Effect of Exhaust Gas Recirculation and Spark Timing on Combustion and Emission Performance of an Oxygen-Enriched Gasoline Engine

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ABSTRACT: Oxygen-enriched combustion (OEC) technology in SI engines can greatly improve the degree of constant volume combustion, increase the torque output, and reduce HC and CO emissions but lead to a sharp increase in NO\textsubscript{x} emissions. Simultaneously, the high temperature from OEC would lead to high nucleation particle emissions. Under the OEC mode, except the oxygen content, spark timing and engine load are important influencing factors on emissions. Exhaust gas recirculation (EGR) technology has been proven to reduce NO\textsubscript{x} emissions effectively. This research investigates the effects of EGR on combustion and emission performance under an oxygen-enriched ratio (OER) of 25% with five EGR ratios (0–20%) for the initial throttle opening of 14% (at an EGR ratio of 0%) with an engine speed of 1500 rpm. The study shows that when the OER is 25%, the output torque increases with the increase of the EGR ratio. At the proper spark timing, the EGR ratio over 15% can obtain lower NO\textsubscript{x} emissions and particle emissions than the baseline (OER of 21%). Although HC emissions increase with the EGR ratio, they are still lower than the baseline. Overall, the OER of 25% coupled with the EGR ratios of 15–20% is the predominant combustion mode to improve power and emission performance in SI engines.

1. INTRODUCTION

With the gradual depletion of petroleum resources and the intensification of environmental pollution, how to achieve simultaneously strong power output and low exhaust emissions has become particularly necessary in internal combustion engines.\(^1\) Sufficient oxygen in the cylinder mixture has always been the key factor to achieving clean combustion in engines, and it is also the key reaction substance to fully convert the chemical energy of the fuel into kinetic energy.\(^2\)\(^,\)\(^3\) In many studies on engine clean combustion technology, renewable oxyfuels, such as methanol, ethanol, and butanol, were the most prominent.\(^4\)\(^,\)\(^5\) Their oxygen atoms contained in the molecular structure can help to increase the chain-reacting rate and improve the microscopic combustion state of the mixture. Compared with traditional alkane fuels, oxyfuels have cleaner combustion products.\(^2\)\(^,\)\(^6\) Among the common oxyfuels, ethanol is the most widely used commercial fuel because of its lower toxicity, lower production cost, better evaporation characteristics, and superior combustion characteristic.\(^5\)\(^,\)\(^7\) However, the large heat of vaporization and low vapor pressure endow the engine with poor cold start capability in cold weather.\(^8\) The dependence on crops makes them unable to be fully promoted in some food-stressed countries and regions.\(^9\) Therefore, this paper aimed to explore another method to improve the oxygen content of the in-cylinder mixture, that is, the intake oxygen enrichment technology. In this technology, the oxygen-enriched air is drawn into the cylinder to increase the oxygen content in the mixture, to achieve clean combustion and strong power output.\(^10\)\(^−\)\(^12\)

The research on oxygen-enriched combustion (OEC) engines began in the 1960s.\(^13\)\(^,\)\(^14\) Wartinbee et al. studied the oxygen-enriched air to reduce engine exhaust emissions. They found that HC and CO emissions were reduced, but NO\textsubscript{x} emissions increased significantly.\(^14\) Detuncq et al. found that compared with the naturally aspirated engine, OEC significantly increased the specific power, pressure rise rate, and exhaust temperature in a single-cylinder natural gas SI engine.\(^15\) Kajitani et al. used high-speed optical infrared numerical photography technology to study the specific situation of an SI OEC engine and observed that high-temperature thermal radiation from the reaction area ran...
through the entire combustion process. They also found that the OEC can reduce the combustion duration period and increase the exhaust temperature.\textsuperscript{16} Sekar et al. tested the intake oxygen-enriched air with an oxygen-enriched ratio (OER) of 25 and 28\% to explore the power performance and emission benefits. The results showed that 1,3-butadiene, formaldehyde, and acetaldehyde in exhaust emissions were reduced obviously under the standard Federal Test Procedure emissions test cycle.\textsuperscript{17}

Promoting the further application of OEC technology in automotive engines is the membrane oxygen enrichment technology. Since the 1990s, membrane oxygen enrichment technology has been developed. The technical principle is that due to the different permeability of N\textsubscript{2} and O\textsubscript{2} in the air when passing through the exchange membrane, where there is a pressure difference, O\textsubscript{2} will preferentially pass through the exchange membrane to generate oxygen-enriched air.\textsuperscript{18} Kashmir first applied the gas membrane separation technology to the intake system in a diesel engine. The highest oxygen concentration in the experiment was 35\%.\textsuperscript{19} The Argonne National Laboratory has carried out CFD simulation and membrane oxygen enrichment experiments on a single-cylinder gasoline engine under cold start conditions. Experimental results showed that OEC had positive effects on gasoline engine emissions under cold start conditions. When the OER was 23\%, HC emissions decreased by 25\% and CO decreased by 40\%, but the NO\textsubscript{x} emissions increased by 56\%.\textsuperscript{20,21}

