A feasibility study of implementation of oxy-fuel combustion on a practical diesel engine at the economical oxygen-fuel ratios by computer simulation

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Abstract
To help achieve zero carbon emissions from inland waterway vessels, this implementation of oxy-fuel combustion on a practical diesel engine at the economical oxygen-fuel ratios were systematically studied and analysed in this paper. A 1-D simulation was used to explore the effect of various operating parameters for recovering the engine power when the engine is modified to the oxy-fuel combustion from conventional air combustion. The brake power of oxy-fuel combustion is only 26.7 kW that has a noticeable decline compared with 40 kW of conventional air combustion with fixed consumption of fuel and oxygen. By optimising some valuable parameters, like fuel injection timing, intake charge temperature, intake components, engine compression ratio and water injection strategy, a benefit of 6.8 kW has been acquired in the engine power. Afterwards, a remarkable benefit was obtained with the increase of lambdaO\textsubscript{2} from 1.0 to 1.5, finally obtaining the same engine power with the conventional air combustion. Above all, taking advantage of various operating parameters, it is expected to further improve the value of the implement of oxy-fuel combustion on diesel engines at the economical oxygen-fuel ratios.

Keywords
Oxy-fuel combustion, diesel engine, computer simulation, carbon capture and storage

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Introduction
During the last few decades, the human-made impacts on climate has started to pose a significant threat to the planet. The impacts are largely linked to an increase in the naturally occurring Greenhouse Gas (GHG) emissions.\textsuperscript{1–5} Statistics show that the level of atmospheric Carbon Dioxide (CO\textsubscript{2}) has increased continuously since the Industrial Revolution, and it is generally recognised as the main factor to the climate changes.\textsuperscript{6} Thus the concept of nations becoming 'carbon-neutral'
is broadly accepted to avoid further deterioration of atmosphere due to GHG emissions.\textsuperscript{2,3} Through the implementation of strict emission standards, the European Union (EU) now also requires to reduce carbon emissions from the non-road mobile machinery.\textsuperscript{7} Hence, to meet the carbon neutral targets, it is also mandatory to cut the carbon emissions from the Inland Waterway (IW) vessels. These vessels usually adopt Internal Combustion (IC) engines fuelled with diesel as their primary power source. Some recent innovative technologies, like hybrid electric, plug-in hybrid electric, battery electric, fuel-cell and solar-powered, have been shown to help achieve near-zero carbon emissions for diesel engines. Nevertheless, due to the bottlenecks in the power output and the high cost of these technologies mean that their application is currently limited only to in light-duty passenger vehicles rather than vessels. The Carbon Capture and Storage (CCS) technology has been widely considered as a promising solution to reduce CO\textsubscript{2} emissions from fossil fuel power generation.\textsuperscript{8}

The chemical reactions for Conventional Air Combustion (CAC) and Oxy-Fuel Combustion (OFC) are shown in equation (1) to equation (2), respectively. The distinguishing feature of oxy-fuel combustion is the reactants and products. Table 1 below indicates the physiochemical properties of CO\textsubscript{2} and N\textsubscript{2}, which plays an important role in the IC engine. Oxygen, which affects the emission level of nitrogen oxides in CAC, is still employed as the oxidiser for combustion. In the OFC, the final products only consist of H\textsubscript{2}O and CO\textsubscript{2} theoretically, hence nitrogen oxides can be eliminated.

\[
\begin{align*}
\text{C}_x\text{H}_y + \text{O}_2 + \text{N}_2 & \rightarrow \text{CO}_2 + \text{H}_2\text{O} + \text{N}_2 \quad (1) \\
\text{C}_x\text{H}_y + \text{O}_2 & \rightarrow \text{CO}_2 + \text{H}_2\text{O} \quad (2)
\end{align*}
\]

Due to the characteristics mentioned above, it is of great value to utilise oxy-fuel combustion technology on IC engines of the IW vessels. In our ongoing research project named RIVER funded by the Interreg North-West Europe, OFC coupled with CCS is used to achieve a zero-carbon emissions for the engines of the IW vessels. The designed configuration for OFC in IC engines can be presented as Figure 1. The exhaust gas, which mainly contains CO\textsubscript{2} and H\textsubscript{2}O, should be condensed, followed by the separation of liquid H\textsubscript{2}O. Subsequently, a portion of remaining CO\textsubscript{2} is recirculated back to cylinders through an EGR system. Then CCS can be implemented effectively after water separation, resulting in the separation of CO\textsubscript{2}. Remaining CO\textsubscript{2} can be stored and used for other applications. The scope of this article covers only the combustion process and its control. The process of CCS will be a different topic for a separate article.

