TECHNO-ECONOMIC ANALYSIS OF PRESSURIZED OXY-FUEL COMBUSTION POWER CYCLE FOR CO₂ CAPTURE

by

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ABSTRACT

Growing concerns over greenhouse gas emissions have driven extensive research into new power generation cycles that enable carbon dioxide capture and sequestration. In this regard, oxy-fuel combustion is a promising new technology for capturing carbon dioxide in power generation systems utilizing hydrocarbon fuels. Combustion of a fuel in an environment of oxygen and recycled combustion gases yields flue gases consisting predominantly of carbon dioxide and water. To capture carbon dioxide, water is condensed, and carbon dioxide is purified and compressed beyond its supercritical state. However, conventional atmospheric oxy-fuel combustion systems require substantial parasitic energy in the compression step within the air separation unit, a flue gas recirculation system and carbon dioxide purification and compression units. Moreover, a large amount of flue gas latent enthalpy, which has high water concentration, is wasted. Both lower the overall cycle efficiency. Alternatively, pressurized oxy-fuel combustion power cycles have been investigated. In this thesis, the analysis of an oxy-fuel combustion power cycle that utilizes a pressurized coal combustor is reported. We show that this approach is beneficial in terms of larger flue gas thermal energy recovery and smaller parasitic power requirements. In addition, we find the pressure dependence of the system performance to determine the optimal combustor operating pressure for this cycle. We calculate the energy requirements of each unit and determine the pressure dependence of the water-condensing thermal energy recovery and its relation to the gross power output. Furthermore, a sensitivity analysis is conducted on important operating parameters including combustor temperature, Heat Recovery Steam Generator outlet temperature, oxygen purity and oxygen concentration in the flue gases. A cost analysis of the proposed system is also conducted so as to provide preliminary cost estimates.

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Chapter 1 Introduction

1.1. Motivation

Most current energy conversion systems utilize fossil fuels. Given their availability, energy density, and relatively low costs, fossil fuels account for about 80% of total U.S. energy supplies and about 90% of worldwide energy supplies [1-3]. Among fossil fuels that include coal, natural gas and oil, coal has been a primary energy source of electricity generation for many decades because of its availability and low costs compared to the others. According to estimates by Energy Information Administration, coal accounts for over 50% of the electricity generated in the United States [2]. Coal, which is cheap and abundant, provides primary energy at a price that is the half of the cost of oil and natural gas [4]. Many researchers and policy makers agree that coal will maintain its important role in power generation in the near future [5-7].

Coal combustion emits large amounts of carbon dioxide, believed to be a significant contributor to global warming. As energy use grows, the amount of carbon dioxide that is emitted to the environment increases considerably. The Energy Information Administration estimates that world carbon dioxide emissions from energy production will increase by 51% by 2030, from 28.1 billion metric tons in 2005 to 42.3 billion metric tons in 2030 [3].

Concerns over global warming may lead to imposing limits on greenhouse gas emissions from fossil fuel plants such as coal-fired power plants. This has stimulated
extensive research into reducing carbon dioxide emissions from fossil energy sources. For the past decade, scientists and investigators have endeavored to find a way to decrease carbon dioxide emissions from coal-fired power plants. Carbon dioxide capture and sequestration technologies shown in Figure 1-1 should play a critical role in this regard, enabling coal to continue to satisfy an increasing fraction of worldwide energy needs. The International Energy Agency estimates that carbon capture and sequestration could play an important role in decreasing carbon dioxide emissions [8]. To achieve a deep reduction in carbon dioxide emissions through carbon capture and sequestration within power generation systems, several technologies are being investigated, one of which is oxy-fuel combustion.

Figure 1-1 Carbon capture and sequestration options
1.2. Oxy-Fuel Combustion

In oxy-fuel combustion, fuels are burned in a nitrogen-lean and carbon dioxide-rich environment, which is achieved by feeding the combustor with an oxygen-rich stream and recycled flue gases. The oxygen-rich stream is produced by an air separation unit that is based on cryogenic distillation. The recycled gases are used to control the flame temperature and replace the nitrogen separated prior to combustion [9]. Oxy-fuel combustion yields flue gases consisting of predominantly carbon dioxide and condensable water, whereas conventional air-fired combustion flue gases are nitrogen-rich with only about 15% (by volume) of carbon dioxide [10, 11]. The high carbon dioxide concentration and the significantly lower nitrogen concentration in the oxy-fuel raw flue gases is a unique feature that lowers the energy and capital costs of oxy-fuel carbon dioxide capture when compared to alternatives [12]. Table 1-1 describes the features of oxy-fuel combustion.

Table 1-1 Features of oxy-fuel combustion

<table>
<thead>
<tr>
<th>Features</th>
<th>Combustion Environment and Flue gases</th>
<th>Heat Transfer</th>
<th>Emissions</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>• Higher combustion temperature requiring the recirculation of the flue gases</td>
<td>• Higher gas emissivity and gas thermal capacities</td>
<td>• Nearly zero NO\textsubscript{x} emission</td>
</tr>
<tr>
<td></td>
<td>• Denser flue gases with tri-atomic molecules</td>
<td>• Change in the heat transfer rate</td>
<td>• Lower SO\textsubscript{x} emission</td>
</tr>
<tr>
<td></td>
<td>• Easier to capture CO\textsubscript{2}</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
1.2.1. Characteristics of Oxy-Fuel Combustion and Its Flue Gases

The elimination of nitrogen prior to combustion provides an environment of high combustion temperature. In conventional air-fired power plants, nitrogen accounts for 79% of the feed air entering into combustors or boilers. Because nitrogen is not a reactant in the combustion process, it simply absorbs and transfers the thermal energy converted from the chemical energy of fuels through combustion. However, in oxy-fuel combustion, nitrogen is separated prior to combustion, and thus, combustion takes place with higher concentration of oxygen. Stoichiometric combustion of a fuel in an environment with pure or nearly pure oxygen gives a combustion temperature of about 3500 °C [13]. Current typical power plant materials cannot sustain this high temperature. Therefore, the recirculation of the flue gases is required in oxy-fuel combustion to reduce the combustion temperature to the reasonable level. To obtain a similar adiabatic combustion temperature to the air-fired case, nearly 60% (mass basis) of the flue gases have to be recycled. For coals evaluated at CANMET, the flame characteristics including the combustion temperature and heat flux, comparable to those of air-fired combustion, can be achieved by 30% of oxygen concentration in the feed gas with the rest being the recycled flue gases [12].

Oxygen and carbon dioxide enriched environment within the oxy-fuel combustor produces denser flue gases with the high proportions of tri-atomic molecules. The molecular weight of carbon dioxide is 44, whereas that of nitrogen is 28. Consequently, the oxy-fuel combustion requiring a great amount of the recycled flue gases yields denser flue gases enriched with oxygen and carbon dioxide, as shown in Table 1-2. In this regard, the volume of the flue gases decreases by nearly 80% [12], which leads to employing
smaller equipment within power plants. In addition, the combustion environment enriched with tri-atomic molecules, oxygen and carbon dioxide, produces higher gas emissivity and heat capacity.

<table>
<thead>
<tr>
<th></th>
<th>Conventional Air-Fired Combustion</th>
<th>Oxy-Fuel Combustion</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flow Rate (kg/s)</td>
<td>809.38</td>
<td>763.49</td>
</tr>
<tr>
<td>Density (kg/m³)</td>
<td>0.8</td>
<td>0.977</td>
</tr>
<tr>
<td>Molecular Weight</td>
<td>29.74</td>
<td>36.95</td>
</tr>
</tbody>
</table>

Combustion without nitrogen gives a better condition for carbon dioxide capture from the flue gases. While the conventional air-fired combustion yields the flue gases in which nitrogen dilutes the carbon dioxide concentration, the oxy-fuel combustion flue gases mainly consist of carbon dioxide and water. Figure 1-2 shows the chemical composition (mole fraction) of the flue gases of each case. As seen in Figure 1-2, nearly 90% of the oxy-fuel combustion flue gases are water and carbon dioxide, whereas the flue gases of air-fired combustion predominantly consist of nitrogen. Thus, without the cost and energy intensive amine stripping process typically adopted in the air-fired system, oxy-fuel combustion leads to easier carbon dioxide capture by condensing out water.
1.2.2. Heat Transfer

The heat transfer rate of the oxy-fuel combustion flue gases is affected by two primary properties that vary during oxy-fuel combustion: gas radiative properties and gas thermal capacities [12]. As mentioned in the previous section, the high concentration of water and carbon dioxide in the flue gases leads to an increase in the overall gas emissivity. In combustion, the heat transfer from a flame is dominated by thermal radiation from water vapor, carbon dioxide, soot and carbon monoxide [15]. Thus, the increase in the concentration of water and carbon dioxide changes the radiative heat transfer rate in oxy-fuel combustion. Moreover, higher thermal capacities of water and carbon dioxide enable the oxy-fuel system to increase the convective heat transfer. On the other hand, the convective heat transfer rate decreases because of two factors, lower mass flow rate of the
flue gases shown in Table 1-2 and lower gas temperature entering the convective section of the boiler due to the increased radiative heat transfer.

1.2.3. Emissions

In oxy-fuel combustion, the amount of nitric oxide emissions is reduced to less than one-third of emissions from air-fired combustion [12]. Most of all, the significantly lower concentration of nitrogen within the combustor enables the oxy-fuel system to decrease thermal NO\textsubscript{x} formation. As shown in Figure 1-2, nitrogen concentration in the oxy-fuel combustion system decreases nearly tenfold. This fact leads to lower NO\textsubscript{x} emissions compared to air-fired combustion. Figure 1-3 explains other important reasons why oxy-fuel combustion has negligible NO\textsubscript{x} emissions. Based on the low NO\textsubscript{x} emission level, NO\textsubscript{x} removal processes that consume a large amount of energy and cost can be eliminated in oxy-fuel combustion.

Compared to the NO\textsubscript{x} emission level, the SO\textsubscript{2} emission rate does not change considerably, but varies greatly inside the combustor. In oxy-fuel combustion, the flue gases achieve higher SO\textsubscript{3} concentration, and this particular environment results in sulphur detention. The conversion rate of sulphur to SO\textsubscript{2} drops from 91% (air-fired case) to 64% (oxy-fuel combustion) [16]. As a result, the amount of work duty of SO\textsubscript{x} removal processes can be lowered.
50-80%: chemical reduction of recycled NO to molecular N\textsubscript{2} by the reburning mechanism in combustor

10-50%: decreased NO\textsubscript{x} formation rate by the reduction of fuel-N to N\textsubscript{2}, promoted by recycled NO

0-10%: inhibited formation of NO via a char catalyzed reaction by recycled CO

High SO\textsubscript{2} concentrations in the flue gas: result in sulphur retention by ash or deposits in the furnace

Higher concentration in the flue gas: may have increased potential for material corrosion

Figure 1-3 Variation in the emissions from oxy-fuel combustion [16-18]

1.2.4. Benefits and Challenges

Table 1-3 summarizes the benefits and challenges of oxy-fuel combustion power plants. Compared to alternatives, the oxy-fuel combustion technology could be a promising option for zero-emission power plants in terms of the cycle efficiency and the cost of electricity. As shown in Figure 1-4, it has a smaller increase in the price of electricity and less energy penalty than the pre-combustion capture based on the amine absorption processes, achieving higher carbon dioxide capture rate. Reduced emissions and efficient carbon dioxide capture and sequestration lead to smaller work duty of the capture processes, and make it possible to lower energy and capital costs of oxy-fuel combustion capture.
Table 1-3 Benefits and challenges of oxy-fuel combustion systems

<table>
<thead>
<tr>
<th>Benefits</th>
<th>Challenges</th>
</tr>
</thead>
<tbody>
<tr>
<td>• Promising option for power plants with zero-emissions</td>
<td>• Significant efficiency penalty due to large air separation units and recycled flue gases</td>
</tr>
<tr>
<td>• Efficient capture and sequestration of carbon dioxide</td>
<td>• Optional SO$_x$ removal processes</td>
</tr>
<tr>
<td>• Potential to be retrofitted to existing power plants</td>
<td>• Scale-up issues since no large capacity oxy-fuel combustion power plants exist</td>
</tr>
<tr>
<td>• Better overall integration with the power island</td>
<td></td>
</tr>
<tr>
<td>• The least increase in the price of electricity compared to other carbon capture technologies</td>
<td></td>
</tr>
<tr>
<td>• Reduced emissions, in particular, NO$_x$</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>SC/Air</th>
<th>USC/Air</th>
<th>SC/Air</th>
<th>USC/Air</th>
<th>SC/Oxy</th>
<th>USC/Oxy</th>
</tr>
</thead>
<tbody>
<tr>
<td>Net Efficiency (HHV, %)</td>
<td>Cost of Electricity (Cents/kWh)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>39.5</td>
<td>6.29</td>
<td>44.6</td>
<td>6.45</td>
<td>27.2</td>
<td>11.44</td>
</tr>
</tbody>
</table>

Figure 1-4 Net efficiency and the cost of electricity of power plants with and without carbon dioxide capture [14]
Another important advantage that the oxy-fuel combustion system has is the possibility to retrofit existing power plants. Because the oxy-fuel cycle utilizes the conventional equipment within the existing power plants, other than the air separation units and the carbon dioxide purification units, the system can be effectively integrated with the current power plants. Because the construction of new power plants requires a large amount of capital investment, and existing power plants have a long lifetime, the feature of easier retrofit makes oxy-fuel combustion systems more promising than other alternatives.

However, the oxy-fuel cycle has a significant efficiency penalty because it needs large capacity air separation units and a great amount of the recycled flue gases. Oxy-fuel combustion requires more than 5000 metric tons per day of oxygen to fit in industrial-scale (over 300 MW_e) plants [19]. To meet the needs of large amounts of oxygen required, typical oxy-fuel combustion systems are based on the energy intensive air separation units using cryogenic distillation columns. The energy and capital cost intensive cryogenic separation is the only option through which the current technologies can produce large amounts of oxygen. The cycle efficiency of oxy-fuel combustion systems is reduced by 5~7 percentage points because of the air separation units [12]. In addition, the recirculation of the flue gases to moderate the combustion flame temperature consumes a great amount of power. By passing through the steam generation units, the flue gases experience pressure drops. Thus, recirculation system requires auxiliary power to compensate for the pressure drop to send it back to the oxy-fuel combustor. This accounts for nearly 4% of the efficiency penalty in the oxy-fuel combustion systems. Consequently, a total reduction in the cycle efficiency of 9~13 percentage points from the conventional
cycle efficiency occurs in the oxy-fuel cycles [12].

Moreover, the fact that no large capacity power plants based on oxy-combustion technologies exist may produce unforeseen technical problems when we scale up the system. Thus, scale-up issues should be carefully investigated in order to utilize the oxy-fuel combustion system as a promising carbon capture and sequestration option.