Recently, researchers have focused many research on the OEC biomass-based diesel.\textsuperscript{22} Dinesha et al. and Vaiyapuri et al. have found that biodiesel fuels such as cardanol-methanol-diesel and methyl ester-diesel, when combined with intake oxygen-enriched combustion technology, provided very good improvements in HC, CO, and particle emissions.\textsuperscript{23} Studies on oxygen-enriched combustion for diesel and biodiesel agree that OEC technology can improve combustion performance and reduce HC, CO, particle emissions, and BSFC in engines. However, the literature on gasoline engines shows different findings. Ji and co-workers explored the effect of standard hydrogen--oxygen mixtures (HHO) on the combustion and emission performance of a SI engine in lean combustion. They found that the HHO reduced engine cyclic variation and improved combustion better than the same proportion of H\textsubscript{2} intake.\textsuperscript{24} More importantly, the results also showed that at a hydrogen--oxygen mixing ratio of 3\%, better particle emissions could only be achieved when a large proportion of hydrogen was mixed, while a high oxygen ratio led to a deterioration of both NO\textsubscript{x} and PN.\textsuperscript{25} The particle emission characteristics of gasoline engines with oxygen-enriched combustion are not the same as those of diesel engines, but unfortunately they have not been adequately analyzed from an oxygen-enriched combustion perspective in Ji’s study. There is still a lack of studies in the current reference literature that specifically address the effects of oxygen-enriched combustion on particle emissions from gasoline engines.

The most significant factor limiting the large-scale deployment of intake oxygen-enriched technology in the automotive sector is NO\textsubscript{x} emissions.\textsuperscript{26,27} For gasoline engines, there is also the potential for uncertain particle emissions. Over the past years, some researchers have considered diesel emulsification and intake humidification, expecting to use H\textsubscript{2}O as a trick to reduce NO\textsubscript{x} emissions to achieve advances on OEC diesel engines.\textsuperscript{28} However, the results have come at the cost of loss of power performance and the potential for erosion of the engine block. Other researchers investigated the role of exhaust gas recirculation (EGR) in reducing NO\textsubscript{x} emissions from oxygen-enriched diesel engines. Zhang et al. investigated the effect of oxygen-enriched combustion on the NO-smoke of a fully loaded EGR diesel engine at 1600 and 2200 rpm. The results showed that the combination of EGR and oxygen-enriched combustion resulted in a significant reduction in NO emissions, but an increase in particle emissions, along with an increase in BSFC.\textsuperscript{29} Perez et al. used CO\textsubscript{2} instead of EGR to study the variation of NO-smoke in oxygen-enriched diesel engines. The results showed that the best NO-smoke trade-off relationship can be achieved with 23\% O\textsubscript{2}–10\% CO\textsubscript{2}, but the advantages of oxygen-enriched combustion in CO, HC, and BSFC were completely canceled out.\textsuperscript{30} In conclusion, the existing literature on the reduction of NO\textsubscript{x} and particle emissions from oxygen-enriched engines is focused on diesel engines, and both H\textsubscript{2}O and inert gas are difficult to achieve excellent results in terms of both power and emissions performance. For oxygen-enriched gasoline engines, there is a lack of research not only on NO\textsubscript{x} emission reduction, but also on the variations of particle emissions. The parallel optimization of power performance with NO\textsubscript{x} and particle emissions is of paramount importance in the field of oxygen-enriched gasoline engines.

In this paper, it is considered that the uncertainty variation of engine particle emissions and the lack of solutions for poor NO\textsubscript{x} emissions in the oxygen-enriched combustion gasoline engines need to be further investigated. Because of the volume regulation of gasoline engines, adding EGR technology can have a different effect from that of oxygen-enriched diesel engines in the literature. Gasoline engines operate usually under small and medium loads under urban conditions. Hence the intake oxygen enrichment technology combining EGR technology to achieve parallel optimization of NO\textsubscript{x} particle emissions, and engine power performance rather than a trade-off relationship deserves to be investigated. In this paper, the purpose is to investigate whether intake oxygen-enriched combustion technology and EGR technology can complement each other and offset mutual disadvantages in gasoline engines.

Therefore, this paper innovatively combined EGR with OEC technology in a SI engine and deeply studied the impact of EGR technology on NO\textsubscript{x} emissions and PN emissions under the oxygen-enriched condition. The power performance and emission characteristics were researched by the SI engine test bench. This paper selected an engine speed of 1500 rpm and the throttle opening without EGR of 14\% as the initial engine operating condition. The OER in the cylinder was maintained at 25\%, and the EGR rate was set as 0\%, 5\%, 10\%, 15\%, and 20\%, respectively. As the important parameter for drastically affecting NO\textsubscript{x} emissions, the spark timing was chosen from 5\(^\circ\)CA BTDC to 25\(^\circ\)CA BTDC in 5 degree increments to explore the optimal operating conditions of the gasoline engine.