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All the previous studies\textsuperscript{16–21} related to OFC in Compression Ignition (CI) engines have mainly focused on the operating conditions of lean-burn mode, in

| Property                      | CO\textsubscript{2} | N\textsubscript{2} | Ratio (CO\textsubscript{2}/N\textsubscript{2}) |
|-------------------------------|----------------------|---------------------|---------------------------------------------|
| Molecular weight              | 44                   | 28                  | 1.57                                        |
| Density (kg/m\textsuperscript{3}) | 0.5362               | 0.3413              | 1.57                                        |
| Kinematic viscosity (m\textsuperscript{2}/s) | 7.69e-5              | 1.2e-4              | 0.631                                       |
| Specific heat capacity (kJ/kg K) | 1.2343               | 1.1674              | 1.06                                        |
| Thermal conductivity (W/m K)  | 7.057e-2             | 6.599e-2            | 1.07                                        |
| Thermal diffusivity (m\textsuperscript{2}/s) | 1.1e-4               | 1.7e-4              | 0.644                                       |
| Mass diffusivity of O\textsubscript{2} (m\textsuperscript{2}/s) | 9.8e-5               | 1.3e-4              | 0.778                                       |
| Prandtl number                | 0.7455               | 0.7022              | 1.06                                        |
| Emissivity and absorptivity   | >0                   | ~0                  | --                                          |
which the overall supplied air is generally excessive for
the burning fuel. However, the consumption of oxygen
is usually quite costly so that it will imply increasing
cost at the lean-burn conditions with high air-fuel
ratios. Hence, to improve the engine economic perform-
ance, it is necessary to facilitate the oxy-fuel combus-
tion process within economical operating conditions,
which are close to stoichiometric oxygen-fuel ratio.
This kind of method has rarely been found in the exist-
ing publications, although oxygen consumption is the
most critical cost factor in the OFC.

According to the project RIVER’s purpose, one-
dimensional simulation with GT-Power software was
performed to investigate the working process of an
oxy-fuel combustion diesel engine at the economical
oxygen-fuel ratios. This simulation’s target is to achieve
the equivalent power output with the CAC mode when
the engine is modified to the oxy-fuel combustion
mode. Moreover, the consumption of oxygen should be
saved as much as possible during the oxy-combustion
process.

Methodology and modelling description
As this OFC study needs achieve the equivalent brake
power with CAC, it is necessary to find an optimum
condition at the stoichiometric oxygen-fuel ratio
($\lambda_{O2} = 1$) by exploring each operating parameter
accompanying with a fixed amount of oxygen con-
sumption. The definition of $\lambda_{O2}$ can be explained
as equation (3). Here, $m_{O2}$ represents the oxygen mass
flow rate under actual condition, $m_{O2}^{st}$ represents the
oxygen mass flow rate at the stoichiometric condition.

$$
\lambda_{O2} = \frac{m_{O2}}{m_{O2}^{st}}
$$

If the power is still not recovered, the amount of
intake charge ($\lambda_{O2}$) can be increased in the final
stage. The overall flow chart of technology roadmap
can be presented as Figure 2 to illustrate this methodol-
ogy in detail.

The first step of this roadmap is to calculate the
engine power by replacing the intake $N_2$ with the same

Figure 1. General configuration of OFC technology in IC engines.

Figure 2. Flow chart of technology roadmap in this simulation.
volume of CO₂. Meantime, fuel injection timing is adjusted and optimised according to explore the variation trend of power performance. In turn, the intake charge temperature and components are also optimised under the previously optimised fuel injection timing. Afterwards, the potential of brake power can be explored by increasing engine compression ratio in the next section. Then, to maximise brake power, the technical measure of water injection is employed as well as accompanying all the optimised parameters previously. Finally, if the brake power is still less than 40 kW, the goal can be achieved by increasing lambdaO₂, and the comparison between OFC with CAC will be performed. All the results and discussions in the process towards the study’s target are described in the next sections, and the effects of each parameter are also systematically analysed in the meantime.