1.3. Pressurized Oxy-Fuel Combustion

Although it benefits from the numerous advantages explained in the previous section, the conventional atmospheric oxy-fuel combustion system has a significant efficiency penalty compared to the existing power plants. Oxy-fuel combustion technologies would not be promising enough to be executed as a viable power plant option, unless this issue is resolved.

Recent research shows that pressurized oxy-fuel combustion systems (Figure 1-5) have the potential of better performance over conventional atmospheric oxy-fuel combustion power cycles. ENEL suggests that oxy-fuel combustion at high pressures may increase the burning rate of char and the heat transfer rates in the convective sections of the heat transfer equipment. To demonstrate these benefits, ENEL started in 2006 a series of experimental activities on a patented pressurized coal-combustion technology, ISOTERM®, and has already performed several tests on a 5 MWth scale, working at 4 bars [20-22]. The pressurized combustion technology achieves high carbon dioxide purity in the flue gases and reduces the energy penalties [23].
CANMET and ThermoEnergy also conducted technical and economic studies on the pressurized oxy-fuel combustion system [24-30]. Their approach shows net efficiency gain, and reduction in the capital cost and the cost of electricity when using high pressure oxy-fuel combustion. By operating at elevated pressure, the cycle recovers larger fraction of thermal energy of the flue gases to generate steam and lowers the heat duty in the regeneration system. As a result, the cycle efficiency increases by producing more gross power through turbines. Moreover, the pressurized flue gases utilize smaller equipment within power plants, and hence the pressurized oxy-fuel system may save in capital investment.

Figure 1-5 Process layout for the pressurized oxy-fuel combustion power cycle
Based on the features of oxy-fuel combustion and higher combustor operating pressure, pressurized oxy-fuel systems are expected to achieve better performance. Operating the system at elevated pressure makes the oxy-fuel combustion cycle more competitive and sustainable by reducing the efficiency penalty and capital costs. Pressurized oxy-fuel systems are novel and have not been yet studied rigorously. Therefore, more examination is required to understand and improve the pressurized oxy-fuel combustion power system.

1.4. Conclusions

Fossil fuel consumption is expected to grow continuously in the near future. Many of existing power plants are based on fossil fuels, and the increasing needs for energy promote the construction of more fossil-fuel fired power plants. However, combustion of fossil fuels emits considerably large amounts of carbon dioxide, a significant contributor to global warming. Accordingly, concerns over global warming impose regulations on fossil-fuel fired power plants, and research to reduce carbon dioxide emissions has been active. For this purpose, carbon capture and sequestration has been proposed: one approach for the former is oxy-fuel combustion.

By burning a fuel in an environment of pure or near pure oxygen and recycled flue gases, oxy-fuel combustion yields flue gases rich in water and carbon dioxide. The characteristic of high carbon dioxide concentration and near-zero nitrogen in the flue gases enables the oxy-fuel system to become a promising option for fossil-fuel fired power plants.
with carbon dioxide capture. Nevertheless, the oxy-fuel combustion system suffers from a large efficiency penalty because it uses energy intensive processes including air separation and recirculation of flue gases.

Pressurized oxy-fuel combustion is proposed as an alternative to atmospheric combustion. High operating pressure leads to increased thermal energy recovery from the flue gases and larger power output. In addition, denser flue gases require smaller but sturdier equipment. Consequently, pressurized oxy-fuel system may achieve an efficiency gain and reduce capital costs.

In Chapter 2, the base-case cycle used in this thesis and its analysis are explained. Based on the base-case cycle analysis, the performance of a pressurized oxy-fuel combustion power cycle with increasing operating pressures is discussed in Chapter 3. A sensitivity analysis performed to determine the impacts of other important parameters including combustor temperature, Heat Recovery Steam Generator outlet temperature, oxygen purity and oxygen concentration in the flue gases is reported in Chapter 4. The cost analysis method and preliminary cost estimates are discussed in Chapter 5. Chapter 6 includes conclusions.
Chapter 2 Base-Case Analysis

2.1. Overview

We analyze the oxy-fuel combustion power cycle based on a pressurized coal combustor and compare it with a case utilizing an atmospheric pressure oxy-fuel combustor. The analysis is conducted based on the base-case cycle of pressurized oxy-fuel combustion that is developed so as to employ commercially available technologies or processes. In order to find out the strengths and weaknesses of the pressurized system, the base-case cycle is compared to the conventional atmospheric oxy-fuel system.

To operate this high combustion pressure system, a high pressure deaerator and a flue gas acid condenser are required. The acid condenser adopted is based on a new commercially available technology. While not typical in the power industry, it is common in the process industry. The acid condenser is modified to work at high pressure with flue gas composition seen in oxy-combustion. In addition, the proposed oxy-fuel combustion power cycle purifies and compresses the concentrated carbon dioxide flue gas stream to 110 bars. The flue gas purification process includes de-SO\textsubscript{x}, de-NO\textsubscript{x}, and a low temperature flash unit to prepare the carbon dioxide stream for transportation to an EOR or a sequestration site.

We show that, because of the raised dew point and the corresponding available latent enthalpy in the raw flue gases, the pressurized oxy-fuel system can recover more thermal energy from the flue gases and eliminate the bleeding from the high-pressure and
the low-pressure steam turbines. Consequently, the cycle efficiency for the pressurized oxy-fuel system is superior to the atmospheric system.

2.2. Methodology

Two commercial simulation packages, Thermoflex® and Aspen Plus®, ¹ are used to model the pressurized oxy-fuel combustion power cycle. Thermoflex® is a good simulation tool for the analysis of power plants including conventional steam cycles, combined cycles and hybrid cycles. It contains detailed model libraries for standard and commercially available components within power plants. Thus, in our base-case analysis, Thermoflex® focuses more on modeling the steam generation unit and the power island. On the other hand, Aspen Plus® has strengths in modeling chemical processes, conceptual design and non-standard components such as distillation columns and low temperature flash units. As general process simulation software, Aspen Plus® gives users chances to model non-standard components. Therefore, it is used to model the air separation unit and the flue gas treatment unit including the carbon dioxide purification and compression unit.

Figure 2-1 shows the thermodynamic analysis method using the two simulation packages, Thermoflex® and Aspen Plus®. Based on a fixed oxygen purity target of 95% and the mass flow rate of oxygen needed in the oxy-coal combustor that is explained in Section 2.3.1, the power consumption and the heat balance of the air separation unit are

¹ Thermoflex® and Aspen Plus® are registered trademarks of Thermoflow LTD and Aspen Technology, Inc., respectively.
estimated, using Aspen Plus®. According to the oxygen stream information from Aspen Plus® and the coal analysis data given in Table 2-1, Thermoflex® is used to analyze the oxy-coal combustor, the steam generation unit, and the power island. The flue gas stream information is brought back to Aspen Plus® to investigate the performance of the flue gas treatment unit. This approach is applied to a coal-fired power plant with the fixed coal flow rate of 30 kg/s corresponding to 874.6 MW_{th} (HHV) or 839.1 MW_{th} (LHV).

Figure 2-1 Integration method of two simulation packages for the thermodynamic analysis of the pressurized oxy-fuel combustion power cycle
Table 2-1 Coal analysis data

<table>
<thead>
<tr>
<th>Composition</th>
<th>(%) w/w</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Proximate Analysis</strong></td>
<td></td>
</tr>
<tr>
<td>Moisture</td>
<td>6.4</td>
</tr>
<tr>
<td>Ash</td>
<td>7.0</td>
</tr>
<tr>
<td>Volatile Matter</td>
<td>33.1</td>
</tr>
<tr>
<td>Fixed Carbon</td>
<td>53.5</td>
</tr>
<tr>
<td><strong>Ultimate Analysis</strong></td>
<td></td>
</tr>
<tr>
<td>Carbon</td>
<td>71.1</td>
</tr>
<tr>
<td>Hydrogen</td>
<td>4.7</td>
</tr>
<tr>
<td>Moisture</td>
<td>6.4</td>
</tr>
<tr>
<td>Ash</td>
<td>7.0</td>
</tr>
<tr>
<td>Sulfur</td>
<td>0.5</td>
</tr>
<tr>
<td>Nitrogen</td>
<td>1.2</td>
</tr>
<tr>
<td>Oxygen</td>
<td>9.1</td>
</tr>
<tr>
<td>Chlorine</td>
<td>0.014</td>
</tr>
<tr>
<td>Fluorine (ppm)</td>
<td>34.6</td>
</tr>
<tr>
<td>HHV (kJ/kg)</td>
<td>29153</td>
</tr>
<tr>
<td>LHV (kJ/kg)</td>
<td>27971</td>
</tr>
</tbody>
</table>

The proposed approach focuses on the base-case of a 10 bars operating pressure combustor and compares it with the case of a near atmospheric pressure combustor. Comparison is mainly based on the overall net efficiency, the flue gas thermal energy recovery, the gross efficiency, and the parasitic power demand of each unit. Through the detailed cycle analysis, the thermodynamic parameters that impact the performance are discussed, and the improvement on the base-case is carefully investigated. Details of the modeling and the simulation results of both cases are discussed in the following sections.
2.3.  Modeling of Base-Case Cycle

Figure 2-2 shows the oxy-fuel combustion power cycle with a pressurized combustor. It consists of five primary units: 1) An air separation unit; 2) A pressurized coal combustor; 3) A steam generation unit; 4) A power island; 5) A carbon dioxide purification and compression unit.

![Figure 2-2 Overall process layout for oxy-fuel combustion power cycle utilizing a pressurized coal combustor (edited from [23])]}

As shown in Figure 2-2, the condensate leaving the condenser at state 1 is compressed by the first feedwater pump. Next, the pressurized condensate at state 2 enters the acid condenser where most of the latent enthalpy in the flue gases is recovered.
while the flue exhaust stream is cooled from state 20 to state 21. The condensate stream recuperates more thermal energy by cooling the combustor walls from state 3 to state 4 before entering the deaerator at 10 bars. In Figure 2-2, the water streams are in green. After the deaerator, the feedwater stream at state 5 is pumped to the supercritical state, by the second feedwater pump. Supercritical states are represented in violet, whereas subcritical states are shown in blue in Figure 2-2.

After leaving the second feedwater pump, the feedwater is heated regeneratively to state 6. Next, the feedwater enters the Heat Recovery Steam Generator (HRSG) where it is heated to 600 °C at 250 bars, state 7. Across the power island and the HRSG, there are two reheat streams, state 9 and state 11, to the intermediate-pressure turbines and the low-pressure turbines, respectively. These two streams are superheated to 620 °C. Besides, the steam is bled from the high-pressure steam turbine to be injected into the pressurized combustor, state 8, in order to atomize the slurry particles.

On the gas side, the oxygen stream from the air separation unit, state 13, is mixed with the recycled flue gases, state 19, and injected into the pressurized coal combustor as state 14. The combustor yields flue gases at about 1550 °C, state 15. The flue gas stream is cooled down to 800 °C by the recycled flue gases, state 18. Next, it enters the HRSG at state 16 and transfers thermal energy to the steam while being cooled down to state 17. The flue gases are recycled after the HRSG, and the rest of flue gases pass through the acid condenser. After the acid condenser, the flue gas stream is purified by the carbon dioxide purification and compression unit, and this unit yields the capture-ready stream of carbon dioxide and the exhaust stream, state 22 and state 23, respectively.
2.3.1. Air Separation Unit

Oxy-fuel combustion requires separation of nitrogen prior to combustion using an air separation unit. The three primary characteristics of air separation units employed in proposed oxy-fuel combustion power plants are large size, low oxygen delivery pressure and lower oxygen purity (85–98%) compared to the high purity (99.5–99.6%) required in the process industry [19]. A commercial-scale power plant based on oxy-fuel combustion needs thousands of tons of oxygen per day. Within the current technologies, cryogenic separation using distillation columns is the only available option to produce the large amount of oxygen required in the oxy-fuel power plant. Cryogenic separation processes are energy and cost intensive. Although the requirement of lower oxygen purity reduces the power requirement (see Figure 2-3), the air separation units consume a significant amount of energy.

![Normalized energy requirement of cryogenic air separation unit based on the oxygen purity (100%-energy at 97%-oxygen purity) [19]](image-url)
Because of the huge power requirement and its impact on the overall performance, the air separation unit is an important part of an oxy-fuel combustion power cycle. The air separation unit in the oxy-fuel combustion power cycle consumes more than 15% of the gross power output [14, 31-33]. The air compression work accounts for most of the air separation unit power consumption.

In this study, the air separation unit is based on cryogenic distillation. Based on a two distillation column system, the cryogenic air separation unit delivers an oxygen stream with 95% purity (by volume) at power consumption comparable to that of commercial air separation units. The specific energy of producing O\textsubscript{2} in the base-case is 0.245 kWh/kg-O\textsubscript{2}, while the commercial air separation units consume 0.247 kWh/kg-O\textsubscript{2} [14] or 0.244 kWh/kg-O\textsubscript{2} [34], as shown in Figure 2-4. Note that the reference values do not necessarily represent the most up-to-date technology of different gas producers.

![Figure 2-4 Specific energy of producing oxygen [14, 34]](image)
The air separation process is shown in Figure 2-5. The feed air is compressed up to 5.5 bars by a two-stage air compressor. For this basic two column cycle with heat integration between the bottom of the low-pressure column and the top of the high-pressure column, the required air pressure is simply dictated by the boiling temperatures of nitrogen and oxygen at the given pressures at the given purity level. In other words, in the high-pressure column we must condense the high-purity nitrogen at the high-pressure. The boiling temperature of nitrogen in the high pressure column must be less than that of 95% oxygen of the low pressure column since the heat from nitrogen must be used to boil oxygen.

![Air Separation Process Diagram](image)

*Figure 2-5 Process layout for the air separation unit*

After air compression, the pressurized air passes through the regenerator to eliminate impurities, which are water and carbon dioxide that must be removed before entering into cryogenic distillation columns. Steam-heated nitrogen is required for this
regeneration, but oxygen is lost via bed switchover and adsorption/desorption. More than 4% of the feed oxygen is lost during the regeneration process. Through the two distillation columns, an oxygen enriched stream with 95% oxygen, 4% argon and 1% nitrogen is produced at 1.24 bars. By compressing this stream using the two-stage oxygen compressor, the air separation unit delivers the pressurized gaseous oxygen-enriched stream at 10 bars to the combustor.

The oxygen delivery temperature to the combustor is controlled to prevent acid condensation when mixed with the recycled flue gases. The flue gases contain acid gases, such as SO$_3$, SO$_2$, NO$_x$, and HCl produced during combustion. As shown in Figure 2-2, the recycled flue gases, state 19, are mixed with the oxygen stream, state 13, which is colder. To avoid corrosion due to the condensation of these acid gases when mixed with the oxygen stream, the oxygen delivery temperature should be carefully controlled. The base-case model sets this temperature to nearly 200 °C. This temperature target is achieved by using a two-stage oxygen compressor with an intercooler.

The mass flow rate of the oxygen stream is determined such that the flue gases exiting the combustor contain 3% oxygen on a molar basis, as shown in Figure 2-6. Because the pressurized oxy-fuel system provides the combustion environment with higher burning rate of char, even 3% oxygen concentration in the flue gases is a conservative target [21, 30].

