

### 2.2 Single-cylinder engine modeling and validation

A GCI engine model based on a real single-cylinder engine was set up to explore the impact of control strategies on soot, NOx emissions, and maximum pressure rise rate. The experiments were conducted on a single-cylinder engine, which is modified from a six-cylinder heavy-duty diesel engine. The sixth cylinder was separated from the other five for test purposes. This individual cylinder was equipped with independent high-pressure common rail fuel injection system, intake temperature, pressure regulating systems, and an EGR system. The cycle fuel quantity of metal engine is 91 mg cycle⁻¹, and the corresponding IMEP is 16.02 bar. As it is in-cylinder direct injection, the switching of engine load is mainly realized by controlling the rotating speed and circulating fuel quantity. It is also equipped with an external air compressor to boost the intake air and an intake air heater to control the intake air temperature. Relevant models adopted in this simulation have been listed in Table 2. The engine specifications are listed in Table 3, and more details about the modified engine can be obtained in reference23 and Figure 6. It should be noted that in order to obtain better simulation of soot emission, the model is coupled with the multi-step phenomenological soot model (Gokul model) proposed by Vishwanathan et al.24 The Gokul model takes the A4 as the precursor of soot. A 45° sector mesh (fixed embedding combined adaptive mesh refinement) was adopted to improve the computational efficiency, as illustrated in Figure 7. The base grid size was set to be 2 mm after the grid sensitivity analysis, with two times adaptive mesh refinement under certain operations. The cells of injector...
and boundary region were embedded, and the maximum total number of grids exceeds 300,000.

For validation, three-component gasoline substitutes TRF (16.3% n-heptane, 31% toluene, and 52.7% iso-octane) were used to simulate the 92# gasoline. The comparison of physicochemical properties between gasoline and the gasoline surrogate is shown in Table 4. The mixture stratification in the cylinder has a significant impact on the ignition and combustion process of GCI, and the injection strategy can effectively improve the state of the mixture in the cylinder. Therefore, the port fuel injection strategy combined with pre and main direct injection in cylinder is adopted in this work. The operating conditions of simulations set according to the test conditions are listed in Table 5.

Figure 8 shows the experimental and simulated cylinder pressure and heat release rate under the selected...
operation for validation. Results show that the simulated cylinder pressure is in good agreement with the test results, and simulated heat release rate can also reasonably reflect the characteristics of two-stage heat release of premixing and diffusion. Table 6 shows the experiment and simulation results of combustion and emission characteristics. The simulated CA10, CA50, soot emission, and combustion efficiency are basically consistent with experiment, while the NOx and maximum pressure rise rate are slightly lower than the test results. On the whole, the model can reasonably reveal the actual combustion and emissions of the engine, which indicates it can be used for the subsequent injection strategy study of GCI combustion.

### 3 | RESULT AND DISCUSSION

#### 3.1 | Results and discussion of optical engine calculation

The advanced injection timing can make the combustion closer to HCCI mode. Therefore, for thoroughly analyzing the effect of mixture stratification on GCI combustion,
FIGURE 11  Emissions and MPRR for different injection timings

| Injection timing | Injection timing | Injection timing |
|------------------|------------------|------------------|
| -90°CA ATDC      | -60°CA ATDC      | -25°CA ATDC      |

**Mixture stratification**

Equivalence ratio

- 2.0e+1000
- 1.5e+000
- 1.0e+000
- 5.0e-001
- 0.0e+000

**CH$_2$O**

- 1.0e-005
- 7.5e-006
- 5.0e-006
- 2.5e-006
- 0.0e+000

**OH**

- 6.0e-003
- 4.5e-003
- 3.0e-003
- 1.5e-003
- 0.0e+000

**C$_2$H$_2$**

- 1.0e-004
- 7.5e-005
- 5.0e-005
- 2.5e-005
- 0.0e+000

**Temperature (K)**

- 2000.0
- 1700.0
- 1400.0
- 1100.0
- 800.0

FIGURE 12  Species and equivalence distribution of different inject timing at CA10
three cases with different injection timings (−90, −60, and −25 °CA ATDC) are set to form different degrees of mixture stratification in the cylinder. Detailed operation settings have been listed in Table 1.