The simulation study was set up based on a commercial high-speed diesel engine which specifications are shown in Table 2. The high injection pressure can be realised by the common-rail injection system, leading to an improvement in the fuel economy and particle emissions.

The selected model of heat transfer and combustion in this simulation of GT-Power is ‘Woschni model’ and ‘Direct-injection diesel multi-pulse combustion model’, respectively. Some important parameters in this model can be explained with this formula below.

\[ h = 110d^{-0.2}P^{0.8}T^{-0.53}\left[C_1 c_m + C_2 \frac{V_S T_1}{P_1 V_1} (P - P_0)\right]^{0.8} \]  

(4)

Here, \( h \) is heat transfer coefficient. \( d, P \) and \( T \) is respectively the bore diameter, gas pressure and temperature. \( C_1 \) and \( C_2 \) is respectively the constants about airflow and combustion chamber. \( c_m, V_S \) and \( P_0 \) is respectively the mean piston speed, cylinder volume, cylinder pressure where engine is ignited. \( T_1, P_1 \) and \( V_1 \) stands for the instantaneous cylinder temperature, pressure and volume at the beginning of compression stroke, respectively.

\[ S_L = S_{L,0} \left( \frac{T_u}{T_{ref}} \right)^{\alpha} \left( \frac{P}{P_{ref}} \right)^{\beta} \left( B_m - B_{\phi}(\phi - \phi_m)^2 \right) \]  

(5)

\[ \left( \frac{T_u}{T_{ref}} \right)^{\alpha} \left( \frac{P}{P_{ref}} \right)^{\beta} f(d) \]

where, \( S_L \) and \( S_{L,0} \) is the instantaneous laminar flame speed and laminar flame speed at the room condition (298 K and 101.325 kPa). \( B_m, B_{\phi}, \phi \) and \( \phi_m \) respectively stand for the maximum laminar speed, laminar speed roll-off value, in-cylinder equivalence ratio, and equivalence ratio at maximum speed. \( \alpha \) and \( \beta \) is respectively the temperature exponent and the pressure exponent. \( f(d) \) is the dilution effect. \( T_u, T_{ref}, P \) and \( P_{ref} \) represents the unburned gas temperature, 298 K, pressure and 101.325 kPa, respectively.

This model has been validated with experimental data for the operating working condition of CAC as Table 3, and the result is shown in Figure 3. The two cylinder pressure curves for simulation and experiment are nearly coincident. The position and degree of the peaks of curves have been well predicted. The features indicate that this model is capable of being used for this research.

Besides, the fuel injection mass and timing is fixed and equal to the value of CAC throughout the simulation research of OFC.

In order to achieve flexible control and stable operating conditions of OFC mode, there should be some

| Parameter | Value |
|-----------|-------|
| Engine speed (rpm) | 2000 |
| Brake power (kW) | 40 |
| Stoichiometric air-fuel ratio | 14.3 |
| Oxygen flow rate (kg/h) | 31.76 |
| Fuel injection mass [mg/(cycle-cylinder)] | 39.7 |
| Fuel injection timing (°CA ATDC) | –4.5 |
| Intake charge temperature (K) | 298 |
| Engine compression ratio | 17 |
| Water injection mass (mg) | 0 |

Figure 3. Comparison of cylinder pressure between experiment and simulation under CAC mode.
useful changes to the engine. In the practical applications, it is essential to establish effective systems to implement the functions of oxygen feeding, intake organisation, water/fuel injection, CO2 separation, capture and storage, etc.

As shown in Figure 4, the system of engine intake charge should be modified. The oxygen can be fed into the inlet with one of the two typical oxygen feeding strategies, then enters into the engine cylinders accompanied by CO2. In this model, the strategy in Figure 4(a) was chosen to realise the transformation from CAC to OFC. Hence, the implemented method to realise OFC can be simplified without considering the EGR system and strategy. The simulation can redefine the initial ambient only involves oxygen and CO2, and the relevant parameters can be flexibly changed, such as intake charge temperature, intake component and mass flow rate. In the practical application, these parameters can be controlled by the utilisation of valves, flowmeters, heaters and mixer.