2. EXPERIMENTAL SETUP AND METHOD

2.1. Experimental Setup. Table 1 includes the main technical parameters of the test engine, which is equipped with combined injection systems of port injection (PI) and direct injection (DI). Figure 1 shows the combustion chamber’s sketch map. In this study, the gasoline was injected by the port fuel system and mixed with fresh air in the intake port, while pure oxygen was directly injected into the cylinder by the direct injector and then formed a certain proportion of oxygen-
enriched mixture in the cylinder to meet the experimental requirements.

Table 1. Specific Parameters of Test Engine

| specifications         | parameters                  |
|------------------------|-----------------------------|
| engine type            | four cylinders              |
| water cooling          | combined injection         |
| working volume         | 1984 mL                     |
| stroke                 | 92.8 mm                     |
| bore                   | 82.5 mm                     |
| compression ratio      | 9.6:1                       |
| ignition order         | 1-3-4-2                     |
| maximum power          | 147 kW                      |
| ODI pressure           | 8 MPa                       |
| EGR configuration      | low pressure                |
| PFI pressure           | 4–5 bar                     |
| maximum torque         | 325 N·m                     |

Figure 2 displays the sketch map of the test bench. Table 2 shows the main information of the experimental instruments. The intake pressure signal was measured by an intake pressure sensor with model APS-05041E, which was installed on the intake port. The fuel consumption signal was measured by the Ono Sokki DF-2420 volumetric fuel consumption meter. The excess air coefficient (λ) signal was collected with a λ sensor of ETAS LA4. The engine’s speed, power, and torque signal were measured by the CW160 eddy current dynamometer. A Kistler-2614B crank angle encoder recorded the crank angle signal, and an AVL-GU13Z-24 cylinder pressure sensor recorded the cylinder pressure signal. The Dewesoft combustion analyzer calculated the combustion data according to the crank angle signal and the cylinder pressure signal. The Horiba exhaust gas analyzer can detect the concentrations of CO, CO2, HC, NOx, and O2 in the exhaust gas. The particle emission data were measured by a particle collector DMS500.

The EGR system was designed to connect with the original engine intake port. According to Figure 2, a pipeline was drawn from the exhaust pipe of the engine. Passing through the EGR valve and intercooler, the exhaust gas was transported to the intake pipeline after a secondary voltage stabilizer, and the cold exhaust gas can enter the cylinder together with the fresh air.

2.2. Experimental Procedure. The experimental scheme and variables are shown in Table 4. The reasons for fixing OER of 25% in the study are as follows. Before this paper, we carried out a test of variable intake OER on the same test bench. As shown in Figures 3 and 4, we tested the OER from 21 to 25% in 1% increments. It was found that at an OER of 24−25%, NOx emissions have been beyond the testing range of exhaust gas analyzers, so a larger OER should not be pursued. Furthermore, we also found that the improvement of power performance was most obvious in the change from 21 to 22% and then gradually weakened. The maximum torque has been increased by 1.47 N·m from the OER of 21% to OER of 22%, but only 0.67 N·m from the OER of 24% to OER of 25%. Many previous studies on OEC technology have found that unlimited growth of the OER cannot increase the engine
In our preliminary test, the engine torque at an OER of 25% was optimal. Therefore, it is considered that the power growth would be little, and the NO\textsubscript{x} emissions increase rapidly when the OER exceeds 25% volume rate, the OER was fixed at 25% to explore the power performance and exhaust emissions coupled with EGR technology.

In this paper, by measuring the CO\textsubscript{2} concentration at the intake pipe and the exhaust pipe, the EGR rate entering the cylinder was calculated according to eq 1:

$$\text{EGR(\%)} = \frac{\text{CO}_{2\text{in}} - \text{CO}_{2\text{air}}}{\text{CO}_{2\text{out}} - \text{CO}_{2\text{air}}} \times 100\%$$

CO\textsubscript{2in} represents the CO\textsubscript{2} concentration (volume ratio) in the intake system, CO\textsubscript{2out} represents the CO\textsubscript{2} concentration in the exhaust system, and CO\textsubscript{2air} represents the CO\textsubscript{2} concentration in the air.

It is necessary to focus on the method to achieve an OER of 25%. As mentioned before, the oxygen was introduced into the cylinder by the DI system. During the experiment, the oxygen-enriched mixture under the stoichiometric ratio was used. The gasoline injection pulse width was fixed to ensure that the total input fuel amount remained unchanged when the OER increased. Through the calculation of the theoretical air-fuel ratio ($\lambda$), the precise control of the OER of 25% was realized. Specifically, when the oxygen volume ratio of the oxidant (air) is changed, no longer 21%, $\lambda$ must also change, and it is no longer 1. In this paper, eqs 2 and 3 were used to calculate the new $\lambda$. When the OER was 25%, the air–fuel ratio $\lambda$ of 1.18. By adjusting the value of $\lambda$ during the experiment, an OER of 25% can be achieved. After the oxygen content in the cylinder reached an OER of 25%, the excess O\textsubscript{2} increased the content of the working medium in the cylinder, which directly affected the change of the specific heat capacity of the mixture in the cylinder. However, after calculation, the maximum deviation was only 0.55%.

$$\text{O}_{\text{OER}} = \frac{32 \times \text{OER}}{32 \times \text{OER} + 28(1 - \text{OER})}$$

$$\lambda = \frac{\text{O}_{\text{OER}}}{\text{O}_{\text{OER=21\%}}}$$

where $\lambda$ stands for the excess air coefficient, and the OER stands for the average oxygen volume ratio of the oxidant (air).