According to commercially available gas compressors, the polytropic efficiencies of the air compressors and the oxygen compressors are set to 85%.
2.3.2. Pressurized Coal Combustor

The pressurized coal combustor used in this work is based on the ISOTHERM® proposed by ITEA with the support of ENEL [21]. The combustion technology claims flameless combustion in which no flame front exists and the temperature of the reactor is completely uniform. As mentioned in Section 1.2.2, high concentration of tri-atomic molecules, water and carbon dioxide, leads to larger gas emissivity within the combustion chamber. This feature enables oxy-fuel combustion to achieve higher radiative heat transfer and thus quickly heat up cold input streams. Within this particular environment plus flue gas recirculation, it is claimed that combustion is stabilized without having a flame front.

Flameless combustion reduces the ash content in the flue gases coming out of the
combustor. At high temperatures of oxy-fuel combustion, most of the ash melts and flows down toward the bottom bath. A high wall temperature that is continuously maintained close to the gas temperature allows the molten ash to drain out from the combustor. Above 97% of the ash removal rate is reported [21], and this reference value is used in this study.

Combustion takes place at elevated pressures and at 1400-1600 °C, which is above the ash melting point and the flameless combustion threshold. Note that stoichiometric combustion of coal in pure oxygen reaches up to 3500 °C [13]. While the proposed oxy-fuel combustion is close to stoichiometric, where the equivalence ratio is 0.989, the low combustion temperature is achieved by using the appropriate amount of the recycled flue gases. In this study, the flue gases are extracted and recycled after the HRSG at state 19 shown in Figure 2-2.

In the proposed oxy-fuel process, the combustor temperature is maintained at 1550 °C, by premixing the oxygen stream with the recycled flue gases (see Figure 2-6). After being cooled in the HRSG, 26.1% (by mass) of the flue gases are recycled to achieve this temperature target. The combustor is not adiabatic; its thermal energy losses are dictated by its size. In this study, with the combustor sized for a net power output of 300 MW, the combustor is assumed to lose 2% of the lower heating value of the fuel to the water-cooled walls. The energy lost during combustion is used to heat the feedwater stream, as shown in Figure 2-2.

Coal is supplied in the form of a coal-water slurry stream which contains 0.35 kg water per 1 kg of its total weight. Based on the given coal (see Table 2-1), the mass flow
rate of feedwater is an important contributor to the water and carbon dioxide concentration in the combustion flue gases, as shown in Figure 2-7. Steam is also injected into the pressurized combustor to atomize the slurry particles. We found that the overall cycle efficiency is higher when we extract the atomization steam from the steam turbines than in the case of utilizing the water from external sources after pressurizing it and heating it up to the desired temperature. Therefore, the steam from the high pressure turbines is used for atomization. To achieve a choked flow, the pressure of the atomization steam is set to be about twice that of the combustor pressure. The mass flow rate of this steam injection is set to be one tenth of the coal mass flow. The atomization steam is superheated by 30 °C.

![Figure 2-7 Dependence of the water and carbon dioxide concentration in the flue gases on the amount of feedwater](image)
2.3.3. Steam Generation Unit

To generate steam for the power island, two superheaters, a once-through boiler, and an economizer are used in the HRSG. Passing through the HRSG, steam reaches the supercritical state of 600 °C at 250 bars and is delivered to the power island. The superheat temperature of 600 °C corresponds to presently commercial technologies of steam power plants, and is used for the purpose of evaluating a representative efficiency. The superheaters, also termed as reheaters, yield two reheat subcritical steam flows at 620 °C each, as shown in Figure 2-8. As seen in Figure 2-9, the pinch in the HRSG is set to be 20 °C.

![Figure 2-8 Steam path within the HRSG](image)
The inlet and outlet temperatures of the flue gases passing through the HRSG must remain within an appropriate range in the pressurized oxy-fuel power cycle. Because of the high pressure of the flue gases, the "acid" dew point is higher than the acid dew point at atmospheric pressure. Also the presence of acid gases, SO\(_x\) and NO\(_x\), increases the dew point of the flue gases relative to pure water condensation. Therefore, the outlet temperature of the HRSG is selected to ensure that it is higher than the acid dew point. In the 10 bars base-case, this temperature is set to be 260 °C. To minimize hot corrosion and oxidation, the inlet temperature of the pressurized flue gases into the HRSG is cooled down to 800 °C by secondary recycled flue gases. For this purpose, 61.9% (by mass) of the flue gases leaving the HRSG is recycled and mixed with the HRSG inlet gas. The detailed stream information is shown in Figure 2-10.
Moreover, as shown in Figure 2-2, the acid condenser is introduced at the end of the steam generation unit to cope with the acid condensation and to recover more thermal energy from the pressurized flue gases which has higher latent enthalpy.

2.3.4. Power Island

A supercritical Rankine cycle has been chosen for the power cycle. Based on the ISOTHERM® combustion technology, the Rankine cycle is the most promising option in terms of integration, operating reliability and feasibility. Within the boundary of the supercritical Rankine cycle which has an inlet steam flow at 250 bars and 600 °C, the power island has two reheat steam flows at 620 °C. The exhaust steam is condensed at 0.05 bars based on condenser cooling by Mediterranean Sea water.
In order to optimize the thermal integration with the rest of the cycle, the steam bleeding from the high-pressure and the low-pressure turbines is replaced by the high-pressure flue gas thermal energy recovery system. To make this possible, the presence of the high pressure deaerator is essential. As shown in Figure 2-11, the thermal energy recovery from the pressurized flue gases is not only dependent on the terminal temperature difference at the hot-side of the acid condenser, but also upon the saturation condition inside the deaerator. As shown in Figure 2-2, the feedwater out of the acid condenser, state 3, goes into the deaerator after recovering the thermal energy loss from the combustor. Thus, increasing the exit temperature of the feedwater at the hot-side of the acid condenser, state 3, to recover more thermal energy from the flue gases implies a higher inlet temperature for the feedwater entering the deaerator, state 4. According to the saturation condition of the deaerator, the design point pressure level fixes the exit temperature of the water leaving the deaerator, state 5. As a result, we should increase the deaerator operating pressure level to keep the temperature of the feedwater exiting the deaerator higher than that of the incoming feedwater to the deaerator.
Figure 2-11 Dependence of thermal energy recovery on the deaerator operating conditions

2.3.5. Carbon Dioxide Purification and Compression Unit

The carbon dioxide purification and compression unit employed in this study is based on a purification process proposed by White et al [35]. Instead of exploiting selective catalytic reduction and wet limestone gypsum flue gas desulfurization, the proposed process uses two successive water-wash columns and a low temperature processing unit, as shown in Figure 2-12. Utilizing this process, NO and NO$_2$ are removed as HNO$_3$, and SO$_2$ is removed as H$_2$SO$_4$. The low temperature processing unit is highly simplified because the detailed information of the process is not published. However, it should be sufficient to perform thermodynamic analysis of the carbon dioxide purification and compression unit. Because the compressors account for most of the energy consumption, we can make reasonable estimates of the energy requirement. Note
that this process has been chosen only for the simulation purposes reported in this thesis. Other processes can be considered and adopted in future research.

To remove NOx and SOx from the oxy-coal combustion flue gases, two pressurized water-wash columns are introduced at 15 bars and at 30 bars, respectively. Due to the complete NO2-catalytic conversion of SO2 to sulfuric acid with increasing pressure [36], SOx removal process operates in the water-wash column at 15 bars. The column separates out all the SO2 and SO3 as sulfuric acid, as well as almost half of the remaining water content in the flue gases. After the de-SOx unit, the flue gases are compressed to 30 bars and introduced into the next water-wash column. Here, more than 90% (by mass) of NOx is removed as nitric acid. Most NOx produced from the high temperature combustion is in the form of NO and it must be converted to NO2 in order to remove it as nitric acid. The reaction rate for converting NO to NO2 increases as the pressure is raised and the temperature is lowered [35]. Hence, the removal of NOx is attained in the water-wash column at 30 bars. The corresponding reactions are as follows:
After cleaning up NO\textsubscript{x} and SO\textsubscript{x}, the carbon dioxide concentrated stream is sent to the low temperature processing unit. The remaining impurities, oxygen, argon, and nitrogen, are removed through this unit. The carbon dioxide stream is cooled down to about -54 °C which is close to the carbon dioxide triple point, -56 °C. This cooling process produces two different streams, a capture-ready carbon dioxide stream that has 96.5% (molar basis) of carbon dioxide concentration and an exhaust stream consisting mostly of inert gases. The capture-ready stream is compressed to 110 bars for transportation to a sequestration or an EOR site.

### 2.4. Evaluation of Performance

As briefly explained in the previous section, the base-case design variables of the pressurized oxy-fuel combustion power cycle, shown in Table 2-2, are used to perform a thermodynamic analysis. These design variables represent commercially available technologies or processes in an advanced development stage.
Table 2-2 Base-case design variables

<table>
<thead>
<tr>
<th>Design Variables</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>1. Air Separation Unit</strong></td>
<td></td>
</tr>
<tr>
<td>Oxygen Purity (mol %)</td>
<td>95%</td>
</tr>
<tr>
<td>Oxygen in the Flue Gases (mol %)</td>
<td>3%</td>
</tr>
<tr>
<td>Oxygen Delivery Temperature</td>
<td>200 °C</td>
</tr>
<tr>
<td><strong>2. Pressurized Coal Combustor</strong></td>
<td></td>
</tr>
<tr>
<td>Combustor Pressure</td>
<td>10 bars</td>
</tr>
<tr>
<td>Combustor Temperature</td>
<td>1550 °C</td>
</tr>
<tr>
<td>Combustor Thermal Energy Loss</td>
<td>2%</td>
</tr>
<tr>
<td>Slurry Water (wt %)</td>
<td>35%</td>
</tr>
<tr>
<td>Steam Injection (wt %)</td>
<td>10%</td>
</tr>
<tr>
<td><strong>3. Steam Generation</strong></td>
<td></td>
</tr>
<tr>
<td>Inlet temperature of HRSG</td>
<td>800 °C</td>
</tr>
<tr>
<td>Outlet Temperature of HRSG</td>
<td>260 °C</td>
</tr>
<tr>
<td><strong>4. Power Island</strong></td>
<td></td>
</tr>
<tr>
<td>Turbine Inlet Pressure</td>
<td>250 bars</td>
</tr>
<tr>
<td>Turbine Inlet Temperature</td>
<td>600 °C</td>
</tr>
<tr>
<td>Reheat Temperature</td>
<td>620 °C</td>
</tr>
<tr>
<td>Deaerator Pressure</td>
<td>10 bars</td>
</tr>
<tr>
<td>Condenser Pressure</td>
<td>0.05 bars</td>
</tr>
<tr>
<td><strong>5. Carbon Dioxide Purification and Compression</strong></td>
<td></td>
</tr>
<tr>
<td>CO₂ Compression Pressure</td>
<td>110 bars</td>
</tr>
</tbody>
</table>

Based on these design variables, the oxy-fuel combustion power cycle utilizing a pressurized coal combustor is evaluated. The pressurized combustor system is compared to the atmospheric oxy-fuel power cycle which is based on the same design variables shown in Table 2-2, other than the combustor pressure and the oxygen delivery temperature. The atmospheric system is based on the 1.1 bars combustion pressure. In the atmospheric combustion system, the oxygen stream, state 13 in Figure 2-2, is not compressed and thermal energy sources are not sufficient to heat up this stream to 200 °C. As a result, the
same oxygen delivery temperature target as the pressurized oxy-fuel power cycle cannot be achieved. However, because the acid dew point of the atmospheric combustion system is considerably lower than that in the pressurized combustion case, a lower oxygen delivery temperature can be used to avoid the acid condensation when it is mixed with the recycled flue gas stream, state 19 in Figure 2-2. Consequently, the atmospheric oxy-fuel power cycle is based on a 100 °C oxygen delivery temperature. Moreover, in the atmospheric pressure combustor system, steam from the low-pressure turbines is used to heat up the feedwater leaving the second feedwater pump. The Table 2-3 shows the conditions for each of the major states for both cases.

The oxy-fuel combustion system utilizing a pressurized coal combustor is beneficial in terms of larger flue gas thermal energy recovery and smaller parasitic power requirements. At high pressure, water condensation begins at higher temperature than the conventional oxy-fuel system based on lower operating pressures. In other words, the quality of recoverable latent enthalpy of water increases as we raise the operating pressure of the oxy-fuel system. Given high water content in the flue gases, this feature leads to better thermal integration within the system and larger thermal energy recovery. In addition, the fact that carbon dioxide needs to be pressurized above supercritical state for the sequestration purpose gives the oxy-fuel system a possibility to increase the operating pressure and reduces the overall parasitic power requirements. The detailed results are discussed in the following sections.
Table 2-3 Stream results for (a) the atmospheric oxy-fuel power cycle and (b) the pressurized oxy-fuel power cycle

<table>
<thead>
<tr>
<th></th>
<th>Pressure (bar)</th>
<th>Temperature (°C)</th>
<th>Mass Flow Rate (kg/s)</th>
</tr>
</thead>
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<tr>
<td>#</td>
<td>(a)</td>
<td>(b)</td>
<td>(a)</td>
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<td>10</td>
<td>1549.3</td>
</tr>
<tr>
<td>16</td>
<td>1.1</td>
<td>10</td>
<td>800</td>
</tr>
<tr>
<td>17</td>
<td>0.983</td>
<td>9.351</td>
<td>260.6</td>
</tr>
<tr>
<td>18</td>
<td>1.1</td>
<td>10</td>
<td>275.8</td>
</tr>
<tr>
<td>19</td>
<td>1.1</td>
<td>10</td>
<td>275.8</td>
</tr>
<tr>
<td>20</td>
<td>0.983</td>
<td>9.351</td>
<td>260.6</td>
</tr>
<tr>
<td>21</td>
<td>0.983</td>
<td>9.351</td>
<td>73.2</td>
</tr>
<tr>
<td>22</td>
<td>110</td>
<td>110</td>
<td>30</td>
</tr>
<tr>
<td>23</td>
<td>1.2</td>
<td>1.2</td>
<td>30</td>
</tr>
</tbody>
</table>

2.4.1. Flue Gas Thermal Energy Recovery

Operating at high pressure in the combustor makes recovering more thermal energy from the flue gases possible. Whereas an air-fired combustion system produces a small amount of water in the flue gases, about 8.7% (by volume) [14], nearly half of oxy-fuel combustion power cycle flue gases is composed of water, nearly 48% (by volume). In addition, the saturation temperature of the water increases with increasing the operating
pressure. While the flue gases begin to condense at about 80 °C in the atmospheric oxy-fuel system, condensation begins to occur at about 150 °C in the pressurized system. These two facts enable the pressurized oxy-fuel system to recover more thermal energy from the flue gas stream.