Despite that there is a risk to enhance corrosion of engine from moisture, water injection is still a considerable way to simultaneously achieve the target of maximum engine performance and avoid the knock’s damage. For the oxy-fuel IC engines, water injection technology has hitherto been adopted as a technical way to optimise combustion.23,24 Its current typical strategies can be presented in two types as Figure 5. In this model, the strategy of Figure 5(a) was selected due to its compact and practical control platform.

In this paper, the authors focus on a feasibility study of implementation of OFC on a practical diesel engine at the economical oxygen-fuel ratios by 1-D simulation. Rest of the paper is arranged as follows: replace N2 with CO2 and recovering power by optimising injection timing, then recovering power by optimising intake charge temperature, intake charge components, engine compression ratio and water injection strategy, finally comparing between OFC of increasing lambdaO2 and CAC, as well as the conclusions.

**Results and discussion**

*Replace N2 with CO2 and recovering power by optimising injection timing*

In this CI engine, diesel can be directly injected to cylinders by a high-pressure common rail, leading to that the fuel droplets can break up into tiny droplets quickly. This can help to improve the quality of vaporisation and atomisation, resulting in that a higher homogenous air-fuel mixture which is beneficial to promote complete combustion. Thus it is essential to find an optimal injection timing to improve the thermal efficiency for enhancing the engine power.

Due to the difference between the property of CO2 and N2, especially that the thermal diffusivity of CO2 is only 64.4% of N2 which can reduce the rate of flame transmission and heat transfer during the combustion process. Therefore, when the N2 of intake charge is
replaced with the same volume of CO₂ and all the other factors keep constant in the meantime, it is notable that brake power of OFC decreases significantly to 26.7 kW which is 13.3 kW less than that of CAC in Figure 6.

Besides, the reduced power can be recovered slightly in a certain range of advanced injection timings, continuously from 25.2 kW to 29.0 kW with the injection timing advances from −2 °CA ATDC (hereinafter referred to as ‘CA’) to −22 °CA. Furthermore, as the power becomes essentially constant around 29 kW and is not easy to be further improved with earlier injection timings, −22 °CA can be an optimised injection timing selection under OFC mode.

It can be seen that the curves of cylinder pressure also change a lot with different injection timings. Almost all the curves exhibit a bimodal distribution form in the OFC mode. All the first peaks which occur around −2°CA mainly reflect maximum cylinder pressure without combustion. Afterwards, the cylinder pressure will slightly decrease owing to the heat absorption of fuel droplets in the combustion chamber, followed by a rapid renewed increase after a couple of degrees. The curves’ second peak which occurs during the combustion process increases significantly up to more than 90 bar as the injection timing advances, and the appearance of the peaks become earlier from 20°CA to around TDC.

The performance under various injection timings can also be illustrated by the Heat Release Rate (HRR) and in-cylinder temperature in detail. As shown in Figure 7, with an advanced injection timing, all the HRR curves and the CA50 (the crank angle where 50% fuel energy is released) are advanced to a proper and finite crank angle period. This is a benefit to enhance the engine power because the heat supply can be ensured in an approximate constant space and occurs much faster, which is close to an idea constant-volume process and helps to increase the pressure at the initial stage of combustion. Figure 8 shows that the maximum in-cylinder temperature also gets an apparent improvement from 1433.8 K to 1565.6 K with the advance of injection timing. The effect of the relatively higher temperature is beneficial to speed up the burning rate. Thus a larger fraction of fuel’s energy can be transferred at the start of expansion to the piston as work.

**Recovering power by optimising intake charge temperature**

Regarding the practical operation of diesel engines, the intake charge temperature can be changed by

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**Figure 6.** Effect of injection timing on brake power and cylinder pressure.

**Figure 7.** Effect of injection timing on HRR and CA50.

**Figure 8.** Effect of injection timing on in-cylinder temperature.
employing an intercooler to increase intake density, leading to an improvement for the volumetric efficiency. Hence, to estimate and analyse the potential of the intake air temperature on recovering the engine power under OFC, varying intake charge temperature on brake power and combustion characteristics is carried out in this simulation study. Meantime the intake mass should be kept constant by adjusting the intake pressure when the intake temperature changes during the simulation.

