Implementation of Oxy-Fuel Combustion (OFC) Technology in a Gasoline Direct Injection (GDI) Engine Fueled with Gasoline—Ethanol Blends

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ABSTRACT: Nowadays, to mitigate the global warming problem, the requirement of carbon neutrality has become more urgent. Oxy-fuel combustion (OFC) has been proposed as a promising way of carbon capture and storage (CCS) to eliminate carbon dioxide (CO2) emissions. This article explores the implementation of OFC technology in a practical gasoline direct injection (GDI) engine fueled with gasoline—ethanol blends, including E0 (gasoline), E25 (25% ethanol, 75% is gasoline in mass fraction), and E50 (50% ethanol, 50% is gasoline in mass fraction). The results show that with a fixed spark timing, φCA50 (where 50% fuel is burned), of E50 and E25 is about 4.5 and 1.9° later than that of E0, respectively. Ignition delay (θF) and combustion duration (θC) can be extended with the increase of the ethanol fraction in the blended fuel. With the increase of the oxygen mass fraction (OMF) from 23.3 to 29%, equivalent brake-specific fuel consumption (BSFCe) has a benefit of 2.12, 1.65, and 1.51% for E0, E25, and E50, respectively. The corresponding increase in brake-specific oxygen consumption (BSOC) is 21.83, 22.42, and 22.58%, respectively. Meanwhile, θF, θC, and the heat release rate (HRR) are not strongly affected by the OMF. With the increase of the OMF, the increment of θF is 0.7, 1.8, and 2.2° for E0, E25, and E50, respectively. θC is only extended by 1, 1.1, and 1.4°, respectively. Besides, by increasing the intake temperature (TI) from 298 to 358 K under all of the fuel conditions, BSFCe and BSOC present slight growth trends; θF and θC are slightly reduced; in the meantime, φCA50, φPmax (crank angle of peak cylinder pressure), and the position of the HRR peak are advanced by nearly 1°.

1. INTRODUCTION

Climate change, particularly global warming, has been a serious problem, which causes a wide range of effects on the environment. Hence, carbon neutrality has been proposed as an urgent need to limit global warming by reducing greenhouse gas (GHG) emissions.1−4 Aiming at reducing the emissions of primary long-lived GHG carbon dioxide (CO2), oxy-fuel combustion (OFC) technology is helpful in achieving carbon capture and storage (CCS) in conventional internal combustion engines (ICEs) fueled with fossil fuels.5−8 OFC technology was proposed by Yaverbaum,5 and the chemical reaction process is shown in eq 1. It presents that the major advantage of OFC is the avoidance of emissions related to the nitrogen element so that the engine exhaust emissions mostly comprise CO2 and H2O. Then, H2O is condensed and separated with a condenser and a gas/water separator. A portion of the remaining CO2 is recirculated back to the cylinders for utilization. Meanwhile, the rest of the CO2 is compressed, captured, and stored. The physicochemical properties of CO2 and nitrogen are listed in Table 1, which render OFC quite different from conventional air combustion (CAC).9,10

\[ \text{C}_2\text{H}_4 + \left( x + \frac{y}{4} \right) \text{O}_2 \rightarrow x\text{CO}_2 + \frac{y}{2} \text{H}_2\text{O} \] (1)

In 1999, Bilger6 introduced a novel system named the internal combustion Rankine cycle (ICRC), initiating the application of OFC technology into spark ignition (SI) ICES. In the ICRC system, CO2 with oxygen enters into the engine combustion chambers. The other prominent characteristic is that water is directly injected into the chambers near the top dead center to control combustion. Over the last decade, Wu et al.11−14 provided valuable inputs into OFC research, which indicated that the power, fuel economy, and emissions could be improved through optimization strategies in the ICRC port fuel injection (PFI) engine fueled with propane. Li et al.15,16 explored the potential of the intake charge to optimize...
Table 1. Gas Physicochemical Properties at 1000 K and 0.1 MPa9,10

| property            | CO₂  | nitrogen | ratio (CO₂/nitrogen) |
|---------------------|------|----------|----------------------|
| molecular weight    | 44   | 28       | 1.57                 |
| density (kg/m³)     | 0.5362 | 0.3413 | 1.57                 |
| kinematic viscosity (m²/s) | 7.69 × 10⁻⁵ | 1.2 × 10⁻⁴ | 0.631 |
| specific heat capacity (kJ/kg K) | 1.2343 | 1.1674 | 1.06                 |
| thermal conductivity (W/m K) | 7.057 × 10⁻² | 6.599 × 10⁻² | 1.07 |
| thermal diffusivity (m²/s) | 1.1 × 10⁻⁴ | 1.7 × 10⁻⁴ | 0.644 |
| mass diffusivity of O₂ (m²/s) | 9.8 × 10⁻⁵ | 1.3 × 10⁻⁴ | 0.778 |
| Prandtl number      | 0.7455 | 0.7022 | 1.06                 |
| emissivity and absorptivity | >0    | ~0      | 1.06                 |