This experiment adopted COV\textsubscript{IMEP} to describe the cycle-by-cycle variations. The definition of COV\textsubscript{IMEP} is shown in eq 4.56.
where $n$ represents the number of samples at each working point, and its value is 200 in this experiment.

Finally, it should be noted that the throttle opening of 14% mentioned in this paper, which only represents the case where OER of 25% and EGR rate of 0% (Table 3). Subsequently, as the EGR rate increased, the throttle opening would increase to ensure that the fresh air intake amount remained the same. The change in the test engine intake manifold pressure after adding the EGR is shown in Table 4. The intake manifold pressure increased by 4.73, 11.04, 14.60, and 20.80% than EGR ratio of 0%. The spark timing of minimum advance for best torque (MBT) was advanced slightly with the increase in EGR from 0 to 20%, 11.04, 14.60, and 20.80% than that of the initial condition (EGR ratio of 0%). With the throttle valve opening slightly decreasing, the pumping loss decreased, and the total intake of fresh air and exhaust gas increased. The increase in the number of in-cylinder working substances led to an increase in the output power of the power stroke.

The explanations for the MBT being advanced as the EGR ratio increased are as follows. In this study, the ODI timing of 300°CA BTDC was used to form cylinder oxygen enrichment, especially near the spark plug. When the ignition timing was delayed, the gas working medium near the spark plug was affected by the airflow movement entrained by the upward movement of the piston. The oxygen enrichment near the spark plug was destroyed and replaced by a relatively homogeneous mixture diluted by the exhaust gas. So that stable fire core and flame propagation were not easy to form. Especially when the EGR ratio was 20%, a higher exhaust gas dilution ratio would greatly prolong the ignition delay period and combustion duration of the mixture, and the flame propagation and combustion process would take longer. A too early ignition timing would result in greater compression loss. Therefore, 20% EGR was more sensitive to the changes in spark timing. Even in an oxygen-enriched atmosphere, the addition of EGR should be matched with the spark timing to achieve the best power performance.

In short, when the OER was 25%, the torque output increased gradually with the increase of EGR from 0 to 20%, and the torque output can be improved by up to 9.40%, which significantly improved the power performance.

| Table 3. Experimental Protocol and Experimental Variables |
|---------------------------------------------------------|
| parameters                                             | value                          |
| oxygen enriched ratio                                  | 25%                            |
| excess air factor                                       | 1.18                           |
| EGR ratio                                              | 0%, 5%, 10%, 15%, and 20%      |
| ignition timing (°CA BTDC)                             | 5, 10, 15, 20, 25              |
| engine testing working state                            | engine speed = 1500 rpm         |
|                                                         | throttle opening of 14%, 15.5%, 16.5%, 18%, and 19.5% |
| oxygen direct injection (ODI) timing                   | 300°CA BTDC                    |

3. RESULTS AND DISCUSSION

The test gasoline engine was operated at a constant speed of 1500 rpm and a fixed OER of 25%. The EGR ratio increased from 0 to 20% with each increment of 5%. The experimental results on pure air (OER of 21%) were set as the baseline. The combustion characteristics and emission performance in different EGR ratios and ignition timings were explored. The results are as follows.

3.1. Torque. Figure 5 demonstrates the engine torque’s variation with the spark timing under different EGR ratios. As the EGR rate increased, the maximum torque increased markedly. The maximum torque corresponding to the EGR ratio of 20% increased by 10.4% compared with that of the EGR ratio of 0%. The spark timing of minimum advance for best torque (MBT) was advanced slightly with the increase in the EGR rate. When the EGR rate was 20%, the torque was sensitive to the spark timing. Delaying the spark timing would seriously affect the torque.

The explanations for the torque improvement are as follows. As shown in Table 4, as the EGR rate increased, the throttle valve opening would increase accordingly. When the EGR rate was 5–20%, the intake manifold pressure increased by 4.73, 11.04, 14.60, and 20.80% than that of the initial condition (EGR ratio of 0%). With the throttle valve opening slightly increasing, the pumping loss decreased, and the total intake of fresh air and exhaust gas increased. The increase in the number of in-cylinder working substances led to an increase in the output power of the power stroke.

3.2. Cylinder Pressure, Pressure Rise Rate, and COV$_{\text{IMEP}}$. Figures 6 and 7 depict the variation of cylinder pressure and the pressure rise rate (dp/dp) corresponding to different EGR ratios at the spark timing of MBT, respectively. Also, the baseline is listed for comparison, which corresponds

| Table 4. Intake Manifold Pressure at Different EGR Ratios |
|----------------------------------------------------------|
| EGR ratio | 0% | 5% | 10% | 15% | 20% |
| throttle opening | 14% | 15.5% | 16.5% | 18% | 19.5% |
| intake manifold pressure | 48.16 kPa | 50.44 kPa | 53.48 kPa | 55.20 kPa | 58.18 kPa |
to the OER of 21% with an EGR ratio of 0%. Figure 8 lists the COV$_{\text{IMEP}}$ under different EGR ratios.