Figure 9 shows that with the increase of intake temperature, the power shows a slight reduction from 28.4 kW to 27.9 kW under the operating condition of −12°C injection timing, and a 0.9 kW reduction would occur under the condition of −22°C injection timing. The power almost remains constant under the operating conditions of 283 K and 298 K. Therefore, 298 K is selected of the two as the intake temperature in the following sections of this simulation, because it is the normal room temperature which is easy to be implemented and more suitable for heating the engine.

With the increase of temperature, the following factors would have a negative influence on the brake power. Firstly, in order to introduce the same amount of intake mass, the intake pressure should be enhanced due to the reduction of intake density. Therefore, as shown in Figure 9, the cylinder pressures at Inlet Valve Closed (IVC) timing shows a slight increase by 0.159 bar and 0.163 bar under the conditions of −12°C and −22°C injection timing, respectively. Secondly, increasing the intake air temperature can advance the combustion process, owing to accelerate the evaporation and atomisation of fuel droplets. Hence, as shown in Figure 10, the peak of HRR is advanced by several degrees. Besides, CA50 is undesirably located in the angles of the compression stroke, which has a negative effect on the enhancement of HRR.

**Recovering power by optimising intake charge components**

During the combustion process, the oxygen is utilised as an oxidiser which concentration affects the combustion process profoundly, including the ignition delay, flame speed and cylinder temperature. In this section, the fraction of CO2 is a variable factor ranging from 70% to 79.7% in volume to investigate the effect of optimising intake charge components on combustion performance and recovering power under OFC mode. Meanwhile, the oxygen fraction can be improved up to 30% in volume, whereas it does not represent a higher mass amount for oxygen consumption due to the fixed oxygen mass flow rate.

Figure 11 shows that as the CO2 fraction decreases from 79.7% to 72%, the brake power generally increases monotonously by approximate 1.8 kW, whereas the curves start to flatten out with the further reduction of CO2. The reason is related to the cylinder pressure at IVC timing which has declined by 0.335 bar and 0.327 bar for the condition of −12°C and −22°C injection timing, respectively. It is also caused by the variation of HRR and CA50 in Figure 12. As the decrease of CO2 fraction from 79.7% (‘20.3/79.7’ in the figure) to 70% (‘30/70’), even the curve is significantly postponed and CA50 increases from 2.32°C to 7.43°C, it is still in the reasonable range which can facilitate the complete combustion. Besides, the peak of HRR increases from 172.45 J/CA to 295.33 J/CA, leading to a promotion to the output of engine power. All of these factors contribute to enhancing the power
output even that the consumption of fuel and oxygen is fixed. Hence the optimised intake components for oxygen and CO₂ is selected at 28% and 72% in volume.

**Recovering power by optimising engine compression ratio**

For a diesel engine, as compression ratio increases, the thermal efficiency of the engine can be improved as well. As the compression ratio generally ranges up to 22 for commercial conventional diesel engines currently, the compression ratio is set up with 17, 19.5 and 22 in this simulation research. Also, the optimal injection timing is likely to be slightly varied under different compression ratios, so a variety of timings should be investigated in this section for better recovering the engine power. As shown in Figure 13, the enhanced power by increasing the compression ratio can be a considerable value of 3.5 kW under the conditions of −7°C injection timing, whereas it just improves by 0.5 kW as the injection timing increases to −17°C. Moreover, it is notable that the maximum brake power is usually achieved at the compression ratio of 22 except the condition of −22°C injection timing. The exception with −22°C injection timing can be explained by the related HRR and CA50, which is quite advanced by the increased compression ratio. The start of heat release can be advanced by 10°C, which would result in an offset to the boost of brake power by the increasing heat losses during the compression stroke. To analyse the apparent gain by increasing compression ratio under the injection timing of −7°C, Figure 14...
illustrates that the peak of cylinder pressure is enhanced significantly from 53.16 bar to 94.93 bar as the increase of compression ratio. Furthermore, the heat release is still kept in the proper range, which peak is several CA after TDC. Therefore, the positive effect of increasing compression ratio to recover the power of diesel engines under OFC mode is limited by the range of injection timing. Therefore, as the maximum power can be obtained by the compression ratio of 22 accompanied with the injection timing of $-7^\circ$/CA and $-12^\circ$/CA, these parameters are kept in the following sections of this simulation.