The incremental improvement in the thermal energy recovery is achieved in the acid condenser which recuperates the latent enthalpy of water. This can be explained through the outlet temperature of the acid condenser represented as stream 21 in Figure 2-2. With the same inlet thermal energy sources (the HRSG outlet stream at 260 °C), the acid condenser of the atmospheric case yields an outlet temperature of 73.19 °C, which is higher than that of the pressurized case, 60.49 °C, as shown in Figure 2-13. Therefore, through the acid condenser, we can recover about 80% more thermal energy from the flue gases.

![Figure 2-13 Thermal conditions of the acid condenser](image)

Figure 2-13 Thermal conditions of the acid condenser
In conventional power cycles operating at atmospheric pressure, the water in the flue gases begins to condense at around 50 °C, and almost all of this condensation enthalpy is lost. On the other hand, our pressurized oxy-fuel system yields flue gases with steam that begins to condense at about 150 °C. At this temperature, we can use the latent enthalpy of condensation to heat up the condensate leaving the first feedwater pump, from state 2 to state 3, as shown in Figure 2-2. With the help of the acid condenser and its thermal integration with the system, a large amount of latent enthalpy recovery is feasible, and the overall efficiency increases.

2.4.2. Power Output and Overall Efficiency

The increased flue gas thermal energy recovery enables us to save a large amount of steam in the feedwater heating system. The thermal energy recovery from the flue gases is sufficient to replace the thermal energy duty of the feedwater heating system from the high-pressure and the low-pressure steam turbines. Thus, no steam bleeding from those turbines is required, as shown in Figure 2-14. A significant reduction in the steam bleeding increases the power generation from the turbines. As shown in Figure 2-2 and Figure 2-14, the overall steam bleeding drops from 32 kg/s in the atmospheric case to 11.9 kg/s in the pressurized system. Consequently, the pressurized oxy-fuel system produces more gross power output and net power output.
Table 2-4 shows the overall performance of both systems. Because of the reduction in the steam bleeding, the gross efficiency reaches 46.2% (HHV) or 48.2% (LHV), which is higher than the atmospheric oxy-fuel power cycle by 2 percentage point. Extracting more thermal energy from the flue gases through suitable thermal integration of the acid condenser and the high pressure deaerator increases the gross efficiency. Even when compared to the previous study on the pressurized oxy-fuel power cycle [27], the proposed approach has a 7 percentage point higher gross efficiency.
Table 2-4 Overall performance of (a) the atmospheric oxy-fuel power cycle and (b) the pressurized oxy-fuel power cycle

<table>
<thead>
<tr>
<th>Performance Parameters</th>
<th>Unit</th>
<th>(a)</th>
<th>(b)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thermal energy Input (HHV) (LHV)</td>
<td>MW&lt;sub&gt;th&lt;/sub&gt;</td>
<td>874.6</td>
<td>874.6</td>
</tr>
<tr>
<td></td>
<td></td>
<td>839.1</td>
<td>839.1</td>
</tr>
<tr>
<td>Gross Power Output</td>
<td>MW&lt;sub&gt;e&lt;/sub&gt;</td>
<td>388.0</td>
<td>404.5</td>
</tr>
<tr>
<td>Net Power Output</td>
<td>MW&lt;sub&gt;e&lt;/sub&gt;</td>
<td>264.3</td>
<td>292.6</td>
</tr>
<tr>
<td>Gross Efficiency (HHV) (LHV)</td>
<td>%</td>
<td>44.4</td>
<td>46.2</td>
</tr>
<tr>
<td></td>
<td></td>
<td>46.2</td>
<td>48.2</td>
</tr>
<tr>
<td>Net Efficiency (HHV) (LHV)</td>
<td>%</td>
<td>30.2</td>
<td>33.5</td>
</tr>
<tr>
<td></td>
<td></td>
<td>31.5</td>
<td>34.9</td>
</tr>
<tr>
<td>Fuel Demand</td>
<td>kg/s</td>
<td>30</td>
<td>30</td>
</tr>
<tr>
<td>Steam Demand</td>
<td>kg/s</td>
<td>211.9</td>
<td>210</td>
</tr>
<tr>
<td>Oxygen Demand</td>
<td>kg/s</td>
<td>73.52</td>
<td>73.52</td>
</tr>
<tr>
<td>Flue Gas Flow Rate (Into the HRSG)</td>
<td>kg/s</td>
<td>1011.1</td>
<td>1004.7</td>
</tr>
<tr>
<td>Recirculation Ratio:</td>
<td>%</td>
<td>88.1</td>
<td>88</td>
</tr>
<tr>
<td>Combustor</td>
<td></td>
<td>(25.8)</td>
<td>(26.1)</td>
</tr>
<tr>
<td>HRSG</td>
<td></td>
<td>(62.3)</td>
<td>(61.9)</td>
</tr>
<tr>
<td>Flue Gas Flow Rate (Into the Purification and</td>
<td>kg/s</td>
<td>107.7</td>
<td>87.7</td>
</tr>
<tr>
<td>Compression Unit)</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Details of the power generated and consumed in the atmospheric and the pressurized oxy-fuel power cycles are shown in Figure 2-15. Most of the parasitic power demand comes from the air separation unit which consumes nearly 20% of gross power output. Because the proposed system requires a pressurized oxygen stream to the combustor, the power consumption of the air separation unit is higher than that of the atmospheric oxy-fuel system. The pressurized system needs more energy by 4 percentage point of the gross power output in the air separation unit than the atmospheric oxy-fuel power cycle. If we consider the power demand without the oxygen compression work, it becomes consistent with the atmospheric case or previous oxy-fuel research [34].
Figure 2-15 Net power and parasitic power demand: (a) the atmospheric oxy-fuel power cycle and (b) the pressurized oxy-fuel power cycle.

Figure 2-15 also shows that the carbon dioxide purification and compression unit power demand is smaller in the pressurized oxy-fuel power cycle. In the proposed approach, this unit requires 3.7% of gross power output, while the atmospheric system needs 9.5%. This is mainly because the flue gases are already pressurized. Within the pressurized oxy-fuel system, the oxygen stream is compressed to 10 bars by the air separation unit. The combustion products are at the same pressure. Therefore, the flue gases flowing into the purification and compression train, state 21 in Figure 2-2, are already at high pressure, reducing the compression work required for liquefaction. This feature is advantageous for the overall performance. Table 2-4 shows that the oxygen demand is smaller than the flue gas flow rate into the purification and compression train. Thus, by compressing a smaller amount of gas before the purification and compression unit, the proposed approach can reduce the overall compression work. In addition, the flow rate of the flue gases undergoing compression in the pressurized system (87.7 kg/s) is smaller than in the atmospheric system (107.7 kg/s), as shown in Table 2-4. The increased thermal
energy recovery leads to larger amounts of water condensation, and hence, the pressurized oxy-fuel cycle flue gases that enter the purification and compression unit have lower mass flow rate after passing through the acid condenser. As a result, the flue gas compression work duty of the pressurized system is smaller than that of the atmospheric case.

Another large reduction in parasitic power demand is derived from the fan of the recycled flue gases, described as fan compression work in Figure 2-15. The energy requirement for the fan drops by 7.6 MW in the pressurized oxy-fuel power cycle. This fact can be explained by the extent of the pressure drop across the steam generation units and the flue gas recirculation pipe, and the corresponding pressure ratio across the fan. The compression power demand grows significantly in the low pressure range in which the compression ratio is higher. Based on the estimated pressure drop, the fan in the atmospheric case compresses the recycled flue gas stream from 0.98 bars to 1.1 bars, whereas that of the pressurized system compresses it from 9.35 bars to 10 bars. With almost the same mass flow rate of the recycled flue gases, the pressurized system has a smaller pressure ratio across the fan, and thus, the pressurized oxy-fuel power cycle requires less fan compression work than the atmospheric cycle.

As a result, the proposed pressurized oxy-fuel combustion system achieves better net efficiency, 33.5% (HHV) or 34.9% (LHV) than that achieved by the atmospheric combustion cycle. The high-pressure flue gas thermal energy recovery, the increased gross power output, and less overall compression work leads to higher net efficiency.

Figure 2-16 shows the contributions of the different units to the improvement of the net efficiency. While larger thermal energy recovery, reduced carbon dioxide
compression work and smaller flue gas recirculation fan compression work lead to the net efficiency gain of 5.5%, a large reduction in the net efficiency (2.13%) occurs in the air separation unit. Therefore, if we improve the performance of the air separation unit or adopt advanced separation technologies such as the Ion Transport Membrane method, the pressurized oxy-fuel combustion system could be a better option for fossil-fired power plants with carbon dioxide capture and sequestration.

![Figure 2-16 Net efficiency gain and loss through each unit from the conventional atmospheric oxy-fuel cycle to the pressurized oxy-fuel system](image)

2.5. Conclusions

The proposed oxy-fuel combustion power cycle that utilizes a pressurized coal combustor shows better performance than the atmospheric pressure system in terms of the thermal energy recovery and the gross power output. Based on the 10 bars operating pressure and a supercritical Rankine cycle, this approach enables the system to recover more thermal energy from the flue gases and avoids the need for the high-pressure and the
low-pressure steam bleeding. Because of the raised dew point and the higher available latent enthalpy in the flue gases, it is possible to recover a large amount of high-pressure water-condensing flue gas thermal energy. Recuperating more thermal energy from the flue gases to generate steam, the system is able to eliminate the high-pressure and the low-pressure steam bleeding and to use more steam in the turbines. As a result, the pressurized oxy-fuel power cycle raises the gross efficiency to 46.2% (HHV) or 48.2% (LHV). The pressurized system yields more gross power output than the atmospheric combustion pressure case.

The parasitic power demand of the pressurized oxy-fuel power cycle is lower than the atmospheric system. The air separation unit and the carbon dioxide purification and compression unit employed in this study improve the overall performance. Effectively-balanced compression work duty between these two units lets the system lower parasitic power demand. The acid condenser and the high pressure deaerator make this approach possible without adding complexity to the system. In addition, compression of the recycled flue gases through a fan at the high pressure range decreases the compression work demand. Consequently, the proposed approach has lower parasitic power demand by 11.8 MWₑ.

As a result, the proposed approach achieves 33.5% (HHV) or 34.9% (LHV) net efficiency which is higher than those of conventional air-fired power cycles with carbon dioxide capture and atmospheric oxy-fuel power cycles.
Chapter 3 Pressure Dependence

3.1. Overview

Based on the finding from the base-case analysis, we have concluded that the combustor operating pressure has a strong impact on the overall performance of the oxy-fuel system, as shown in Figure 3-1.

![Figure 3-1 Pressure dependence of the performance of the pressurized oxy-fuel combustion power cycle](image)

In the pressurized oxy-fuel system, oxygen is pre-compressed in the air separation unit where the mass flow rate is smaller than that of the flue gases into the carbon dioxide purification and compression unit. By pre-compressing the gas stream in the air separation unit, the combustion flue gases are at the high pressure and the compression work duty of the carbon dioxide purification and compression unit, which compresses the...
flue gases from the combustor operating pressure to 110 bars, decreases. Increasing the compression work duty in the air separation unit while decreasing the work duty in the carbon dioxide purification unit, we lower the parasitic power demand.

In addition, the elevated dew point and higher available latent enthalpy in the flue gases lead to higher thermal energy recovery from the flue gases. The available latent enthalpy is defined as the recoverable amount of the latent enthalpy of the water in the flue gases (hot stream) based on the inlet and outlet temperatures of the feedwater (cold stream) across the heat exchanger. The increased water-condensing thermal energy recovery enables the system to produce more gross power by eliminating the steam bleeding from the low pressure steam turbines. Consequently, compared to a conventional atmospheric oxy-fuel combustion system, the pressurized system was shown to be more efficient, taking advantage of lower overall compression work demand and higher gross power output.

Moreover, our base-case analysis showed that the fan compression work has a significant impact on the overall performance. Most oxy-fuel combustion systems have implemented recirculation of the flue gases to decrease the combustion temperature to a reasonable level. Thus, a significant amount of the flue gases is re-circulated within the system, requiring a substantial amount of energy to compensate for the pressure drop across the steam generation unit and the recirculation pipe.

In this chapter, we perform a pressure sensitivity analysis to find the optimal combustor operating pressure. We calculate the pressure dependence of the thermal energy recovery rate, the overall parasitic power demand, the gross power output and the overall efficiency. The analysis enables us to determine which parameters are important
in operating the pressurized system and to examine how to improve the overall performance.

3.2. Methodology

A 300 MWₑ coal-fired power plant, which we use in the base-case cycle analysis, is modified and used as the base-case for the pressure sensitivity analysis. In the previous chapter, we discussed the characteristics of the base-case; all important design variables described in Chapter 2 are used again in this analysis. Through the base-case cycle analysis, we find that the high-pressure and the low-pressure turbine feedwater heaters can be eliminated. Increased thermal energy recovery from the flue gases lowers the heat duty of the regeneration system, and thus, feedwater heaters that exploit the steam bleeding from the high-pressure and the low-pressure turbines are removed.

The combustor operating pressure was controlled by changing the oxygen delivery pressure to the combustor from 1.1 bars to 30 bars. The proposed approach is explained and discussed in the following sections. Two commercial simulation packages, Thermoflex® and Aspen Plus®, were used to conduct the study. Aspen Properties® provides thermodynamic and transport properties used to estimate the pressure drop through the flue gas recirculation path, as explained in Section 3.2.2.

3.2.1. Base-Case and Design Variables

The pressure sensitivity analysis was performed using the base-case cycle we
developed in the base-case analysis. As explained in Chapter 2, the base-case plant consists of five primary units, the air separation unit, the pressurized coal combustor, the steam generation unit, the power island and the carbon dioxide purification and compression unit. Each unit represents commercially available technologies or processes at an advanced development stage. The combustor, the HRSG and the double recycling scheme are patented technologies developed by ITEA. Detailed information on the overall processes and coal analysis data can be found in Chapter 2.

Based on the base-case processes, we fixed a set of important design variables to focus on system’s operating pressure dependence. Table 3-1 shows the design variables used in the pressure sensitivity analysis. As explained in Section 2.3, the gas side variables describe the characteristics of the available air separation unit, the combustor and the HRSG, whereas parameters in the steam side represent the supercritical Rankine cycle.