Figure 1. Schematic of OFC in the application of an ICE with CCS.
some items are listed in Table 3.30 Besides, the spark timings were optimized to be the minimum advance for maximum brake torque (MBT) or the knock-limited spark advance (KLSA). The fuels used in this study include E0 (gasoline), E25 (25% ethanol, 75% is gasoline in mass fraction), and E50 (50% ethanol, 50% is gasoline in mass fraction), which are mixed to ensure they are completely miscible before the test. The fuel properties of gasoline and ethanol used are presented in Table 4.

### 2.2. Model Description and Research Approach.

The one-dimensional model of this numerical study is established by GT-Power, which is commonly used in academia in the research of SI engines.31−33 The main submodels are the “Woschni model”34 and the “SI turbulent flame combustion model.”35 The heat transfer coefficient $h$ and laminar flame speed $S_L$ are presented in eqs 2 and 3, respectively.

$$h = 110d^{-0.2}P^{0.8}T^{-0.5}\left[ C_m + C_s \frac{V_S}{P \Delta T} \left( P - P_0 \right) \right]^{0.8}$$

Here, $h$ denotes the heat transfer coefficient, $d$ denotes the diameter of the cylinder bore, $P$ denotes the cylinder pressure, $T$ denotes the in-cylinder mean gas temperature, $C_m$ denotes a constant related to the airflow velocity coefficient, $C_s$ denotes a constant related to the combustion chamber, $c_m$ denotes the mean piston speed, $V_S$ denotes the cylinder volume, and $P_0$ denotes the cylinder pressure when the engine is started. $T_1$, $P_1$, and $V_1$ are cylinder temperature, pressure, and volume at the beginning of compression, respectively.

$$S_L = \left[ B_m - B_0 (\delta - \delta_m)^2 \right] \frac{T_u}{T_{ref}} \left( P \right)^{\alpha} \left( P_{ref} \right)^{\beta} f(D)$$

Here, $S_L$ denotes the laminar flame speed, $B_m$ denotes the maximum laminar speed, $B_0$ denotes the laminar speed roll-off value, $\delta$ denotes the in-cylinder equivalence ratio, $\delta_m$ denotes the equivalence ratio at the maximum speed, $T_u$ denotes the unburned gas temperature, $T_{ref}$ denotes 298 K, $P$ denotes the pressure, $P_{ref}$ denotes 101.325 kPa, and $\alpha$ denotes the temperature.
exponent, $\beta$ denotes the pressure exponent, and $f(D)$ denotes the dilution effect.

In this research, fueled with E0, E25, and E50, the engine runs at 1500 revolutions per minute (rpm) with 10 bar brake mean effective pressure (BMEP), representing a mid-high load of engine operating conditions. The research approach of this study is illustrated in Figure 3.

![Figure 3. Flow chart of the research approach.](image)

First, model validation is completed based on the experimental data. Second, the optimization of OFC performance by changing spark timing is conducted. In the meantime, the throttle opening angle and stoichiometric air–fuel ratio are held constant. Third, the performance optimization by changing the oxygen mass fraction (OMF) is analyzed. When the OMF changes, the throttle opening angle remains unchanged and the spark timings should be optimized to be the MBT under each OMF condition. Lastly, simulation work is conducted in an attempt to optimize the engine performance by changing the intake temperature.

In this study, ignition delay ($\theta_i$) denotes the crank angle (CA) interval between spark timing and $\varphi_{C10}$ (where 10% of the fuel is burned). Combustion duration ($\theta_C$) denotes the CA interval between $\varphi_{C10}$ and $\varphi_{C90}$ (where 90% of the fuel is burned). Besides, $\varphi_{CASO}$, $T_m$, and $\varphi_{P_{max}}$ are introduced to denote the CA where 50% of the fuel is burned, the maximum in-cylinder temperature, and the CA of the peak cylinder pressure. Brake-specific oxygen consumption (BSOC), equivalent brake-specific fuel consumption (BSFC$_E$), and $\lambda_{O_2}$ are introduced in eqs 4–6, respectively.

$$\text{BSOC} = \frac{\tau_O \times 1000}{P}$$  \hspace{1cm} (4)

$$\text{BSFC}_E = \frac{\tau_F \times 1000}{P} \times \frac{(\omega_{E_i} \times H_E) \times (\omega_{G_i} \times H_G)}{H_G}$$  \hspace{1cm} (5)

$$\lambda_{O_2} = \frac{\tau_O}{\tau_{soi}}$$  \hspace{1cm} (6)

Here, $P$ (kW) denotes the engine brake power. $\tau_O$ (kg/h) and $\tau_F$ (kg/h) are the consumption rates of oxygen and fuel under actual conditions, respectively. $\tau_{soi}$ (kg/h) denotes the oxygen mass flow rate at the stoichiometric condition. $\omega_{E_i}$ and $\omega_{G_i}$ are the mass fractions of ethanol and gasoline in the fuel, respectively. $H_E$ and $H_G$ are the low heating values of ethanol and gasoline, respectively.