Recovering power by utilising water injection strategy

Recent years, water injection strategy has been an advanced technology to inhibit knock by reducing the in-cylinder temperature of IC engines. The constant pressure process can also be extended as the result of the water evaporation at the beginning of the power stroke. Furthermore, water injection strategy has also been adopted as a useful measure to optimise the combustion process for oxy-fuel engines.

In this section, the effect of water injection strategy to recover engine power is explored in detail. Water liquid with an initial temperature of 298 K is directly injected into cylinders. To better understand the influence extent of water injection mass and timing, the injection mass ranges from 1 mg to 5 mg per cylinder per working cycle, and the succeeding part for water injection timing ranges from $-7^\circ$/CA to $-40^\circ$/CA.

In Figure 15, it is observed that the brake power can be enhanced by increasing the water injection mass with a fix injection timing at $-100^\circ$/CA. By this way, not only more working medium is added, but also burn rate is also accelerated owing to that both the oxygen concentration and the flame propagation process are changed. However, the influence extent is not very obvious, which only has a slight increment by 0.9 kW and 1.1 kW as the water injection increases from 0 mg to 5 mg under the condition of $-7^\circ$/CA and $-12^\circ$/CA fuel injection timing, respectively. It is because the excessive injected water can lead to a slight positive effect on cylinder pressure and heat release rate, as shown in Figures 16 and 17. Meanwhile, the phasing of combustion cannot be easily changed by water injection. Under the cases of 5 mg water injection, CA50 is just postponed by 2.18 $/\circ$/CA. Besides, as the maximum 32.8 kW is obtained under this condition of $-12^\circ$/CA injection timing, so it was selected in the following sections about recovering the engine power.
Unlike the tendency with increasing water injection mass, the effect of different water injection timing on engine power is a bit complicated, as shown in Figure 18. As the water injection timing postpones from $-120^\circ$CA to $-80^\circ$CA, the brake power keeps broadly stable. With a further delay, the brake power commences a gradual rise to 33.5 kW at $-50^\circ$CA then it decrease rapidly to 31.5 kW at $-40^\circ$CA. The changes tendency with water injection timing can be mainly attributed to the two aspects. Firstly, owing to the postponed from $-120^\circ$CA to $-50^\circ$CA, it can ensure higher efficiency to utilise the injected water to increase the working fluid. However, in the case of $-45^\circ$CA and later injection timings, the duration of water injection will interrupt the fuel atomisation, leading a strong negative effect for the early stage of combustion process which displays an abnormal in the curve of cylinder pressure in Figure 19. This instability characteristic had also been demonstrated in the experimental study of the existing publication.26 Secondly, as shown in Figure 20, the combustion phasing is postponed with the delay of water injection timing, which leads to a slight reduction of the peak of cylinder pressure in Figure 19.

**Comparison between OFC of increasing lambda$_{O2}$ and CAC**

Based on the study roadmap presented in Figure 2, in order to obtain the same power out with the CAC, the final step is increasing the intake mass, leading to the growth of lambda$_{O2}$ with 0.1 intervals until the power is not less than 40 kW. The effect of lambda$_{O2}$ on recovering power under OFC is in shown Figure 21. As the
lambdaO₂ increases from 1.0 to 1.5, the brake power has significantly improved from 33.5 kW to 40 kW, even though other operating parameters keep constant. It is regarded as the eventual solution to recovery engine power in this simulation research. With the increasing amount of intake charge, the effective value of adiabatic index during the expansion process will be increased. It will enhance the fuel conversion efficiency at a given expansion ratio, resulting in higher expansion stroke work for per unit mass of fuel.

Figures 22 and 23 illustrate the comparison between OFC and CAC, including brake power, CA50 and HRR. The ‘air’, ‘oxy’, ‘O-1.0’, ‘O-1.5’ is introduced to represent the previous simulation case of conventional air combustion, initial OFC (replacing the N₂ with CO₂), optimised OFC (lambdaO₂ = 1.0) and final OFC (lambdaO₂ = 1.5), respectively. It can be found that CA50 is significantly delayed from 7.18°Cc to 15.48°Cc when the N₂ is replaced by CO₂, which seriously affects the peak of cylinder pressure and power output. However, the engine power obtains an apparent recovery by using all the proposed optimisation solution in the technical roadmap, achieving 40 kW by increasing lambdaO₂ eventually.