Figures 9-11 show the results of IMEP, temperature, combustion phase, emission etc at different injection timings. When the injection timing is −90 °CA ATDC, the mixture in the cylinder is well distributed. Consequently, the engine works much closer to HCCI low-temperature combustion mode and the average combustion temperature is the lowest among the three cases, which can be confirmed in Figure 9B. However, due to the low combustion temperature, the THC emission is relatively high. Moreover, because the combustion phase is closest to the TDC (CA50 = 11.8 °CA ATDC) for injection timing −90 °CA ATDC, the heat release process is more concentrated, resulting in the highest in-cylinder pressure and PRR. Compared with −90 °CA ATDC, combustion with injection timing of −60 °CA ATDC has relatively worse performance on pressure, heat release rate. This is mainly because the combustion phase (CA50 = 17.3 °CA ATDC) is the most delayed. Therefore, the in-cylinder pressure and PRR are significantly reduced compared with that of injection timing −90 °CA ATDC. In addition, its combustion efficiency is also at a low level. Consequently, part of the fuel failed to completely oxidize to CO2, but partially oxidize to CO, which explains high CO emission in Figure 11. Different from the above two cases, the injection timing of −25 °CA ATDC has better performance on combustion and emission. Consequently, from above discussion, it can be deduced that mixture stratification in the cylinder has an important impact on combustion and emissions, which deserves to be further explored.

Figures 12 and 13 show important species and equivalence distribution of CA10 and CA50 at different inject timing. C2H2O is used to characterize the low-temperature

| Injection timing | Injection timing | Injection timing |
|------------------|------------------|------------------|
| -90°CA ATDC      | -60°CA ATDC      | -25°CA ATDC      |

![Mixture stratification](image1)

![Low temperature reaction](image2)

![High temperature reaction](image3)

![OH](image4)

![C2H2](image5)

![Temperature (K)](image6)
reaction, and OH is used to characterize the combustion at high temperature. \( \text{C}_2\text{H}_2 \) is an important component in the formation and growth of soot precursors, which is also considered in analysis.

Under the condition of \(-90^\circ\text{CA ATDC}\) injection timing, the distribution of equivalence ratio of mixture in cylinder tends to be homogeneous both at CA10 and CA50. At CA10 of \(-90^\circ\text{CA ATDC}\) injection timing, \( \text{CH}_2\text{O} \) in the middle of the combustion chamber has been partially consumed, and at the same time, a large amount of OH is generated. The ignition process is closer to the way of HCCI, which results in higher pressure rise rate and relatively large high-temperature area in whole cylinder. Although more \( \text{C}_2\text{H}_2 \) is generated at CA10 than the other two cases, it is basically oxidized at CA50.

Different from injection timing of \(-90^\circ\text{CA ATDC}\), at CA10 of \(-25^\circ\text{CA ATDC}\) injection timing, there is a significant concentration stratification of the mixture in the cylinder, and there is still strong mixture stratification in the cylinder until CA50. The concentration stratification makes the combustion process of GCI generate a significant chemical interaction (high reactive small molecules and low reactive isoctane), which has been circled in Figures 12 and 13, through most of combustion process. As can be seen from the Figure 12, the ignition begins at one side of the combustion chamber. In addition, the area of high-temperature zone is obviously smaller than that at CA10 of \(-90^\circ\text{C ATDC}\) injection timing. The flame propagates from the ignition core to the whole combustion chamber, and its propagation speed is significantly lower. Therefore, it is the synergistic effect of physics and chemistry, induced by concentration stratification of the mixture that makes the pressure rise rate of GCI combustion process lower, combustion process more moderate, and controllable with an acceptable emission.

**Figure 14** In-cylinder pressure, heat release rate, and mean temperature under different EGR

**Figure 15** MPRR, NOx, and soot emissions and oxidation process of soot under different EGR
3.2 Results and discussion of single-cylinder engine calculation

The concentration stratification in cylinder can effectively improve the combustion process, reduce the soot emission, and control the pressure rise rate. In addition, EGR can significantly reduce the temperature in cylinder, thereupon then reducing NO\textsubscript{x} emissions. Therefore, the influence of port fuel injection combined with pre and main injection strategy and coupled with EGR on the soot emission of GCI is explored in the following study.