3. RESULTS AND DISCUSSION

3.1. Model Validation. Figure 4 presents the model validation by a comparison of cylinder pressure between experimental and simulation results under E0, E25, and E50 conditions. It can be seen that the curves are in good agreement under all of the conditions. The locations and magnitudes of the curve peak have been well predicted. This indicates that this model is capable of being used for this numerical research.

![Figure 4. Comparison of cylinder pressure between experimental and simulation results.](image)

3.2. Performance Optimization by Changing Spark Timing. This section shows the effects of spark timing on engine combustion performance. Meanwhile, the OMF and $T_i$ are kept at 23.3% and 298 K, respectively.

![Figure 5. Effects of spark timing on BSFC$_E$ and $\varphi_{CASO}$.](image)

Figure 5 presents the effects of spark timing on BSFC$_E$ and $\varphi_{CASO}$. It can be observed that with the advance of spark timing from $-40$ to $-68$ °CA, the overall trend of BSFC$_E$ initially has a small reduction and then increases. For E0, E25, and E50, the lowest BSFC$_E$ is 317.62, 306.48, and 295.82 g/kWh, which is achieved with the spark timing of $-52$, $-54$, and $-58$ °CA, respectively.

These differences can be observed with the combustion phasing characterized by $\varphi_{CASO}$, $\theta_i$, and $\theta_C$. The corresponding
$\phi_{CA50}$ values with MBT timing are 4, 4.5, and 4.4 °CA, respectively. Besides, there is a clear contrast between the overall trend of $\theta_F$ and $\theta_C$, as shown in Figure 6. By advancing spark timing from −40 to −68 °CA, $\theta_F$ increases by about 11.2, 11.2, and 11.1° for E0, E25, and E50, respectively. In the meantime, $\theta_C$ has a corresponding decline of 9.2, 11.2, and 14.5°. It can be explained by the heat release rate (HRR), an example case with E25 of Figure 7; it also demonstrates that the combustion phasing is very sensitive to spark timing, which leads to a considerable variation in both the location and magnitude of the HRR.

Another important result presented in this section is that with a fixed spark timing, the $\phi_{CA50}$ of E50 and E25 is about 4.5 and 1.9° later than that of E0, as shown in Figure 5. Meanwhile, $\theta_F$ and $\theta_C$ can be extended with the increase of ethanol fraction in the blended fuel. The changes can be attributed to two main reasons by fuel properties, as shown in Table 4. First, the latent heat of vaporization of ethanol is significantly higher than that of gasoline, leading to a stronger cooling effect and suppression of combustion rate. The $T_M$ of E50 is the lowest among the three fuels, while that of E0 is the highest, as shown in Figure 8. Second, the laminar flame speed of ethanol is higher than that of gasoline, which would promote combustion rate and complete combustion. However, the benefit cannot counteract the negative effects of the high latent heat of vaporization of ethanol in this operating condition.

3.3. Performance Optimization by Changing the OMF. This section presents the optimization results on engine combustion performance by changing the OMF. Furthermore, $T_1$ is kept at 298 K, and MBT timing is applied for all operating conditions.

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Figure 6. Effects of spark timing on $\theta_F$ and $\theta_C$.

Figure 7. Effects of spark timing on the HRR (E25).

Figure 8. Effects of spark timing on $T_M$.

Figure 9. Effects of the OMF on BSFC_E.

Figure 10. Effects of the OMF on $\theta_F$ and $\theta_C$.
OMF. With the increase of the OMF, the increment of $\theta_F$ is 0.7, 1.8, and 2.2° for E0, E25, and E50, respectively. Meanwhile, $\theta_C$ is only extended by 1, 1.1, and 1.4°, respectively (Figure 11). This is mainly attributed to the negative impact of the lean fuel–air mixture ($\lambda_{O_2} > 1.1$) on laminar burning velocity on increasing the OMF to 27 or 29%, although the impact is partially offset by the influence of decreasing CO2 fraction in the intake.35,36 This can also be further explained by the HRR, an example case with E25 in Figure 12. It demonstrates that there is no apparent discrepancy in the HRR on increasing the OMF. The peak of the HRR is just slightly decreased by 2.5 J/CA and delayed by 1.5°.