Conclusion

This research is about a feasibility study for the implementation of OFC on a practical diesel engine at the economical oxygen-fuel ratios using 1-D simulation. It is a research conducted part of the project RIVER (funded by the Interreg North-West Europe), to promote the design of one OFC diesel engine coupled with CCS technology to achieve zero carbon emissions from IW vessels.

In this simulation, when the engine is modified to the OFC, it achieves the equivalent power output with the CAC mode by utilising some technical measures. Moreover, the consumption of oxygen should be saved as much as possible during the oxy-combustion process. For this purpose, various parameters including fuel injection timing, intake charge temperature, intake components, engine compression ratio and water injection strategy, have been used and optimised for the OFC engine’s power performance. The brake power of oxy-fuel combustion is only 26.7 kW that has a noticeable decline compared with 40 kW of conventional air combustion with fixed consumption of fuel and oxygen. Based on the roadmap of this study, after optimising a lot of valuable parameters, the obtained maximum brake power is 33.5 kW, which is still 6.5 kW less than the conventional air combustion. However, a benefit of 6.8 kW has been acquired from this optimisation method. Afterwards, the increasing lambdaO₂ from 1.0 to 1.5 can bring remarkable improvement to brake power, resulting in the same power with the conventional air combustion. In the meanwhile, some significant conclusions of this research can be summarised as follows:

1. The brake power has grown continuously from 25.2 kW to 29.0 kW when the injection timing advances from −2°CA to −22°CA. With a retarded injection timing, the peaks usually present decrease trends for all curves of cylinder pressure, HRR and cylinder temperature.
2. Within a certain range, decreasing intake temperature or CO₂ fraction can be a helpful way to recover the engine power.
3. The effect of increasing compression ratio on power output would be limited by the range of injection timing. Because under the conditions

![Figure 22. Comparing of brake power and CA50 between oxy-fuel combustion and conventional air combustion.](image1)

![Figure 23. Comparison of HRR between oxy-fuel combustion and conventional air combustion.](image2)
of early injection timings, there would be an offset by the enhanced heat losses during the compression stroke.

4. The brake power can be recovered by increasing the water injection mass, while this effect is very limited. Besides, within a suitable range of water injection timing, the brake power can be enhanced.

5. By increasing \( \lambda_{O2} \) from 1.0 to 1.5, which is selected as the final solution due to enhance the oxygen consumption, can significantly improve engine power from 33.5 kW to 40 kW, even though other operating parameters are unchanged.

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References

1. Kellogg WW. Climate change and society: consequences of increasing atmospheric carbon dioxide. Routledge, 2019.

2. Pike H. Doctors call on UK to be carbon neutral by 2030. BMJ (Online) 2019; 364.

3. Stott R, Arulkumaran S, Gilmore I, et al. Legislating for carbon net zero by 2030. Lancet 2019; 393: 981.

4. Bouzalakos S and Mercedes M. Overview of carbon dioxide (CO2) capture and storage technology //Developments and innovation in carbon dioxide (CO2) capture and storage technology. Woodhead Publishing, 2010, pp.1–24.

5. Figueroa J D, Fout T, Plasynski S, et al. Advances in CO2 capture technology—the US Department of Energy’s Carbon Sequestration Program. Int J Greenh Gas Control 2008; 2: 9–20.

6. NASA. https://climate.nasa.gov/ (2020, accessed 29 June 2020).

7. RIVER – Non-Carbon River Boat Powered by Combustion Engines. Interreg NWE, https://www.nweurope.eu/projects/project-search/river-non-carbon-river-boat-powered-by-combustion-engines/ (2017, accessed 25 June 2020).

8. Huang X, Guo J, Liu Z, et al. Opportunities and challenges of oxy-fuel combustion. Oxy-fuel combustion. Academic press, 2018, pp.1–12.