3.2.1 Effect of different EGR on soot emission

In this section, the EGR rate is increased from 0% to 40% under the conditions of 40% port injection ratio, 10% pre-injection, −8 °CA ATDC of the main injection timing, and 15 °CA of the pre and main injection interval. Figure 14 shows the in-cylinder pressure, heat release rate, and mean temperature under different EGR. It can be seen from the results that with the increase in EGR, the maximum pressure, and average temperature in the cylinder decreases significantly, the ignition delay period extends, the combustion phase moves backward. In addition, when the EGR is more than 30%, the total amount of combustion heat release is significantly reduced.

Figure 15A shows the maximum pressure rise rate, NO\textsubscript{x}, and soot emissions under different EGR. Results reveal that with the increase in EGR from 0% to 40%, the NO\textsubscript{x} emission is significantly reduced, but the soot emission is promoted, especially, when the EGR is greater than 20%. This can be explained by the decrease in oxygen concentration and the temperature with the increasing of EGR. In addition, with the increase in EGR from 0% to 20%, the MPRR first increased slightly. When the EGR continues to increase from 20% to 40%, the MPRR begins to decrease significantly, which is mainly due to the decrease in total heat release and combustion efficiency under the cooperation of high EGR. Figure 15B shows the overall process of soot formation and oxidation in the cylinder at different EGR. As shown in the results, due to the decrease in heat release and overall temperature in cylinder with the increasing EGR, the soot production shows a remarkable declining trend. However, in the subsequent stage of oxidation, due to the decrease in temperature and oxygen concentration in the cylinder, the oxidation of soot deteriorates with the increase in EGR. Therefore, although the soot formation at high EGR is relatively low, the ultimate soot emission is higher due to the deteriorating oxidation process.

A4 is the precursor of soot, and oxygen and OH radical are the critical components of the soot oxidation process. In addition, the temperature and equivalence ratio environment also have a crucial influence on soot formation and oxidation. Therefore, the in-cylinder temperature, equivalence ratio, and essential components affecting soot formation and oxidation of GCI under different EGR rates of 20 °CA ATDC are analyzed in Figure 16. As a typical radial of high-temperature reaction, OH is mainly distributed in the high-temperature area around the spray beam. At the center of the combustion chamber and the end of
the spray, near the wall, the equivalence ratio is relatively higher and the temperature is lower, and the distribution of OH keeps at a low level, while the concentration precursor A4 of soot is high. In addition, with the increase in EGR, the distribution of OH radical generally decreases, while that of A4 increases. A4 and OH are the essential components in the process of soot formation and oxidation, which also have an important influence on its nucleation rate, surface growth rate, and oxidation rate. The distribution of nucleation rate and surface growth rate is basically consistent with A4 and enhances with the increase in EGR. For the oxidation rate of soot, it is generally consistent with the distribution of OH and decreases with the increase in EGR. Hence, one can see that under the condition of high EGR rate, the oxygen concentration in cylinder decreases resulting in a decline in OH radical. Subsequently, the oxidation rate of soot is weakened under high EGR operation. It should be noted that at the selected crank angle (20 °CA ATDC), the soot mass and number at low EGR are almost close to the peak value, and the production of soot begins to decrease during the subsequent stage, while the soot mass and number at high EGR will continue to increase under high nucleation and surface growth rate. Therefore, considering the pressure rise rate, NOx, and soot emissions, 20% EGR is selected to further study the influence of the control parameters on GCI combustion.

3.2.2 | Effect of main injection timing on soot emission

The effect of main injection timing (second in-cylinder injection: −4 to −12 °CA ATDC) on GCI combustion is investigated under the condition of 20% EGR, 40% port fuel injection (port manifold injection), 10% preinjection (first in-cylinder injection), and 15 °CA injection interval. Figure 17 shows in-cylinder pressure, heat release rate, and mean temperature under different injection timings.