Table 3-1 Fixed design variables used in the pressure sensitivity analysis

<table>
<thead>
<tr>
<th>Design Variables</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Gas Side</strong></td>
<td></td>
</tr>
<tr>
<td>Oxygen Purity (mol %)</td>
<td>95%</td>
</tr>
<tr>
<td>Oxygen in the Flue Gas (mol %)</td>
<td>3%</td>
</tr>
<tr>
<td>Combustion Temperature</td>
<td>1550 °C</td>
</tr>
<tr>
<td>Inlet and Outlet Temperatures of HRSG</td>
<td>800 °C/260 °C</td>
</tr>
<tr>
<td>CO₂ Compression Pressure</td>
<td>110 bars</td>
</tr>
<tr>
<td><strong>Steam Side</strong></td>
<td></td>
</tr>
<tr>
<td>HPT Inlet Pressure/Temperature</td>
<td>250 bars/600 °C</td>
</tr>
<tr>
<td>Reheat Pressure/Temperature:</td>
<td>57 bars/620 °C, 10 bars/620 °C</td>
</tr>
<tr>
<td>Deaerator Pressure</td>
<td>10 bars</td>
</tr>
<tr>
<td>Condenser Pressure</td>
<td>0.05 bars</td>
</tr>
</tbody>
</table>

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3.2.2. Pressure Drop Estimation Method

As explained in the previous section, the fan compression works for the flue gas recirculation system account for a significant amount of the parasitic power demand in the pressurized oxy-fuel system. In this system, recirculation of 70% to 80% of the flue gases must be used in order to reduce the combustion temperature [12]. Because a considerable pressure drop occurs across the steam generation unit and the flue gas recirculation pipe, a fan is required to re-circulate the flue gases.

As expected, the pressure drop and the corresponding fan compression work depend on the combustor operating pressure. To estimate the operating pressure dependence of the fan compression work, we have to first estimate the pressure drop. From the pressure drop, $\Delta P$, correlations [37, 38] we know that:

\[
\Delta P_{HRSG} = N \frac{\rho_{V_{\text{max}}}^2}{2} f_{HRSG}, \quad \text{where } f_{HRSG} = F \left( Re_D, \frac{S_T}{D} \right), \quad \text{Eq. 3-1}
\]

\[
\Delta P_{\text{pipe}} = \rho f_{\text{pipe}} \frac{L V^2}{d}, \quad \text{where } f_{\text{pipe}} = F \left( Re_d, \frac{S_T}{d} \right), \quad \text{Eq. 3-2}
\]

Aspen Properties® is used to find density ($\rho$), dynamic viscosity ($\mu$), specific heat capacity ($c_p$) and thermal conductivity ($\kappa$) of the flue gases and water. Using these correlations, we are able to evaluate the pressure drop across the HRSG and the recirculation pipe, respectively. The friction factors, $f$, are found from the following correlations [39, 40],

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\[ f_{\text{HRSG}} = \chi \left( 0.162 + \frac{0.181 \times 10^4}{\text{Re}} + \frac{0.792 \times 10^8}{\text{Re}^2} - \frac{0.165 \times 10^{13}}{\text{Re}^3} + \frac{0.872 \times 10^{16}}{\text{Re}^4} \right). \quad \text{Eq. 3-3} \]

\[ f_{\text{pipe}} = \left\{ -2.0 \log \left[ \frac{(2 \varepsilon/d)}{7.4} - \frac{5.02}{\text{Re}_d} \log \left( \frac{(2 \varepsilon/d)}{7.4} + \frac{13}{\text{Re}_d} \right) \right] \right\}^{-2} \quad \text{Eq. 3-4} \]

The heat exchanger and the recirculation pipe design assumptions include,

- The correction factor: \( \chi = 1 \)
- The HRSG tube diameter: \( D = 50 \text{ mm} \)
- The HRSG tube length: \( L_{\text{tube}} = 18.29 \text{ m} \)
- The HRSG tube transverse pitch: \( S_T = 100 \text{ mm} \)
- The HRSG tube longitudinal pitch: \( S_L = 100 \text{ mm} \)
- The recirculation pipe length: \( L \) (explaining the pressure drop across the valves and the curves within the recirculation path)
- The recirculation pipe diameter: \( d \) (based on the volumetric flow rate and the velocity \( V \))
- The roughness: \( \varepsilon = 0.046 \text{ mm} \) (commercial steel)
- The maximum velocity: \( V_{\text{max}} \approx 15.5 \text{ m/s} \)
- The density of flue gases: \( \rho_0 \) and \( \rho \) (based on the given pressure and the temperature)
- The inlet and outlet temperatures of each heat exchanger:

\( T_{\text{H,in}} = \) the inlet temperature of the flue gas stream

\( T_{\text{H,out}} = \) the outlet temperature of the flue gas stream
\[ T_{C,\text{in}} = \text{the inlet temperature of the feedwater stream} \]

To calculate \( \Delta P_{HRSG} \), the following three equations [41-43] are used to evaluate the Nusselt number, \( \overline{Nu_D} \); the number of transfer units, \( N_{tu} \); and the number of tube rows, \( N \).

\[ \overline{Nu_D} = \frac{hD}{\kappa} = \left(1 + \frac{2D}{3S_L}\right) \left\{ 0.3 + \frac{0.62Re_0^{1/3}Pr^{1/3}}{1 + \left(\frac{0.4}{Pr}\right)^{2/3}} \left[ 1 + \left(\frac{Re_0}{282000}\right)^{1/2}\right] \right\}, \quad \text{Eq. 3-5} \]

where, \( h \) = heat transfer coefficient, \( \kappa \) = thermal conductivity

\[ N_{tu} = \frac{2C_{\text{min}}(T_{H,\text{in}}-T_{C,\text{in}})(1+R)}{m_{H}C_{P,H}(T_{H,\text{in}}-T_{H,\text{out}})(1+R^2)^{1/2}}, \quad \text{Eq. 3-6} \]

where, \( R = C_{\text{min}}/C_{\text{max}} \), \( C_{\text{min}} = \min\{m_{H}C_{P,H}, m_{C}C_{P,C}\} \), \( C_{\text{max}} = \max\{m_{H}C_{P,H}, m_{C}C_{P,C}\} \).

\( m \) = mass flow rate of cold and hot streams, \( C_{P} \) = specific heat capacity of cold and hot streams,

\[ N = \frac{N_{tu}C_{P,H}S_T}{\rho \lambda D_{\text{tube}} \max\left\{ \frac{S_T}{S_T-B'}, \frac{S_T}{2L} \right\}}, \quad \text{Eq. 3-7} \]

The number of tube rows \( N \) is calculated by combining Eq. 3-5, Eq. 3-6 and Eq. 3-7. Inserting the numbers from Eq. 3-3, Eq. 3-4 and Eq. 3-7 into Eq. 3-1 and Eq. 3-2, and adding these two, we estimate the overall pressure drop across the HRSG and the recirculation pipe. The result is shown in Section 3.3.2.
3.3. Pressure Dependence of System Performance

The performance of the pressurized oxy-fuel power system depends on the flue gas thermal energy recovery and the parasitic power requirements, which includes the compression work of the air separation unit, the carbon dioxide purification and compression unit and the flue gas recirculation fan. Thus, the pressure dependence of these quantities must be carefully investigated.

3.3.1. Thermal Energy Recovery and Gross Power Output

As the combustor operating pressure rises, so does the thermal energy recovery from the flue gases that is used to generate steam. At higher pressures, the saturation temperature of the water in the flue gases rises as well. In this case, we can implement more effective thermal integration between a hot stream (flue gases) and a cold stream (feedwater).

As shown in Figure 3-2, the inlet temperature of the cold stream to the acid condenser, 33 °C, is determined by the condenser, and the maximum exit temperature of the feedwater out of the acid condenser is fixed by the deaerator operating conditions. Because the deaerator controls the feedwater inlet temperature so that it does not allow the feedwater to become steam before the second feedwater pump, we fix the maximum temperature we can reach at the deaerator operating pressure (10 bars in our study). Under these cold stream temperature constraints, increasing the dew point enables us to recover the latent enthalpy at higher temperatures. Consequently, we can recover more
latent enthalpy from the flue gases at higher temperatures by increasing the operating pressure.

Figure 3-3 shows the dependence of the saturation temperature of the water (the blue line) and the available latent enthalpy (the red line) on the flue gas pressure. As mentioned above, the available latent enthalpy is defined as the recoverable amount of the latent enthalpy of the water in the flue gases (hot stream) based on the inlet and outlet temperatures of the feedwater (cold stream) across the acid condenser.
Because we have a large amount of water in the oxy-fuel combustion flue gases, 48% (molar) in our base-case, the increase in the saturation temperature of the water enables us to recover a significant amount of the available latent enthalpy from the flue gases. As shown in Figure 3-4, the total thermal energy (the blue line) recovered from the flue gases increases by nearly 7%, as the combustor operating pressure is raised from 1.1 bars to 30 bars.

The considerable increase in the thermal energy recovery lowers the regeneration heat duty from the turbines. While the conventional atmospheric oxy-fuel system cannot recover the HHV of fuel through the steam generation unit, the pressurized oxy-fuel power cycle is able to recuperate most of the HHV of the fuel because of the increased dew point as discussed above. As a result, the former needs a higher portion of the steam from steam turbines. As we recover more thermal energy from the flue gases, the regeneration heat duty and the corresponding steam bleeding from the low pressure and the intermediate
pressure steam turbines decrease. Figure 3-4 shows the variation in the overall steam bleeding (the red line) with increase in the combustor operating pressure.

![Graph showing variations in thermal energy recovery and overall steam bleeding.]

Figure 3-4 Variations in the thermal energy recovery (blue line) and the overall steam bleeding to feedwater heaters (red line) according to the change in the combustor operating pressure.

As shown in Figure 3-5, the gross power output increases with increasing the operating pressure, following the thermal energy recovery curve in Figure 3-4. Larger thermal energy recovery leads to the reduction in the steam bleeding from the steam turbines, and thus, the system is able to produce more gross power.
3.3.2. Deaerator Feedwater Inlet Temperature, Acid Condenser Pinch Point and Feedwater Mass Flow Rate

The fact that the deaerator feedwater inlet temperature should be less than the saturation temperature determines the maximum exit temperature of the feedwater out of the acid condenser. As shown in Figure 2-2, the condensate leaving the condenser flows into the acid condenser to recover the latent enthalpy from the flue gases. Next, the heated water stream enters the deaerator after recuperating more thermal energy by cooling the combustor walls. Therefore, the deaerator feedwater inlet temperature is strongly related to the amount of thermal energy recovery through the acid condenser. However, as explained in the previous section, the deaerator limits the feedwater inlet temperature in order to avoid injecting steam into the feedwater pump. In other words, the deaerator operating condition (10 bars) fixes the maximum exit temperature of the feedwater out of
the acid condenser. Figure 3-6 shows that the deaerator feedwater inlet temperature reaches the maximum value as the process operating pressure increases. Consequently, the deaerator operating conditions have an impact on the amount of thermal energy recovery from the flue gases through the acid condenser.

![Figure 3-6 Dependence of the deaerator feedwater inlet temperature on the combustor operating pressure](image)

**Figure 3-6 Dependence of the deaerator feedwater inlet temperature on the combustor operating pressure**

As we increase the combustor operating pressure, the pinch point temperature within the acid condenser varies, as shown in Figure 3-7. While the pinch point occurs at the dew point of the water at low pressures, it moves down to the exit temperature of the flue gas stream above 11 bar (see Figure 3-8). Note that the two-phase region is not a flat line as seen in pure water condensation because the mole fraction of the water in the flue gases decreases as the water is condensed. As the combustor operating pressure increases, water condensation begins at higher temperature, and more water is condensed through the latent
enthalpy recovery across the acid condenser. Based on the larger latent enthalpy recovery at higher pressure, the fact that the partial pressure of the water in the flue gases drops at a given operating pressure following water condensation leads to an increase in the slope of the two-phase region. This is why the pinch point temperature can move down to the exit temperature of the flue gas stream. Since the condensate temperature (inlet temperature of the cold stream to the acid condenser) is fixed by the water condenser, the flue gas temperature cannot decrease any more if the pinch point occurs at the hot stream exit, which is at the condensate inlet. Therefore, the acid condenser pinch point temperature also plays an important role in the thermal energy recovery from the flue gases.

Figure 3-7 Variation in the acid condenser pinch point temperature with increasing the combustor operating pressure
Figure 3-8 Movement of the pinch point temperature within the acid condenser from 1.1 bars to 30 bars (shown by TQ diagrams, red line=flue gases, blue line=feedwater)

The deaerator feedwater inlet temperature and the acid condenser pinch point affect the amount of thermal energy recovery from the flue gases as discussed above. Therefore, these two parameters are limiting factors for the cycle efficiency. According to the change in the deaerator feedwater inlet temperature and the acid condenser pinch point, the increase in the mass flow rate of the feedwater, which is closely related to the overall steam mass flow rate, is also limited, as shown in Figure 3-9.

Figure 3-9 Dependence of the deaerator feedwater inlet mass flow rate on the combustor operating pressure
3.3.3. Pressure Drop and Fan Compression Work

The pressure drop and the fan compression work play important roles in the overall performance of the pressurized oxy-fuel power system. Because the flue gas recirculation is a unique characteristic of the oxy-fuel system, it is essential to examine the related power consumption in the thermodynamic cycle analysis. As explained in Section 3.2.2, the pressure drop is strongly dependent on the operating pressure.

To achieve higher heat transfer coefficients, we kept the maximum velocity ($V_{\text{max}}$) constant within the HRSG and had denser flue gases at higher operating pressures. This was possible by reducing the cross sectional area of the HRSG. In addition, the diameter of the flue gas recirculation pipe was determined so as not to have a huge pipe by increasing the flue gas velocity. Because the volumetric flow rate grows as the combustor operating pressure is decreased from 10 bars to 1.1 bars, as shown in Figure 3-10, we had to linearly increase the flue gas velocity. On the other hand, above 10 bars, we kept the flue gas velocity constant because we have reasonable pipe diameters as the volumetric flow rate decreases.

Table 3-2 shows the reference velocities used in the recirculation pipe pressure drop estimation. Moreover, the flue gas recirculation pipe has the valves and the curves within the recirculation path. This fact accounts for a large amount of the pressure drop across the recirculation pipe. Figure 3-11-(a) shows the pressure drop across the HRSG (the blue line) and the pressure drop across the flue gas recirculation pipe (the red line). As a result, a higher pressure drop is encountered as the operating pressure increases, as shown in Figure 3-11-(b).
Figure 3-10 Volumetric flow rate into the recirculation pipe versus the combustor operating pressure

Table 3-2 Reference flue gas velocities within the flue gas recirculation pipe

<table>
<thead>
<tr>
<th></th>
<th>1.1 bars</th>
<th>10 bars</th>
</tr>
</thead>
<tbody>
<tr>
<td>Recirculation to the combustor</td>
<td>25 m/s</td>
<td>4 m/s</td>
</tr>
<tr>
<td>Recirculation to the HRSG</td>
<td>30 m/s</td>
<td>14 m/s</td>
</tr>
</tbody>
</table>

Figure 3-11 Impact of the change in the combustor operating pressure on (a) the HRSG pressure drop (blue line) and the flue gas recirculation pipe pressure drop (red line), (b) the overall pressure drop
Based on the pressure drop, the pressure ratios across the HRSG (the blue line) and the recirculation pipe (the red line) vary, as shown in Figure 3-12-(a). Pressure ratios are defined as the ratio of the inlet pressure to the outlet pressure. While the HRSG pressure ratio rises continuously like the HRSG pressure drop, the pressure ratio across the flue gas recirculation pipe decreases below 10 bar and increases above 10 bar. Because the recirculation pipe pressure drop is governed by the pipe diameter (Eq. 3-2), which is controlled by the flue gas velocity, the recirculation pipe pressure drop varies (see Figure 3-11-(a)). This variation leads to the pressure ratio across the flue gas recirculation pipe, as shown in Figure 3-12-(a). Combining the pressure ratios across the HRSG and the recirculation pipe, we have the overall pressure ratio across the fan, as shown in Figure 3-12-(b).