9. Cooper JR and Dooley RB. IAPWS release on surface tension of ordinary water substance. In: International Association for the Properties of Water and Steam (IAPWS), Charlotte, NC 2, September 1994. International Association for the Properties of Water and Steam (IAPWS).

10. Chun B-S and Wilkinson GT. Interfacial tension in high-pressure carbon dioxide mixtures. Ind Eng Chem Res 1995; 34: 4371–4377.

11. Stephen R. An introduction to combustion: concepts and applications. McGraw-hill, 1996.

12. Lemmon EW, McLinden MO and Friend DG. NIST chemistry webbook: thermophysical properties of fluid systems. Gaithersburg MD: National Institute of Standards and Technology, 2005, p. 20899.

13. Incropera FP, DeWitt DP, Lavine A, et al. Fundamentals of heat and mass transfer. Wiley, 2007.

14. Wall T, Liu Y, Spero C, et al. An overview on oxyfuel coal combustion—state of the art research and technology development. Chem Eng Res Des 2009; 87: 1003–1016.

15. Cengel Y. Heat and mass transfer: fundamentals and applications. McGraw-Hill Higher Education, 2014.

16. Shaw R and Oman H. Non-air working fluids for closed-cycle diesel engines. In: Proc., Intersoc. Energy Convers. Eng. Conf.;(United States), 1983, vol. 2, no. CONF-830812-. Seattle, Washington: The Boeing Company.

17. Reader GT and Hawley JG. Closed and re-cycled engine development for submersible power plants (submarines). In: Proceedings of the 24th Intersociety Energy Conversion Engineering Conference, 1989. IEEE.

18. Hawley JG, Ashcroft SJ and Patrick MA. The effects of non-air mixtures on the operation of a diesel engine by experiment and by simulation. Proc Inst Mech Eng 1998; 212: 55.

19. Wu HW and Shu CT. Effects of operating parameters on steady and transient behaviors of a closed cycle diesel engine. Energy Convers Manag 2006; 47: 2070–2080.

20. Wu HW, Wu ZY, Yang JY, et al. Combustion characteristics of a closed cycle diesel engine with different intake gas contents. Appl Therm Eng 2009; 29: 848–858.

21. Wu HW, Wang RH, Chen YC, et al. Influence of port-induced ethanol or gasoline on combustion and emission of a closed cycle diesel engine. Energy 2014; 64: 259–267.

22. Bilger RW. Zero release combustion technologies and the oxygen economy. In: The fifth international conference on technologies and combustion for a clean environment, Lisbon, Portugal, 1999, pp.12–15. Centro Cultural de Belém, Lisbon–Portugal: Instituto de Comustão.

23. Bilger RW and Wu Z. Carbon capture for automobiles using internal combustion Rankine cycle engines. J Eng Gas Turbines Power 2009; 131; 034502.

24. Yu X, Wu Z, Fu L, et al. Study of combustion characteristics of a quasi internal combustion Rankine cycle engine. SAE Technical Paper, 2013.
25. Fu L, Yu X, Deng J, et al. Development of internal combustion Rankine cycle engine test system (In Chinese). *Chin Intern Combust Engine Eng* 2013; 6: 87–92.

26. Fu L, Deng J, Yu X, et al. Experimental study of effect of water injection on combustion stability in internal combustion Rankine cylinder engine (In Chinese). *Chin Intern Combust Engine Eng* 2014; 35: 38–45.

**Appendix**

**Abbreviations**

| Abbreviation | Definition                                           |
|--------------|------------------------------------------------------|
| CA50         | the crank angle where 50% fuel energy is released    |
| CAC          | Conventional Air Combustion                          |
| CCDE         | Closed Cycle Diesel Engine                           |
| CCS          | Carbon Capture and Storage                           |
| CI           | Compression Ignition                                 |
| CO₂          | Carbon Dioxide                                       |
| EGR          | Exhaust Gas Recirculation                            |
| EU           | European Union                                       |
| GHG          | Greenhouse Gas                                        |
| HRR          | Heat Release Rate                                    |
| IC           | Internal Combustion                                  |
| ICRC         | Internal Combustion Rankine Cycle                    |
| IVC          | Inlet Valve Closed                                   |
| IW           | Inland Waterway                                      |
| OFC          | Oxy-Fuel Combustion                                  |