![Figure 17](image17.png)

**Figure 17** In-cylinder pressure, heat release rate, and mean temperature under different injection timings

![Figure 18](image18.png)

**Figure 18** Maximum pressure rise rate, NOx, and soot emissions and oxidation process of soot under different injection timings
and mean temperature under different injection timings. It can be seen from the results that with the advance of the main injection timing, the heat release of premixed combustion increases significantly, and the heat release of diffuse combustion decreases. Although the combustion starting point moves backward slightly, the overall combustion duration shortens, and the maximum pressure in the cylinder also increases. This is because when the main injection timing is advanced, as the temperature and pressure conditions in the cylinder cannot meet the critical conditions of ignition, the fuel injected into the cylinder can be more fully mixed with the surrounding premixed fuel. Therefore, the equivalence ratio of premixed fuel increases, which weakens the stratification of the mixture in cylinder to some extent, delays the start of combustion, and enhances the premixed combustion heat release.

Figure 18A shows MPRR, NO$_x$, and soot emissions under different injection timings. As shown in the figure, with the advance of main injection timing, the MPRR and NO$_x$ emission increase, and when the main injection timing is earlier than $-8^\circ$CA ATDC, the MPRR will exceed 1.3 MPa ($^\circ$CA)$^{-1}$, which will lead to large combustion noise and mechanical load. For soot emission, it first decreases and then increases slightly with the advance of main injection timing and hits bottom at main injection timing of $-10^\circ$CA ATDC of, but the MPRR and NO$_x$ are higher at this main injection timing. Figure 18B shows the overall oxidation process of soot under different injection timings. Results reveal that with the advance of the main injection timing, the initial time of soot generation is advanced, and the peak value of soot is increased. This may be due to the increase in premixed combustion heat release and the rapid temperature rise in the cylinder at the earlier main injection timing. However, the ultimate soot emission has been reduced. Therefore, it can be concluded that the advance of the main injection timing not only promotes the formation of soot, but also promotes its oxidation process significantly, which makes the ultimate soot emission decrease with the advance of main injection timing within a certain range.

In order to further explore the influence of the main injection timing on the soot formation and oxidation process, the in-cylinder temperature, equivalence ratio, critical components affecting the soot formation, and oxidation under different main injection timings of 20 $^\circ$CA ATDC are compared and analyzed, which are shown in Figure 19. It can be concluded that along with the advance of main injection timing, the overall temperature of the spray increases significantly, and the near-wall equivalence ratio at the end of the spray decreases. In addition, it can be found that the distribution region of A4 is basically similar to that of equivalence ratio, and

![Figure 19](image_url)

**FIGURE 19** In-cylinder temperature, equivalence ratio, OH, A4, nucleation rate, surface growth rate, oxidation rate, soot mass, and soot number under different main injection timings at 20 $^\circ$CA ATDC.
However, the oxidation rate of soot strengthens significantly with the advance of main injection timings. In addition, under the main injection timing of $-10 \, ^\circ\text{CA ATDC}$, the high oxidation rate is not only in the region with more OH radicals around the spray beam, but also extends to the region with high temperature inside the spray beam. Therefore, it can be concluded that the effect of temperature on the soot oxidation process is the dominant mechanism for the ultimate emission of soot at different main injection timings. It should be noted that the soot mass is still in increasing trend at the selected crank angle 20 $^\circ\text{CA ATDC}$ in the case of $-4 \, ^\circ\text{CA ATDC}$ main injection timing, and the soot mass and number will further increase under the high soot nucleation rate. Therefore, considering the pressure rise rate, NOX, and soot emissions, $-8 \, ^\circ\text{CA ATDC}$ main injection timing with 20% EGR is selected to further study the influence of injection intervals on GCI combustion.