3.4. Performance Optimization by Changing the Intake Temperature. In this section, the simulation work is conducted in an attempt to optimize the engine performance by changing the intake temperature from 298 to 358 K. Furthermore, the OMF conditions are selected with 23.3 and 29%, and the spark timings are optimized to be MBT.

Figure 13 and Figure 14 show the effects of $T_I$ on BSFC_E and BSOC, respectively. It can be seen that with the increase of $T_I$ from 298 to 358 K, all of the curves of BSFC_E and BSOC present steady but slight growth trends. Hence, an analysis of normalization is also depicted in Figure 13 to show the comparisons with the condition of $T_I = 298$ K. The average increase rate is, respectively, 0.28 and 0.23% for OMF = 23% and OMF = 29%. This is mainly because the intake density will be reduced on increasing $T_I$ under a fixed opening angle of engine throttle.

In the meantime, the combustion phasing will be slightly affected on changing $T_I$ under E0, E25, and E50 fuel conditions. As shown in Figures 15 and 16, on increasing $T_I$ from 298 to 358 K, $\theta_F$ and $\theta_C$ will be reduced by around 0.7°. $\varphi_{CASO}$ $\varphi_{Pmax}$ and the position of the HRR peak will be advanced by nearly 1°. This is principally because the atomization of fuel droplets could be enhanced with a higher temperature intake.

4. CONCLUSIONS

This work belongs to the “RIVER” project to develop a noncarbon riverboat powered by an ICE with conventional hydrocarbon liquid fuels, which is expected to make a valuable contribution to carbon neutrality in the world. The findings of
this work not only provide a critical analysis on the implementation of OFC technology in a practical GDI engine fueled with gasoline–ethanol blends but also continue to contribute to this growing area by exploring the methods of improving the efficiency of OFC SI engines. The major conclusions in this article can be summarized as follows:

(1) BSFC, $\theta_F$, and $\phi_C$ are sensitive to spark timing under OFC mode for all of the fuel conditions (E0, E25, and E50).

(2) With a fixed spark timing, the $\phi_{CAS0}$ of E50 and E25 is about 4.5° and 1.9° later than that of E0, respectively. $\theta_F$ and $\phi_C$ can be extended by increasing the ethanol fraction in the blended fuel.

(3) With MBT timing under each OMF condition and by increasing the OMF from 23.3% to 29%, the saving rate of BSFC is 2.12, 1.65, and 1.51% for E0, E25, and E50, respectively. The corresponding increase in BSOC is 2.12, 1.65, and 1.51% for E0, E25, and E50.

(4) $\theta_F$, $\phi_C$, and HRR are not sensitive to the OMF. With the increase of the OMF, the increment of $\theta_F$ is 0.7, 1.8, and 2.2° for E0, E25, and E50, respectively. $\phi_C$ is only extended by 1, 1.1, and 1.4°, respectively.

(5) By increasing $T_i$ from 298 to 358 K, BSFC and BSOC present steady but slight growth trends under all of the fuel conditions. $\theta_F$ and $\phi_C$ could be slightly reduced, while $\phi_{CAS0}$ and $\phi_{pmax}$ and the position of HRR peak could be advanced by nearly 1°.

In the future, more research on OFC in GDI engines fueled with gasoline–ethanol blends would be beneficial. For example, further studies could include other new parameters, such as the effects of variable valve actuation strategies, exhaust gas recirculation (EGR), intake pressure and temperatures, fuel injection rate and pressure, etc. Besides, the studies about combustion performance under some other representative load points can also be considered. Thus, future works can further benefit the implementation of OFC in GDI engines fueled with gasohol, providing a practical and meaningful way to help achieve zero carbon emissions from ICE.

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### ABBREVIATIONS

- BMEP brake mean effective pressure
- BSFC brake-specific fuel consumption
- CA crank angle
- CAC conventional air combustion
- CCS carbon capture and storage
- CO2 carbon dioxide
- E0 gasoline
E25 25% ethanol, 75% is gasoline in mass fraction
E50 50% ethanol, 50% is gasoline in mass fraction
ECU electronic control unit
ERG gas recirculation
ERDF European regional development fund
GDI gasoline direct injection
GHG greenhouse gas
HRR heat release rate
IC internal combustion engine
ICRC internal combustion Rankine cycle
KLSA knock-limited spark advance
MBT maximum brake torque
OFC oxy-fuel combustion
OMF oxygen mass fraction
PFI port fuel injection
rpm revolutions per minute
SI spark ignition

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