![Figure 3-12](image)

Figure 3-12 Impact of the variation in the combustor operating pressure on (a) the pressure ratio across the HRSG (blue line) and the pressure ratio across the flue gas recirculation pipe (red line), (b) the pressure ratio across the fan
The fan compression work is a strong function of the pressure ratio, and therefore, it follows the variation in the pressure ratio over the fan and has the minimum value at 10 bars, as shown in Figure 3-13.

![Figure 3-13](image)

**Figure 3-13** Dependence of the fan compression work demand on the combustor operating pressure

### 3.3.4. Air Separation Unit and Carbon dioxide Purification and Compression Unit Work Requirements

The pressurized oxy-fuel power cycle needs more compression work in the air separation unit than that of the atmospheric oxy-fuel system. Because the operating pressure is governed by the oxygen delivery pressure to the combustor, the air separation unit requires more compression work as the operating pressure increases. As shown in Figure 3-14, the air separation unit power consumption (the blue line) grows significantly by nearly 44%, as the combustor pressure is increased from 1.1 bars to 30 bars. The air
separation unit is the largest power consumer of the equipment implemented in the pressurized oxy-fuel system. At higher operating pressures, it is important to control the air separation unit power consumption to improve the overall performance of the system.

On the other hand, because the flue gases are already at high pressures, the compression works in the carbon dioxide purification unit decrease as the operating pressure increases. The carbon dioxide purification and compression unit compresses the CO₂-concentrated gases from the operating pressure up to 110 bars. Therefore, the increase in the operating pressure lowers the pressure ratio across this unit, requiring less auxiliary compression work (the red line), as shown in Figure 3-14.

![Figure 3-14 Compression work demands of the air separation unit (blue line) and the carbon dioxide purification and compression unit (red line) with increasing the combustor operating pressure](image)

Figure 3-14 Compression work demands of the air separation unit (blue line) and the carbon dioxide purification and compression unit (red line) with increasing the combustor operating pressure.
3.3.5. Overall Compression Work

Based on the results in Section 3.3.3 and 3.3.4, we now define the pressure dependence of the overall compression work demand in the pressurized oxy-fuel system. Figure 3-15 shows the variation in the overall compression work as the operating pressure increases. The total parasitic power demand follows the trend of the fan compression work shown in Figure 3-13. When we combine the air separation unit compression work and that of the carbon dioxide purification and compression unit, their sum does not vary significantly with the operating pressure. As a result, the overall compression work, which includes the power consumptions of the air separation unit, the carbon dioxide purification unit and the recirculation fan, is mostly dependent on the fan compression work. We conclude that the overall parasitic power demand of the pressurized oxy-fuel system is a strong function of the flue gas recirculation power requirement. Again, the minimum total parasitic power demand occurs at 10 bars as is the case with the fan compression work.

![Overall Compression Work Graph](image)

Figure 3-15 Overall parasitic compression work demand versus the combustor operating pressure

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The variation in the overall parasitic work demand points out that, in order to improve the performance of the pressurized oxy-fuel system, we should focus on the pressure drop within the flue gas recirculation path.

3.3.6. Net Efficiency

The pressure sensitivity analysis provides the optimal pressure, as shown in Figure 3-16. Based on the parasitic power requirements and the thermal energy recovery, the overall net efficiency varies as the operating pressure increases. At low pressures, gross power output, shown in Figure 3-5, increases and the overall compression work, shown in Figure 3-15, decreases. Thus, the net efficiency grows by 3 percentage point when the combustor pressure is raised from 1.1 bars to 11 bars. On the other hand, at high pressures, the efficiency decreases slowly as the parasitic compression power dominates the overall performance. As a result, the maximum efficiency is achieved in the vicinity of the 11 bar operating pressure.
3.4. Conclusions

The combustor operating pressure has a critical impact on the overall performance of the pressurized oxy-fuel combustion power cycle. With increasing the operating pressure, the thermal energy recovery from the flue gases grows as we recuperate more available latent enthalpy at a higher dew point. Increasing the energy recovery enables the pressurized oxy-fuel system to save some of the steam bleeding needed in the regeneration steps, and it subsequently produces more gross power. In addition, the fan compression work varies significantly with the operating pressure as the overall pressure drop across the steam generation unit and the recirculation pipe increases. While the air separation unit compression consumes more energy at higher pressures, the carbon dioxide purification and compression unit requires less auxiliary energy. Consequently, the overall parasitic
compression work demand is strongly dependent on the recirculation fan compression work. The maximum efficiency can be achieved in the vicinity of the 11 bar operating pressure.

The overall performance of the pressurized oxy-fuel combustion power cycle is a strong function of the thermal energy recovery from the flue gases and the energy requirement in the flue gas recirculation system. As the operating pressure increases, the variation in the net efficiency shows that the overall performance is sensitive to the change in these two parameters. Therefore, it is crucial to control the flue gas recirculation system and the thermal energy recovery rate in order to improve the overall performance of the pressurized oxy-fuel combustion power cycle with increasing the operating pressure.
Chapter 4 Sensitivity Analysis

4.1. Overview

Preliminary study on the characteristics of the pressurized oxy-fuel power cycle and the results from the base-case analysis suggest the important parameters that need further examination. Table 4-1 shows the variables with which we conduct sensitivity analyses.

Table 4-1 Sensitivity analysis parameters

<table>
<thead>
<tr>
<th>Variables</th>
<th>Base-Case</th>
<th>Min.</th>
<th>Max.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Combustion Temperature</td>
<td>1550 °C</td>
<td>1400 °C</td>
<td>1700 °C</td>
</tr>
<tr>
<td>Oxygen Purity</td>
<td>95%</td>
<td>90%</td>
<td>98%</td>
</tr>
<tr>
<td>Oxygen Concentration in Flue Gases</td>
<td>3%</td>
<td>3%</td>
<td>6%</td>
</tr>
<tr>
<td>HRSG Outlet Temperature</td>
<td>260 °C</td>
<td>240 °C</td>
<td>280 °C</td>
</tr>
</tbody>
</table>

Combustion in an environment with near pure oxygen provides a high combustion flame temperature, which could reach 3500 °C [13]. Thus, most oxy-fuel combustion power systems implement recirculation of the flue gases to reduce the combustion temperature to a reasonable level. This is a unique feature of oxy-fuel combustion. In this regard, the impact of the variation in the combustion temperature or the amount of the flue gas recirculation on the overall system performance should be investigated.
The air separation unit is the largest power consumer within the pressurized oxy-fuel combustion power system. Currently, the cryogenic separation process is the only option to meet the needs of the industrial size oxy-fuel combustion power plants. The cryogenic separation process is energy intensive and consumes 15% to 20% of the gross power output of the power island. If we can lower the energy penalty from the air separation unit, the overall cycle efficiency increases significantly. Because the power requirement of the air separation unit is closely related to the oxygen purity, we conduct a sensitivity analysis with respect to oxygen purity. This study leads to better understanding on the dependence of the pressurized oxy-fuel system on the oxygen purity or the air separation unit energy requirement.

In pressurized oxy-fuel combustion, the burning rate of char increases with operating pressure. This feature allows the system to utilize smaller amounts of oxygen to burn the fuel. In other words, the oxygen concentration in the flue gases can be lowered. It is an important characteristic of pressurized oxy-fuel combustion compared to conventional oxy-fuel systems. Therefore, the benefits from higher burning rate of char needs to be examined in terms of the thermodynamic performance.

The amount of the recycled flue gases depends on the extraction point within the power cycle. If the flue gases are extracted at lower temperature, the system requires smaller mass flow rate of recycled flue gases. Because the recirculation of the flue gases is a unique feature of oxy-fuel combustion, we need to study how much flue gases is required in oxy-fuel combustion and what impacts it has on the system performance. As a result, we conduct a sensitivity analysis with respect to the HRSG outlet temperature,
which is the extraction point in our base-case cycle, so as to investigate the relationship between the amount of the flue gases and the performance of the pressurized oxy-fuel power system.

### 4.2. Combustion Temperature

In oxy-fuel combustion, as mentioned in Section 1.2.1, the combustion flame temperature increases up to 3500 °C if no recirculation of flue gases exists. Currently available combustors cannot be sustained at this high temperature. As a result, typical oxy-fuel combustion power cycles employ the recirculation of the flue gases to reduce the combustion flame temperature. Depending on the amount of the recycled flue gases, the combustion temperature is determined. Conventional oxy-fuel combustion power cycles implement the recirculation of 70% to 80% of the flue gases [12]. In our base-case cycle for the pressurized oxy-fuel system, 88% of the flue gases extracted at the exit of the HRSG at 260 °C are recirculated to decrease the combustion temperature to 1550 °C.

#### 4.2.1. Methodology

We conducted a sensitivity analysis with respect to the combustion temperature, which is one of the important parameters in the oxy-fuel combustion power cycles (Figure 4-1). Based on the base-case design variables discussed in Chapter 2, we vary the recirculation ratio of the flue gases sent back to the combustor so as to change the
combustion temperature. Note that the change in the combustion temperature leads to the variation in the recirculation ratio of the flue gases flowing into the HRSG as well. The flue gases out of the combustor directly enter the steam generation unit after being mixed with the secondary recycled flue gases, as shown in Figure 2-2. Therefore, to keep the HRSG inlet temperature constant at 800 °C, the amount of the recycled flue gases to the HRSG should be also modified. By changing these recirculation ratios, the combustion temperature is varied from 1400 °C to 1700 °C.

Figure 4-1 Impact of the variation in the combustion temperature on the overall performance

4.2.2. Results

Although the combustion temperature is varied from 1400 °C to 1700 °C by changing the recirculation ratio of the flue gases flowing back to the combustor, the overall performance is not dependent on this variation. The HRSG inlet temperature, which is
fixed at 800 °C, leads to the constant inlet condition of the heat source to the steam generation unit. As seen in Figure 4-2, the amount of the recycled flue gases flowing into the HRSG changes according to the variation in the recirculation ratio of the flue gases entering the combustor in order to keep the HRSG inlet temperature constant. Consequently, the total recirculation ratio is kept constant at 88% and the thermal energy input into the steam generation unit remains at the same level as the combustor temperature is varied from 1400 °C to 1700 °C.

![Recirculation Ratio Chart](chart.png)

**Figure 4-2 Recirculation ratio of the flue gases to the HRSG and the combustor**

Based on the constant amount of thermal energy (flue gases) brought into the steam generation unit, the power island produces a constant gross power. In addition, because the variation only occurs within the flue gas recirculation loop, parasitic power requirements for the air separation unit and the carbon dioxide purification unit do not change. Both are located at the outside to the loop. Moreover, the fact that the total
mass flow rate of the flue gases is constant leads to no change in the fan compression work. Thus, the overall performance is independent of the variation in the combustion temperature. Thermodynamically, the combustion temperature does not have an impact on the performance of the pressurized oxy-fuel power system.

However, this parameter could be important in terms of the capital costs. The price of the combustor depends on the combustor temperature, and hence, the total capital investment would change as the combustor temperature is increased from 1400 C to 1700 C. Economically, the combustion temperature could affect the total capital costs.

4.3. Oxygen Purity

As discussed above, the largest power consumer within the pressurized oxy-fuel system is the air separation unit. This unit consumes nearly 20% of the gross power output and reduces the overall cycle efficiency by more than 5%. In Chapter 3, we found that it is critical to control the performance of the air separation unit in order to improve the pressurized oxy-fuel power cycle.

It is well known that the power requirement of the air separation unit is strongly dependent upon the oxygen purity. Many gas producers have done research on this issue to estimate the dependence of the air separation unit power consumption on the oxygen purity [19, 44]. Their results show that the power consumption increases significantly after 97% oxygen purity and drops as oxygen purity decreases. However, because the oxygen purity level also has an impact on the emission level that is discussed in the
following section, we cannot simply rely on the thermodynamic analysis results alone. Therefore, a sensitivity analysis is required to study the impact of the oxygen purity on the thermodynamic performance of the system and the emissions.

4.3.1. Methodology

The relationship between the oxygen purity and the power consumption of the air separation unit is difficult to estimate. This work is beyond the scope of our research. Hence, we adopt the result from one of air separation unit producers and use it as the base line for the sensitivity analysis. Figure 4-3 shows the normalized energy requirement of the air separation unit as the oxygen purity is varied from 85% to 100% (molar basis) [19].

![Figure 4-3 Normalized energy requirement of the air separation unit [19]](image)

Utilizing this data and using the fixed base-case design variables discussed in Chapter 2, the oxygen purity is varied from 90% to 98% (Figure 4-4). Note that the
adopted data only represent the air compression work, which accounts for most of the energy requirement of the air separation unit. Thus, we split the power consumption into the air compression work and the oxygen compression work. The former is scaled from the result of the base-case, where the air separation unit operates at 95% oxygen purity (Chapter 2), by using Figure 4-3, and the latter is fixed at the power consumption of the oxygen compressors estimated in the base-case analysis. By adding these two energy requirements, the overall power consumption of the air separation unit is calculated.

Figure 4-4 Scheme of the sensitivity analysis with respect to the oxygen purity
4.3.2. Results – Thermodynamic Performance

As the oxygen purity is increased, the quantities of impurities in the oxygen stream delivered from the air separation unit drop, and hence, we need smaller mass flow rate of the oxygen-enriched stream flowing into the combustor (see Figure 4-5). Note that the combustor needs a constant amount of pure oxygen to achieve 3% oxygen concentration in the flue gases. Fixed amounts of pure oxygen plus the lower concentration of impurities lead to lower mass flow rate of the oxygen-enriched stream. This feature enables the pressurized oxy-fuel system to reduce oxygen compression works as the oxygen purity is varied from 90% to 98%, as shown in Figure 4-6 (the red line).

Figure 4-5 Variation in the mass flow rate of the oxygen with increasing the oxygen purity
Figure 4-6 Dependence of the air (blue line) and oxygen (red line) compression works in the air separation unit on the oxygen purity

According to the published data shown in Figure 4-3, the air compression work is estimated based on the power consumption from the base-case analysis. Figure 4-6 (the blue line) describes the variation in the air compression work with increasing the oxygen purity. To achieve purity above 97%, most of argon should be removed as well as nitrogen. However, argon separation processes need a significant amount of energy because argon has a boiling point that is close to that of oxygen. This is why the air separation unit energy requirement increases considerably above 97% oxygen purity.