It can be seen from Figure 6 that the crank angle corresponding to the peak cylinder pressure continued to move backward after the EGR ratio increased, but it was still earlier than the baseline. The peak cylinder pressure decreased with the increase of the EGR ratio, but the minimum value was still larger than the baseline. This is because the violent collision of fuel molecules with oxygen molecules triggers combustion, and the supply of oxygen determines whether the combustion process is complete. Even if exhaust gas diluted the oxygen content of the in-cylinder mixture, the high oxygen content can still ensure that the fuel molecules and oxygen molecules have a large collision probability under the OER of 25%. Therefore, a higher cylinder pressure level than the baseline was guaranteed after adding EGR.

The cylinder pressure curve was very steep when EGR was 0% and gradually became flat as EGR was added. This is mainly because at an OER of 25%, the flame propagation speed is fast, and the rapid combustion and heat release process lead to a rapid rise in cylinder pressure. From the reference, the LFS of iso-octane increases linearly with oxygen enrichment concentration. When the OER is 25%, the LFS is 1.5 times that of the OER of 21%. This can also be verified from Figure 7, in which the maximum pressure rise ratio at an EGR rate of 0% was much higher than others.

It can also be seen from Figure 8 that the COV$_{\text{IMEP}}$ was relatively high when EGR was 0%, which was 2.36–3.24% under different spark timing. With the addition of EGR, the COV$_{\text{IMEP}}$ was gradually reduced to equal to or less than the baseline, which meant that the EGR can greatly alleviate the cycle-by-cycle variation in the oxygen-enriched condition. EGR can effectively reduce the possibility of knocking combustion under OEC thus reducing the COV$_{\text{IMEP}}$. However, the cycle-by-cycle variation after adding EGR was more sensitive to the spark timing, and a more stable state can only be achieved at about 15°CA BTDC to 20°CA BTDC. The main reason was consistent with the sensitivity of torque to spark timing.

3.3. Cylinder Temperature, CA$_{\text{0–10}}$, and CA$_{\text{10–90}}$.

Figure 9 shows cylinder temperature under different EGR ratios and the baseline, in which the ignition timing is set at the MBT. As the EGR ratio increased, the maximum cylinder temperature decreased, and the crank angle corresponding to the maximum cylinder temperature was delayed. This is
because the exhaust gas increases the specific heat capacity of the air–fuel mixture, and the cylinder temperature decreases when the amount of gasoline is unchanged. After adding EGR, the cylinder temperature at an EGR ratio of 5–20% was 46.05, 58.04, 96.73, and 145.01 K lower than the EGR ratio of 0%. However, the cylinder temperature at an EGR ratio of 20% was higher than those of 159.00 K the baseline. This can be explained that although the exhaust gas slows down the heat release rate (HRR) and the LFS, OEC can fully convert the chemical energy of fuel into thermal energy. The effects of EGR and OEC on the formation of flame core and flame propagation during combustion are opposite. According to ref 32, the initial flame core transitions to flame growth earlier, and the flame front surface propagates faster on the OEC. Based on the chemical reaction mechanism of gasoline combustion, H, O, and OH radicals have a decisive influence on the combustion chain reaction. Under the oxygen-enriched condition, the branching reaction of $H + O_2 \rightarrow OH + O$ becomes very active. The H active radical concentration is rapidly consumed at an OER of 25%, which increases the formation of O and OH reactive radical concentration. For gasoline combustion chemical reactions, many OH radicals represent a faster combustion process and cleaner combustion products. Oxygen concentration and combustion temperature play important roles in promoting the generation of O and OH radicals. The combustion intensity and reaction rate are accelerated on OER of 25%. So, when the OER was 25% and EGR ratio was 20%, the cylinder temperature was higher than the baseline.

It can be seen from Figure 10, when O$_2$ was 25% and EGR was 0%, the period of CA from spark timing to 10% of the total energy released (CA$_{0-10}$) was 3°C–4°C less than the baseline. With the increase in the EGR rate, CA$_{0-10}$ was gradually extended. It has exceeded the baseline when the EGR ratio was 15%. From Figure 11, when the EGR ratio was 0%, the period of CA from spark timing to 90% of the total energy released (CA$_{10-90}$) of the mixture was slightly lower than the baseline at a late spark timing, while higher than the baseline at an early ignition timing. The CA$_{10-90}$ was longer than the baseline when the EGR rate was higher than 10%. This phenomenon also reflects the strong promotion effect of O$_2$ on combustion and the strong hindering effect of EGR on combustion. Previous content has described how the OEC actively promoted the combustion of the mixture. For EGR, the exhaust gas mainly affects the combustion characteristics and emission characteristics of the engine by affecting the specific heat capacity of the mixture, diluting the oxygen concentration and reducing the collision probability between oxygen molecules and fuel molecules. With the addition of EGR, the positive promotion of OEC on the formation of the fire core was gradually offset, and CA$_{0-90}$ gradually increased.