### 3.2.3 Effect of injection interval on soot emission

The effect of injection intervals on GCI combustion is investigated under the condition of 20% EGR, main injection timing ($-8 \, ^\circ\text{CA ATDC}$, fuel injection quantity: 50%), 40% port manifold injection (fuel injection quantity: 40%), and 10% pre fuel injection (injection timing: $-33$, $-23$, $-13 \, ^\circ\text{CA ATDC}$, fuel injection quantity: 10%). Figure 20 shows in-cylinder pressure, heat release rate, and mean temperature under different injection intervals. It can be seen from the results that with the advance of the preinjection timing, the preinjection fuel can obtain better mix with the surrounding port injection fuel before the temperature and pressure in the cylinder reach the ignition condition, which weakens the mixture stratification. Therefore, the total premixed heat release is increased, and combustion phase and peak heat release shift back slightly.
Figure 21A shows the MPRR, NOx, and soot emissions under different injection intervals. Results reveal that with the advance of the preinjection timing, NOx, and soot emissions are reduced, and the MPRR hardly changes, which keeps approximately at 1.2 MPa (°CA)$^{-1}$. Figure 21B shows the overall oxidation process of soot under different injection intervals. With the advance of preinjection timing, the peak value of soot formation declines, and the ultimate soot emission also decreases, especially under the preinjection timing of $-33$ °CA ATDC. Generally, compared with EGR and main injection timing, the effect of injection interval on combustion is relatively less.

Furthermore, the in-cylinder temperature, equivalence ratio, critical components affecting the soot formation, and oxidation under different injection intervals are shown in Figure 22. At 20 °CA ATDC, the soot formation has reached the maximum value at different preinjection timings, and the discrepancy of temperature distribution in the cylinder is small. For the equivalence ratio, the mixture stratification in the cylinder is weakened with advanced preinjection timing and the near-wall equivalence ratio at the end of the spray beam is reduced. For the distribution of essential components, with the advance of preinjection timing, the distribution of OH radical shows little discrepancy, while the soot precursor of A4 reduces slightly. In addition, the nucleation rate and surface growth rate of soot decrease with the advance of preinjection time, while the oxidation rate increases slightly, which leads to the declining of soot mass and number. Generally, properly increasing the interval between the pre and main injection can reduce NOx and soot emissions under the premise of ensuring that the MPRR is within the upper limit. However, the effect is not significant, which is mainly because the proportion of preinjection fuel is small during the GCI combustion.

Overall, 40% port injection, $-8$ °CA ATDC main injection timing and $-23$ °CA ATDC preinjection timing, 20% EGR can keep the pressure rise rate lower than the limit, with the soot emission 0.0209 g/(kW-h), the NOx emission 3.213 g/(kW-h), and the indicated thermal efficiency 46.798%.

4 | CONCLUSION

The main conclusions are as follows:

1. Mixture stratification is the dominant control mechanism of GCI combustion, which can significantly affect the MPRR, NOx, and soot emissions.

2. Proper amount EGR inducted can effectively improve the MPRR and NOx and soot emission of GCI combustion. Applying higher EGR can significantly reduce the OH radical and weaken the soot oxidation process in the later stage, resulting in the increase in soot emission.

3. Under the premise of reasonable NOx emission and MPRR, the soot emission can be effectively reduced with a proper advanced main injection timing. The effect of temperature on the soot oxidation process is the primary mechanism. The overall temperature in the cylinder will be increased with earlier main injection timing, which strengthens the subsequent soot oxidation process.

4. Properly increasing the interval between the pre and main injection can reduce NOx and soot emissions. However, the effect is not significant, which is mainly because the proportion of preinjection fuel is small during the GCI combustion under the selected operations.
The injection strategy of 40% port injection, −8 °CA ATDC main injection timing, and −23 °CA ATDC preinjection timing, 20% EGR can keep the pressure rise rate lower than the limit with the soot emission 0.0209 g/(kW·h), the NOx emission 3.213 g/(kW·h), and the indicated thermal efficiency 46.798%.

ACKNOWLEDGMENT
The authors would like to acknowledge the financial support provided by the National Natural Science Foundation of China (NSFC) through its projects of 51976134 and 51876140.

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How to cite this article: Zhang L, Wang H, Zhong X, et al. Study on the influence mechanism of mixture stratification on GCI combustion and the compound injection strategy under high load operation. Energy Sci Eng. 2021;9:2434–2448. https://doi.org/10.1002/ese3.997