Combining the air compression work and the oxygen compression work that vary in the opposite directions as the oxygen purity changes, the overall air separation unit power requirement is estimated, as shown in Figure 4-7 (the blue line). Basically, the curve follows the trend of the air compression work, which is the dominant contributor to the overall power air separation unit power consumption. On the other hand, increase in
the oxygen compression work below 95% offsets the reduction in the energy used in the air compressors. Consequently, the minimum power requirement occurs at 95% oxygen purity.

Figure 4-7 Air separation unit (blue line) and carbon dioxide purification unit (red line) energy requirements with increasing the oxygen purity.

Figure 4-7 also shows the power consumption in the carbon dioxide purification and compression unit (the red line) as the oxygen purity is varied. Higher oxygen purity enables the system to capture more carbon dioxide from the flue gases, as explained in the following section. As a result, the mass flow rate of the capture-ready stream undergoing compression grows (Figure 4-8), so does the power requirement in the carbon dioxide purification unit with increasing the oxygen purity.
Figure 4-8 Mass flow rate of the capture-ready stream varying with the oxygen purity

According to the variation in the energy consumption in these two units, the overall parasitic power requirements change, as shown in Figure 4-9. The change in the air compression work dominates the overall power consumptions, and the curve shown in Figure 4-9 follows the trend of the power requirement of the air compressors (the blue line in Figure 4-6). Based on the constant gross power output, this result leads to the net efficiency variation shown in Figure 4-10. Thus, although the efficiency gain is small, it is desirable to reduce the oxygen purity to achieve better thermodynamic performance.
4.3.3. Results – Emissions

Higher oxygen purity leads to larger carbon dioxide capture rate. At high purity,
the feed gases into the combustor have smaller amounts of impurities including nitrogen and argon compared to the gases at low purity. In this case, the flue gases flowing into the carbon dioxide purification and compression unit have higher carbon dioxide concentration or smaller impurities. In other words, part of carbon dioxide purification duties is taken up by the air separation unit. Based on the fixed carbon dioxide concentration target of 96.5% in the capture-ready stream (see Section 2.3.5), this fact enables the system to capture more carbon dioxide from the flue gases through the purification unit. Figure 4-11 shows the impact of the oxygen purity on the carbon dioxide capture rate.

![Figure 4-11 Impact of the oxygen purity on the carbon dioxide capture rate](image)

On the other hand, higher oxygen purity also increases the oxygen concentration in the capture-ready stream, as shown in Figure 4-12. This is not a desirable result because the chances of oxidation occurring within the sequestration pipe grow with the oxygen concentration. Oxygen concentration should be minimized to achieve enough safety in
maintenance and operation of the sequestration system. Although larger amounts of carbon dioxide can be captured from the flue gases at high oxygen purity, we cannot simply increase the oxygen purity level because of the oxidation issue.

![Graph showing oxygen concentration in the capture-ready stream changing with oxygen purity]

Figure 4-12 Oxygen concentration in the capture-ready stream changing with the oxygen purity

4.3.4. Remarks

The sensitivity analysis with respect to oxygen purity demonstrates the need for further investigation on the thermodynamic performance and emission level. While lower oxygen purity allows the system to achieve higher overall cycle efficiency, higher purity is desirable in terms of the carbon dioxide capture rate. In addition, change in the oxygen purity has an impact on the oxygen concentration in the capture-ready stream that is transported to the sequestration site by the pipe. Therefore, more study should be done to find the optimum oxygen purity to meet all requirements.
4.4. Oxygen Concentration in the Flue Gases

In pressurized oxy-fuel combustion, the burning rate of char increases significantly, as discussed in Section 1.3, allowing the system to achieve near stoichiometric combustion; the equivalence ratio is 0.989 in our base-case. In other words, complete combustion can be achieved with less amounts of the oxidant stream.

Because it is an important characteristic of pressurized oxy-fuel combustion, the impact of higher burning rate of char on the overall system performance should be carefully examined. The change in the char burning rate makes it possible to vary the mass flow rate of the oxidant flowing into the combustor and affects the oxygen concentration in the combustion flue gases. In this regard, we conduct a sensitivity analysis with respect to the oxygen concentration in the flue gases to find out the dependence of the system performance on the amount of the oxidant stream.

4.4.1. Methodology

Based on the fixed coal flow rate of 30 kg/s corresponding to 874.6 MW\textsubscript{th} (HHV) or 839.1 MW\textsubscript{th} (LHV), oxygen concentration in the flue gases depends on the mass flow rate of the oxygen-enriched stream flowing into the combustor. To change this value, we need to vary the mass flow rate of the feed air entering the air separation unit because the amount of the oxygen-enriched stream is determined by the performance of this unit. By modifying the mass flow rate of the feed air, the oxygen concentration in the flue gases is varied from 3% to 6%.
Figure 4-13 shows the scheme of the sensitivity analysis with respect to the oxygen concentration in the flue gases. The base-case design variables used in other sensitivity analyses still apply.

4.4.2. Results – Thermodynamic Performance

Higher oxygen concentration requires larger mass flow rate of the oxygen stream, and it leads to increase in the amount of the feed air. Thus, the air separation unit consumes more energy by compressing more air and oxygen. As shown in Figure 4-14, the air separation unit power requirement (the blue line) grows linearly as the oxygen concentration increases.
Figure 4-14 Impact of the oxygen concentration in the flue gases on the energy requirements of the air separation unit (blue line) and the carbon dioxide purification and compression unit (red line).

Figure 4-14 also shows the reduction in the energy consumption rate of the carbon dioxide purification unit (the red line) associated with increasing the oxygen concentration in the flue gases. Compared to the result from the oxygen purity sensitivity analysis, larger oxygen concentration results in lower concentration of the carbon dioxide in the flue gases flowing into the purification unit and decreases the carbon dioxide capture rate, as shown in the following section. Therefore, the mass flow rate of the capture-ready stream drops as the oxygen concentration is varied from 3% to 6% (Figure 4-15).
Figure 4-15 Mass flow rate of the capture-ready stream varying with the oxygen concentration in the flue gases

According to these two power requirements, the overall parasitic power requirement varies, as shown in Figure 4-16. As expected, the air separation unit dominates the overall parasitic power consumption. Based on the constant gross power output, the overall cycle efficiency drops as the oxygen concentration in the flue gases increases (Figure 4-17). Therefore, thermodynamically, to improve the overall performance of the pressurized oxy-fuel power cycle, it is desirable to decrease the oxygen concentration or lower the mass flow rate of the feed air.
4.4.3. Results – Emissions

As mentioned above, higher oxygen concentration in the flue gases leads to lower carbon dioxide capture rate. The amount of the carbon dioxide produced from combustion
is constant because the coal flow rate is fixed at 30 kg/s. Larger oxygen stream mass flow rate yields higher oxygen concentration and more impurities including nitrogen and argon in the flue gases. In other words, the flue gases have lower carbon dioxide concentration. As a result, based on the fixed carbon dioxide concentration target of 96.5% in the capture-ready stream, we capture less carbon dioxide as the oxygen concentration in the flue gases is increased from 3% to 6%, as shown in Figure 4-18.

![Figure 4-18 Impact of the oxygen concentration in the flue gases on the carbon dioxide capture rate](image)

Figure 4-19 shows that the oxygen concentration in the capture-ready stream also grows as we inject more oxygen stream into the system. As discussed in the previous section, higher oxygen concentration in the sequestration-ready stream is undesirable and should be minimized before the stream is sent to the sequestration site. Therefore, higher concentration in the flue gases results in the undesirable condition of the capture-ready stream.
Combining these two results, in terms of the flue gas treatment, higher oxygen concentration in the flue gases should be avoided. By decreasing the oxygen-enriched stream injected into the combustor, we achieve larger carbon dioxide capture rate and lower oxygen concentration in the sequestration-ready stream.

4.4.4. Remarks

The parametric study discussed above shows that lower oxygen concentration in the flue gases is desirable in the pressurized oxy-fuel system. With decreasing amounts of the oxygen stream flowing into the combustor, we achieve gains in the overall cycle efficiency and in the carbon dioxide capture rate. Moreover, the capture-ready stream includes less oxygen as the oxygen concentration in the flue gases is decreased. Higher burning rate of char enables the pressurized oxy-fuel system to benefit from these features.
Therefore, in the pressurized oxy-fuel combustion system, the overall system performance and the emissions are improved by lowering the oxygen concentration in the flue gases.

4.5. HRSG Outlet Temperature

The flue gas recirculation is a unique characteristic of the oxy-fuel combustion power cycles. Typical oxy-fuel combustion systems recycle 70% to 80% of the flue gases [12]. In our base-case analysis, we found that the proposed pressurized oxy-fuel power cycle utilizes 88% of the flue gases at the exit of the HRSG to reduce the combustion temperature and the HRSG inlet temperature.

The amount of the recycled flue gases is dependent on the extraction point within the cycle. Recycling flue gases at lower temperature leads to smaller mass flow rate of the recycled flue gases required to decrease combustion and HRSG inlet temperatures. On the other hand, the minimum extraction temperature is bound by the acid dew point. Because the flue gases include acid gases, we should avoid their condensation within the HRSG. Instead they should be removed in the acid condenser. Our base-case uses the recycled flue gases at 260 °C, which is higher than the condensation temperature at 10 bars. In this regard, we conduct a sensitivity analysis with respect to the HRSG outlet temperature or the flue gas recirculation point so as to study the impact of the amount of the recycled flue gases on the overall performance.
4.5.1. Methodology

The heat exchanger surface area within the HRSG decides the outlet temperature. Larger thermal energy recovery within the HRSG by increasing the surface area results in lower outlet temperature (Figure 4-20). In our base-case cycle, the economizer is the last heat exchanger within the HRSG by which the flue gases pass (see Figure 2-8). Thus, we change the heat exchanger surface area of the economizer to vary the HRSG outlet temperature from 240 °C to 280 °C. Base-case design variables other than the HRSG outlet temperature are fixed during this sensitivity analysis.

![Diagram showing the sensitivity analysis with respect to the HRSG outlet temperature.](image)

Figure 4-20 Scheme of the sensitivity analysis with respect to the HRSG outlet temperature

4.5.2. Results

As expected, higher HRSG outlet temperature leads to larger mass flow rate of the recycled flue gases, as shown in Figure 4-21. Required amounts of the recycled flue gases grow gradually as the flue gas recirculation temperature or the HRSG outlet temperature is
increased. Based on a fixed coal flow rate and an oxygen stream from the air separation unit, the recycled flue gases at higher temperature need more gases to cool down the combustion temperature and the HRSG inlet temperature.

![Graph showing the dependence of the mass flow rate of the recycled flue gases on the flue gas recirculation temperature.](image)

**Figure 4-21** Dependence of the mass flow rate of the recycled flue gases on the flue gas recirculation temperature

According to the variation in the mass flow rate of the recycled flue gases, the pressure drop across the HRSG changes, as shown in Figure 4-22. Given nearly constant flue gas density and flue gas velocity, duct cross-sectional area is determined by the mass flow rate of the flue gases. An increase in the mass flow rate allows the heat exchangers to have larger cross-sectional area. Thus, the pressure drop across the HRSG lowers as the HRSG outlet temperature is increased from 240°C to 280°C. Based on the change in the pressure drop, the recirculation fan compression work also changes (see Figure 4-23).
Figure 4-22 Pressure drop across the HRSG according to the variation in the HRSG outlet temperature

Figure 4-23 Flue gas recirculation fan compression work versus the HRSG outlet temperature

The variations occur only within the flue gas recirculation loop, and thus, the power requirements of other components in the pressurized oxy-fuel system are not
affected by the change in the flue gas recirculation temperature. With constant power requirements of the air separation unit and the carbon dioxide purification unit, the overall cycle efficiency dictates the recirculation fan compression work, as shown in Figure 4-24. Consequently, although the efficiency gain is small, higher HRSG outlet temperature is desirable for the overall thermodynamic performance of the pressurized oxy-fuel power cycle.

![Figure 4-24 Impact of the change in the HRSG outlet temperature on the net efficiency](image)

Another interesting result is a large reduction in the HRSG heat exchanger surface area as its outlet temperature is increased. The surface area of the economizer is sensitive to the flue gas exit temperature and varies by nearly 60%, as seen in Figure 4-25. The fact that the price of the heat exchanger is a strong function of its surface area leads to a large reduction in the HRSG cost as the outlet temperature is increased from 240 °C to 280 °C.
However, we impose more heat duty on the acid condenser with increasing the HRSG outlet temperature. Thus, further study should be done on the thermodynamic conditions and the cost of the acid condenser.

![Normalized HRSG heat exchanger surface area varying with the HRSG outlet temperature](image)

**Figure 4-25** Normalized HRSG heat exchanger surface area varying with the HRSG outlet temperature

### 4.6. Conclusions

The pressurized oxy-fuel power system has important characteristics such as the recirculation of the flue gases, higher burning rate of char and the need for an air separation unit. To verify what impacts they have on the overall performance, we conduct sensitivity analyses with respect to the combustion temperature, oxygen purity, oxygen concentration in the flue gases and the HRSG outlet temperature.

The combustion temperature does not affect the thermodynamic performance of the
system. What decides the HRSG inlet temperature or thermal conditions of the heat source of the steam generation unit is the amount of the secondary flue gases to the HRSG. Thus, although the combustion temperature is varied by changing the recycled flue gases to the combustor, the overall cycle efficiency is independent of this temperature.

The oxygen purity makes two different contributions to the system performance and emissions. If the oxygen purity is lowered, we achieve higher overall cycle efficiency. While higher purity is desirable to capture more carbon dioxide from the flue gases, it increases the oxygen concentration in the capture-ready stream that is transported to the sequestration site by the pipe. Therefore, further study is required to find the optimal oxygen purity.

Lower oxygen concentration in the flue gases enables the pressurized oxy-fuel system to achieve higher cycle efficiency and capture more carbon dioxide. Moreover, the oxygen concentration in the capture-ready stream is decreased as well. Higher burning rate of char enables the pressurized oxy-fuel system to take advantage of these benefits.

The HRSG outlet temperature determines the mass flow rate of the recycled flue gases. The HRSG outlet temperature is the flue gas recirculation temperature. Higher outlet temperature means that larger amounts of the flue gases need to be recirculated. If more flue gases are recycled, the pressure drop across the HRSG is lowered, and hence, the recirculation fan compression work decreases. As a result, the overall cycle efficiency increases.
Chapter 5 Cost Estimation

5.1. Overview

Carbon capture and sequestration technologies are being considered as important options for combating the rise of carbon dioxide in the atmosphere. These technologies need new equipment and advanced processes that are cost-effective. While post-combustion capture requires thermo-chemical processes such as the amine scrubbing method, pre-combustion capture utilizes air separation units, gasifiers and reformers. Oxy-fuel combustion capture uses air separation units and carbon dioxide purification processes. All of them need a great amount of capital investment as well as advanced technologies.

The cost of the pressurized oxy-fuel system should be examined as part of determining its viability. In previous chapters, we discussed the proposed technologies or processes and important parameters within the power cycle. The thermodynamic analyses were carefully conducted to understand the system performance and sensitivity to operating variables. To make this technology more viable, the capital cost needs to be minimized. For this purpose, we should first estimate the cost of the pressurized oxy-fuel power system to gain better understanding of the different economic scenarios.