From the cylinder temperature, CA$_{0-10}$, and CA$_{10-90}$, it is believed that OEC greatly increases the collision probability of molecules and accelerates the chain reaction. However, the formation of the flame core would be more affected by dilution and chemical reaction rate reduction caused by the exhaust gas, increasing the difficulty in ignition. However, once the flame core is formed, the OEC plays a decisive role again in the subsequent combustion process, greatly increasing the combustion temperature and allowing the fuel to release chemical energy more fully.

**3.4. Heat Release Rate.** Figure 12 depicts the variation of the HRR under different EGR ratios at the spark timing of MBT. When the EGR ratio changed from 0 to 20%, the peak value of the HRR decreased gradually, and the crank angle of the peak value was delayed. The reasons for this phenomenon are as follows. In the oxygen-enriched environment, oxygen molecules and fuel molecules underwent intense combustion at a very high reaction rate, resulting in a relatively higher and faster HRR. The main components of the exhaust gas emitted by the engine are CO$_2$ and N$_2$. With the increase of the EGR ratio, the CO$_2$ increased the specific heat capacity of the mixture and decreased the probability of collision between oxygen molecules and fuel molecules. Therefore, when the EGR ratio increased, the peak value of the HRR was reduced, and the crank angle corresponding to the peak was delayed. The HRR in the cylinder decreased, the combustion slowed down, and more heat was released during the downward movement of the piston, thus resulting in a decrease in the maximum cylinder temperature and maximum cylinder pressure.
It also can be seen from Figure 12 that the peak value and the crank angle corresponding to the peak of the baseline are between the characteristics at EGR of 5% and EGR of 10%. The HRR of the baseline had a higher and earlier peak than EGR of 15%−20%, which was mainly because the dilution effects of a large amount of exhaust gas on the mixture gradually reduced the positive effects of oxygen enrichment on combustion, making combustion and heat release delayed. However, the effects of oxygen-enriched combustion on the total heat release were still larger than that of EGR. When EGR was 20%, although the heat release lagged, the total heat release was still higher than the baseline, so the cylinder temperature and cylinder pressure were also higher than the baseline.

3.5. CO Emissions. Figure 13 shows the change in CO emissions under different EGR ratios. It can be seen that as the EGR ratio increased, the CO emissions decreased gradually. The main reasons for the generation of CO emissions in gasoline engines are as follows. One is incomplete combustion, and the other is thermal cracking of the combustion product CO\(_2\) under the effect of high temperature. Compared with the baseline (OER of 21%), OEC promoted the chemical reaction of CO + O\(_2\) → CO\(_2\) and accelerated the reaction velocity, which decreased the CO emissions greatly. When the EGR ratio changed in the range of 0−20%, exhaust gas increased the specific heat capacity of the cylinder mixture, and the total amount of working medium in the cylinder also increased. In the case of the same amount of fuel, the intake gas increased, and the maximum cylinder temperature decreased, so the CO emissions due to thermal cracking were reduced. Meanwhile, as the EGR rate increased, CA\(_0−90\) became longer, leading to the postcombustion temperature and the postoxidation increase, which reduced the CO emissions. In the whole experiment process, the chemical reaction environment of combustion was oxygen-enriched, which was more conducive to the full combustion of the mixture, so that the CO emissions were relatively reduced. Overall, oxygen enrichment greatly improved CO emissions. In an oxygen-enriched atmosphere, CO emissions maintained a decreasing trend as EGR increased.

3.6. HC Emissions. Figure 14 shows the HC emissions at different EGR ratios and spark timings. As the EGR ratio increased, the HC emissions at different spark timings were higher. The main origins of HC emissions include fuel incomplete combustion, flame quenching closing to the cylinder wall, and the narrow cracks. According to the simplified mechanism of unburned hydrocarbon proposed by Semenov,\(^3\) if part of the fuel in the combustion chamber cannot go through the high-temperature combustion, it would either slowly oxidize or undergo low-temperature combustion. Finally, unburned hydrocarbons would be discharged from the cylinder. As the EGR ratio increased, the maximum cylinder temperature decreased, and the incomplete combustion and flame quenching increased, increasing unburned HC emissions. The flame quenching closing to the cold cylinder wall at the early combustion stage would also lead to serious interruption of flame propagation in the combustion chamber’s narrow cracks at the later combustion stage.\(^5\) When the cylinder pressure dropped in the expansion stroke, the unburned hydrocarbons accumulated in the narrow cracks would re-enter the cylinder. However, the cylinder temperature has decreased at this time, and the oxygen content at the end
of the main combustion period would be low. Therefore, the unburned hydrocarbons cannot be oxidized, but rather be directly discharged in the exhaust stroke. Meanwhile, considering the effect of the EGR on HC emissions, the author believed that the EGR rate should not be continued to increase above the ratio of 20%. In short, in the case of OER of 25% with an EGR ratio of less than 20%, HC emissions can be kept lower than the baseline, but the deterioration caused by EGR was obvious. Compared with the CO emissions, the effects of OEC on HC reduction cannot have an overwhelming advantage. It needs to be used in conjunction with a proper EGR ratio so that HC emissions are at a lower level.

3.7. NOx Emissions. Figure 15 shows the change in NOx emissions for different EGR ratios under five spark timings.

EGR can effectively reduce NOx emissions under oxygen-enriched conditions. With the increase in the EGR rate, NOx emissions gradually decreased. It should be noted that due to the limitation of the measuring instrument range, there were some values of NOx emissions that exceed the measuring instrument range at several operating points (EGR ratio of 20% under all spark timings, EGR ratio of 15% under spark timing of 15°CA BTDC–25°CA BTDC, EGR ratio of 10% under spark timing of 20°CA BTDC–25°CA BTDC). Meanwhile, when the EGR ratio was 20%, the NOx emissions can be reduced to below 900–4000 ppm under different spark timings, which were all lower than the baseline. When EGR ratios were 15–20%, NOx emissions decreased to 92.3 and 51.4% of the baseline at MBT of 15°CA BTDC and dropped to 57.6 and 32.1% of the EGR ratio of 0% at the same MBT. EGR worked effectively for reducing NOx emissions from OEC.

NOx emissions were sensitive to the change in spark timing. At the spark timing of MBT, the EGR rate of 15% can reduce the NOx emissions produced by OEC to below the baseline. EGR can effectively reduce NOx emissions generated by the high temperature and oxygen-enriched environment during the OEC process. When the spark timing was advanced, the cylinder temperature and cylinder pressure were higher during the combustion process. Under the oxygen-enriched environment, once the cylinder temperature increased, lots of NOx would be generated. While the spark timing was delayed, the cylinder pressure and cylinder temperature were lower, and the NOx emissions would be lower. In general, because of the large specific heat capacity and flame-retardant properties of the exhaust gas, the maximum cylinder temperature is reduced, which greatly reduces the high NOx emissions caused by oxygen-enriched combustion. EGR can reduce the NOx emissions caused by OEC significantly. In this study, EGR ratios of 15–20% can significantly suppress NOx emissions.

3.8. Particle Emissions. Figure 16 shows the variation of particle number (PN) distribution characteristics corresponding to different EGR ratios at the spark timing of MBT. As the EGR ratio increased, the PN decreased in all particle size ranges. When the OER was 25% and EGR was 0%, the size distribution curve of the PN was mainly unimodal, and the peak value appeared in the small particle size range of 8.66 nm.

However, after adding EGR, the PN distribution curve gradually changed to a bimodal or multimodal distribution, while the corresponding particle size of the peak appeared in the middle particle size region, and the PN decreased significantly. In this study, the decoupling of the two particle forming factors, high temperature and oxygen deficiency, has been achieved. It can be found that when the OER was 25% and the EGR ratio was 0%, the PN was much higher than that of the baseline. This showed that high temperature was the primary factor in the sharp increase of the PN in this paper. In OEC technology, at some level, the oxygen enrichment could also promote soot oxidation in the postflame region, but the high oxygen concentration accelerated the fuel consumption and increased the flame temperature, which was very favorable to the formation of soot. The combustion temperature in the cylinder takes 1700 K as the dividing line. The particles below 1700 K are mainly evolved from polycyclic aromatic hydrocarbons (PAHs). At the combustion temperature above 1700 K, polycetylene and carbon vapor become the main precursors of particles. In a high-temperature environment, among the inferior hydrocarbons formed by thermal decomposition, the substance that is not in contact with O2 would continuously dehydrogenate to eventually become carbon particles.

However, with the increase of the EGR ratio, the maximum cylinder temperature and the in-cylinder maximum pressure were reduced. Du et al. held the view that EGR can affect the PN emissions through the chemical reaction and thermostatic. EGR exerts a strong influence on the soot particle
inception limit by limiting the formation of particle precursors and surface growth, as a participating species in radiative heat transfer. Meanwhile, CO\(_2\) in the exhaust gas can suppress soot formation due to chemical effects, not just through decreasing reactant concentrations and flame temperature reduction. Specifically, the chemical effect is associated with the reaction CO\(_2\) + H → CO + OH, which increases the concentration of hydroxyl radicals (OH) and decreases the H-radicals. The increase of OH concentration would enhance the oxidative attack on soot precursors. This change is beneficial to inhibit the thermal cracking of gasoline fuel in the combustion reaction, which reduce the conversion rate of carbon vapor, inhibit the formation of primary soot, and decrease its polymerization and cyclization. On the other hand, with the assistance of EGR, the inhibitory effect of oxygen enrichment on PN can be reflected. After the cylinder temperature is lowered, the combustion mode of EGR plus OEC avoids the two factors that lead to particle emissions, high temperature and lack of oxygen. EGR greatly improved particle distribution and emissions.

In this paper, the nucleation mode particle number (NPN) denotes the particle number of nucleation mode (5–50 nm diameter), and the accumulation mode particle number (APN) denotes the particle number of accumulation mode (50–1000 nm diameter).

Figure 17 shows the variations of NPN, APN, and TPN with EGR ratios at a spark timing of MBT, respectively. With the addition of EGR, both NPN and APN showed the same great decrease. When the EGR ratio was 0%, the NPN was the main component in particle emissions. With the increasing EGR ratio, the NPN decreased strikingly, lower than the APN. Typically, the majority of particle emissions in a PFI engine are NPN because of premixed combustion reducing local oxygen-deficient areas and the relatively lower carbon content of gasoline molecules. Moreover, in the high-temperature and oxygen-enriched environment at OER of 25% (EGR ratio of 0%), the small size particles are not easy to aggregate and form APN. While as the increasing EGR ratio, the cylinder temperature decreased, which reduced the oxidation rate of soot particles, leading to the increased accumulation mode particles. This appears to have been more significant than the decreases in primary carbon particles formed by the thermal pyrolysis and dehydrogenation reaction of fuel droplets, which would have occurred with the reduced in-cylinder temperatures. The inferior hydrocarbon particles became coarser, and the PAHs were more likely to aggregate into huge size particles. Meanwhile, the increase in unburned hydrocarbon emissions with the increasing EGR ratio also prompted the production of APN. Therefore, when the EGR ratio was 0%, the majority of particle emissions were the NPN, and the APN became even more than NPN with the increasing EGR ratio.

In general, when the EGR rate was larger than 10%, TPN can be reduced to a level lower than the baseline. While the EGR ratios were 15–20%, the NPN and the APN both were lower than those of the baseline. The particle emissions under an EGR ratio of 20% corresponded to the optimal particle emissions under an OER of 25%. TPN, NPN, and APN can be reduced to 45.9, 35.9, and 64.3% of the baseline, respectively.

4. CONCLUSIONS

In this paper, the bench test was used to study the engine combustion and emissions characteristics when the OEC and EGR were applied. In the experiment, the method of fixing the OER (25%) and changing the EGR rate (0, 5, 10, 15, and 20%) were used at an engine speed of 1500 rpm and an intake manifold pressure of 48.16–58.18 kPa. The deterioration of NO\(_x\) and particle emissions from gasoline engines caused by oxygen-enriched combustion was addressed simultaneously, while maintaining the significant advantages of oxygen-enriched combustion in terms of power and emission performance. The main research conclusions are as follows.

1. With the increase in the EGR ratio, the engine torque would increase significantly, which was 10.4% higher than the baseline at an EGR ratio of 20%. The maximum cylinder pressure and maximum cylinder temperature gradually decreased, and the mechanical losses and heat transfer losses were reduced. The addition of EGR in an oxygen-enriched environment gradually reduced the combustion HRR, and the HRR of EGR ratios of 15–20% were lower than the baseline. However, the total heat release value remained above the baseline level. Meanwhile, the appropriate EGR rate reduced the cylinder pressure change rate, making the engine power process smoother in an oxygen-enriched environment.

2. With the increase in the EGR ratio, the CO and NO\(_x\) emissions were greatly reduced. Especially at an OER of 25%, the NO\(_x\) emissions after adding EGR ratios of 15–20% can be reduced to the baseline or even lower. EGR had a very strong inhibitory and improvement effect on the deterioration of NO\(_x\) emissions on intake oxygen-enriched combustion. However, HC emissions increased slightly after adding EGR, indicating that adding EGR would worsen the generation of HC emissions. Importantly, in this paper, the EGR ratio of 20% can make the HC emissions lower than the baseline, which is mainly because the OER of 25% plays a huge role in it.

3. After adding EGR, a large number of PN caused by OEC were greatly improved. TPN at the baseline was about 2 × 10\(^7\) (particles/cm\(^3\)). When the OER was 25%, due to the influence of the high cylinder temperature, the TPN was about 10 times higher than the baseline. However, when EGR was added, the TPN decreased significantly. When the EGR rate exceeded 5%, particle emissions could be reduced to below the baseline. When the EGR ratio was
20% at the spark timing of MBT, particle emissions were only 45.9% of the baseline.

(4) In general, adding 15–20% EGR can effectively reduce NOx emissions and particle emissions, which maintain good engine power and effective thermal efficiency. Also, from the viewpoint of HC emissions, it is considered that the EGR rate should not exceed 20%. Therefore, the OER of 25% combined with an EGR ratio of 15–20% is the remarkable condition in this paper.

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Notes
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■ ABBREVIATION

CA crank angle
BTDC before top dead center
N2 nitrogen
O2 oxygen
HC hydrocarbon
CO carbon monoxide
CO2 carbon dioxide
NOx nitrogen oxide
EGR exhaust gas recirculation
OEC oxygen-enriched combustion
OER oxygen-enriched ratio
H2O standard hydrogen–oxygen mixtures
PI port injection
DI direct injection
SI spark ignition
ECU electronic control unit
PFI port fuel injection
ODI oxygen direct injection
MBT minimum advance for best torque
HRR heat release rate
TPN total particle number
APN accumulation mode particle number
NPN nucleation mode particle number

■ SYMBOL

CO2in the CO2 concentration (volume ratio) in the intake system
CO2ex the CO2 concentration in the exhaust system
COVIMEP the standard deviation of COVIMEP
IMEP the mean value of COVIMEP

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