In this regard, we conduct a cost analysis to estimate the preliminary costs of the pressurized oxy-fuel power cycle. The cost estimates include the price of each component or process within the power system. Based on the results, we find which processes are
cost-intensive. Then, the total capital cost is compared to published data so as to verify our costing method. This study enables us to understand the competitiveness of the pressurized oxy-fuel power system compared to other options.

### 5.2. Methodology

As discussed in Chapter 2, our base-case consists of five primary units including the air separation unit, the pressurized coal combustor, the steam generation unit, the power island and the carbon dioxide purification and compression unit. The cost estimation is based on this organization. The costs of the air separation unit, the steam generation unit and the power island are estimated by the Thermoflex® costing module. The carbon dioxide purification unit dictates the process proposed by white el al [35], and its cost estimates from their report [34] is used in this study. Because we utilize the pressurized coal combustor, ISOTHERM®, developed by ITEA with the support of ENEL, the price of the combustor is given by ENEL. ENEL’s cost assessment includes the coal water slurry (CWS) system as well.

The cost estimates are compared to the published data from the NETL report [14] in order to evaluate our cost estimation and costing method. NETL’s cost estimates are based on the mid-west region of the United States; we also use this area as the reference to calculate labor costs, construction costs, transportation costs and so on. In addition, all of the costs are escalated to January 2009 basis by assuming 4% escalation. Because ENEL’s cost assessment is based on Euro, we use a fixed currency rate of 1.5 USD/Euro.
5.3. Cost Estimates

Figure 5-1 shows the specific costs of components or processes within the pressurized oxy-fuel power cycle. The specific cost is defined as the capital cost of the equipment divided by the net power output. The HRSG represents four heat exchangers shown in Figure 2-8. Note that it does not include the cost of the acid condenser because it is difficult to estimate. The miscellaneous includes accessory electrical systems, building structure and instrumentation/control systems.

As seen in Figure 5-1, three primary units: the air separation unit, the carbon dioxide purification unit and the pressurized coal combustion system, account for most of the capital investment. Compared to conventional power plants, these components are either added to the cycle or significantly modified for the pressurized oxy-fuel power system.

From the thermodynamic analyses, we know that the air separation unit and the carbon dioxide purification unit are the two largest power consumers within the cycle. Figure 5-1 shows that these two units also require large amounts of capital investment. Thus, as expected, the air separation unit and the carbon dioxide purification unit are energy- and cost-intensive processes within the pressurized oxy-fuel power system. Moreover, the price of the pressurized coal combustion system is more expensive than any other components within the cycle. Therefore, we need to focus on the costs and the performance of these three primary units so that we can improve the pressurized oxy-fuel power system.
5.4. Cost Classification and Comparison

Although our cost estimates provide a reasonable base line for the economic analysis of the proposed system, we do not know how big or small those values are until we compare them with reference numbers. In this regard, we compare our values with the cost estimates by NETL. This comparison gives us some confidence in our cost estimates of the pressurized oxy-fuel power system. In addition, we can validate our costing method and find the strengths and weaknesses of this method.

Before the comparison of the cost estimates with the data from NETL, we categorize the components into four groups, which are the steam generation, the power...
island, the flue gas disposal and the miscellaneous. The first three groups represent
primary systems within power plants. This categorization allows us to validate our
costing method more effectively and find where our cost estimates have large differences
compared to the conventional prices. Table 5-1 shows these four groups. Note that the
air separation units in oxy-fuel plants are categorized as a part of the steam generation.

Table 5-1 Cost classification groups

<table>
<thead>
<tr>
<th>Steam Generation</th>
<th>Power Island</th>
<th>Flue Gas Disposal</th>
<th>Misc.</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Air-Fired Plant</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>✓ Furnace Boiler</td>
<td>✓ Steam Turbine</td>
<td>✓ deSO$_x$</td>
<td>✓ Instrumentation &amp; Control</td>
</tr>
<tr>
<td>✓ Coal Handling</td>
<td>✓ Deaerator</td>
<td>✓ deNO$_x$</td>
<td>✓ Administration-Relevant Bldg.</td>
</tr>
<tr>
<td><strong>Oxy-Fuel Plant</strong></td>
<td>✓ Pump</td>
<td>✓ Filter</td>
<td></td>
</tr>
<tr>
<td>✓ Air Separation Unit</td>
<td>✓ Feedwater Heater</td>
<td></td>
<td></td>
</tr>
<tr>
<td>✓ Combustor</td>
<td>✓ Water-cooled Condenser</td>
<td></td>
<td></td>
</tr>
<tr>
<td>✓ HRSG</td>
<td></td>
<td>✓ CO$_2$ Purification and Compression Unit</td>
<td></td>
</tr>
<tr>
<td>✓ CWS System</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

As shown in Table 5-2, the prices of the four groups in the pressurized oxy-fuel
power cycle are compared to three other power plants. Note that NETL’s air-fired power
plant with capture utilizes amine-based absorption and its atmospheric oxy-fuel case uses
dry coal delivery. Because air separation units account for large capital costs within oxy-
fuel systems, the steam generation of NETL’s atmospheric oxy-fuel case and our
pressurized oxy-fuel system needs more than the double capital investment. This is why
oxy-fuel systems are cost-intensive compared to conventional power plants.
On the other hand, the costs of flue gas disposal in oxy-fuel systems are significantly less than that of amine absorption processes in air-fired power plants. Purification processes in oxy-fuel systems mostly depend on thermodynamic methods, whereas post-combustion capture requires complicated thermo-chemical processes that are cost-intensive and consume large amounts of energy. The fact that the flue gases of oxy-fuel combustion predominantly consist of carbon dioxide and condensable water makes it possible to have more economic purification processes.

Compared to NETL’s atmospheric oxy-fuel case, our pressurized oxy-fuel power system has large differences in the steam generation and the power island. To find out the reasons why these differences exist, we break these two groups into their components. Component by component comparison is conducted to understand the deviation from the published data.

Table 5-3 shows the detailed cost information of the steam generation in each
power plant. The largest difference occurs in the coal handing unit or coal water slurry system. As mentioned above, NETL’s oxy-fuel system uses dry coal delivery, whereas our pressurized oxy-fuel cycle utilizes the wet coal water slurry system. The latter requires considerably more capital costs, as seen in Table 5-3. In addition, the pressurized coal combustor plus the HRSG need larger amounts of costs than that of conventional boilers. However, this difference could be decreased because the ISOTHERM® combustion is a newly developed technology and needs to be optimized before it is used in commercial oxy-fuel power plants. Note that conventional boilers have been improved over many years. Another big difference exists in the air separation unit. The pressurized oxy-fuel system utilizes high-pressure oxygen compressors that conventional atmospheric oxy-fuel systems do not need. As a result, the air separation unit in the pressurized system needs more capital costs.

Table 5-3 Detailed costs of the steam generation

<table>
<thead>
<tr>
<th></th>
<th>NETL</th>
<th>MIT/ENEL</th>
<th>Coal Handling</th>
<th>Boiler Combustor + HRSG</th>
<th>Air Separation Unit</th>
<th>Total</th>
</tr>
</thead>
<tbody>
<tr>
<td>NETL Air-Fired w/o CO₂ Capture</td>
<td>61</td>
<td>540</td>
<td></td>
<td></td>
<td>0</td>
<td>601</td>
</tr>
<tr>
<td>NETL Air-Fired w/ CO₂ Capture</td>
<td>76</td>
<td>683</td>
<td></td>
<td></td>
<td>0</td>
<td>759</td>
</tr>
<tr>
<td>NETL Atmospheric Oxy-Fuel</td>
<td>79</td>
<td>672</td>
<td>456</td>
<td></td>
<td></td>
<td>1,207</td>
</tr>
<tr>
<td>MIT-ENEL Pressurized Oxy-Fuel</td>
<td>193</td>
<td>760</td>
<td>547</td>
<td></td>
<td></td>
<td>1,500</td>
</tr>
</tbody>
</table>
Differences in the costs of the power island arise from the feedwater system and the cooling water system. As shown in Table 5-4, the pressurized oxy-fuel cycle's feedwater system needs less than one tenth of atmospheric oxy-fuel system's cost. As discussed in Chapter 2, the pressurized oxy-fuel power cycle recovers more thermal energy than conventional atmospheric oxy-fuel cases. This feature enables the pressurized system to save steam bleeding used for heating up the feedwater and reduce the heat duty of the regeneration system. As a consequence, the pressurized oxy-fuel system has less feedwater heaters and achieves a considerable reduction in the cost of the feedwater system. Moreover, our pressurized system does not have a cooling tower, and thus, no capital costs are needed for this unit. Note that our system is based on the Mediterranean sea-water-cooled condenser.

<table>
<thead>
<tr>
<th>($/kW)</th>
<th>Feedwater Heater + Pump + Deaerator</th>
<th>Steam Turbine + Water-cooled Condenser</th>
<th>Cooling Water System</th>
<th>Accessory Electric</th>
<th>Bldg. Structure</th>
<th>Total</th>
</tr>
</thead>
<tbody>
<tr>
<td>NETL Air-Fired w/o CO₂ Capture</td>
<td>115</td>
<td>180</td>
<td>58</td>
<td>75</td>
<td>45</td>
<td>473</td>
</tr>
<tr>
<td>NETL Air-Fired w/ CO₂ Capture</td>
<td>158</td>
<td>205</td>
<td>102</td>
<td>118</td>
<td>51</td>
<td>634</td>
</tr>
<tr>
<td>NETL Atmospheric Oxy-Fuel</td>
<td>184</td>
<td>242</td>
<td>71</td>
<td>136</td>
<td>61</td>
<td>694</td>
</tr>
<tr>
<td>MIT-ENEL Pressurized Oxy-Fuel</td>
<td>13</td>
<td>234</td>
<td>0</td>
<td>136</td>
<td>62</td>
<td>445</td>
</tr>
</tbody>
</table>

Table 5-4 Detailed costs of the power island
Consequently, the total specific capital cost of the pressurized oxy-fuel system is somewhat smaller than that of the atmospheric oxy-fuel power cycle, as shown in Figure 5-2. However, because our system is not optimized yet, the total capital cost could be further decreased. Even at this point, in terms of capital costs, the pressurized oxy-fuel power system is a promising option for power plants with carbon dioxide capture and sequestration.

Figure 5-2 Total specific capital costs of power plants

5.5. Conclusions

The pressurized oxy-fuel power cycle has three largest power consumers, the air separation unit, the carbon dioxide unit and the pressurized coal combustor. These units account for nearly 70% of the total capital cost. Thus, we need to focus on the
performance and the costs of these units to improve the pressurized oxy-fuel power system.

Compared to conventional air-fired power plants with carbon dioxide capture, oxy-fuel systems require more capital investment in air separation units and less in carbon dioxide purification processes. While air separation units utilize cost-intensive processes, oxy-fuel systems’ carbon dioxide purification units are cheaper and less complicated processes, based on high concentration of carbon dioxide and condensable water in the flue gases.

In the pressurized oxy-fuel system, the air separation unit requires high-pressure oxygen compressors, and hence, more costs are needed. However, larger amounts of thermal energy recovery lead to smaller heat duty of the regeneration system and less feedwater heaters. This saves significant amounts of costs. As a result, the overall capital cost of the pressurized oxy-fuel power system is somewhat smaller than that of conventional atmospheric oxy-fuel power cycles. However, its cost could be further lowered through improvement and optimization of the system.
Chapter 6 Conclusions

6.1. Summary

We analyzed a pressurized oxy-fuel combustion power cycle. Base-case analysis shows that we have an efficiency gain over atmospheric pressure oxy-fuel combustion achieved by recovering more flue gas thermal energy and reducing parasitic power requirements. Pressure parametric study provides the optimal combustor operating pressure, and a better understanding of the pressure dependence of the system performance. Sensitivity analysis with respect to combustion temperature, oxygen purity, oxygen concentration in the flue gases and HRSG outlet temperature demonstrates the impact of these operating variables on the overall system performance. These variables are related to the unique characteristics of pressurized oxy-fuel combustion, such as the higher burning rate of char and the recirculation of the flue gases. Cost analysis yields preliminary cost estimates of the pressurized oxy-fuel system and lets us find which units or processes should be focused and optimized.

6.2. Future Work

6.2.1. Deaerator and Acid Condenser

The high-pressure deaerator is an important component within the pressurized oxy-
fuel power cycle. Through the pressure parametric study in Chapter 3, we found that the flue gas thermal energy recovery rate is dependent upon the deaerator feedwater inlet temperature. In other words, its operating conditions could be limiting factors for the overall cycle efficiency. Therefore, the relationship between its operating conditions and the system performance should be further investigated.

The acid condenser is the component in which we recover the latent enthalpy of water. Because the latent enthalpy recovery is a critical feature of the pressurized oxy-fuel power cycle, the operating conditions and characteristics of the acid condenser should be studied further. The pressure parametric study shows that the pinch point temperature movement could limit the amount of latent enthalpy recovery and overall cycle efficiency. In addition, we took only the pure water condensation into account. However, the flue gases include acid gases as well, which affect the condensation temperature. Thus, condensation with acid gases and its impact on the thermal energy recovery rate should be examined in the future research.

6.2.2. Sensitivity Analysis with Respect to Steam Conditions and Coal Characteristics

In the sensitivity analysis in Chapter 4, we concentrated on the unique features of the pressurized oxy-fuel combustion power system. However, overall system performance is also strongly dependent on other variables in the power island and the chemical energy input to the system. Steam conditions including the turbine inlet pressure,
the inlet temperature and reheat temperatures are key design variables in the power island. They have an impact on the gross power output and the overall cycle efficiency. Hence, a sensitivity analysis is required to determine the dependence of the pressurized oxy-fuel system on the steam conditions. Moreover, coal characteristics define the chemical energy input to the system. Depending on the concentrations of carbon, sulfur, ash and water, the amount of chemical energy input is determined. They also affect the amount of the thermal energy recovery and the energy requirements in the purification units. Thus, further research should be done with these coal characteristics.

6.2.3. Improvement on the Costing Method

The cost estimates in Chapter 5 are preliminary. The costing module has not been fully validated, and some costs depend on the published data because they are difficult to estimate through our costing module at this point. To achieve more reliable cost estimates, these issues should be addressed. The costing method needs to be validated through further investigation and comparison with other costing modules and published data. The Aspen Plus® costing module could be also used to estimate the prices of the non-standard processes, such as the air separation unit and the carbon dioxide purification and compression unit.

6.2.4. Optimization

We proposed the base-case cycle and its important operating parameters. Based
on the cycle, we conducted sensitivity analyses to understand the dependence of the system performance on the design variables. We also estimate the capital costs of the components within the pressurized oxy-fuel power cycle. These studies are all focused on the characteristics of the proposed system. To achieve more viable and promising option for power plants with carbon dioxide capture and sequestration, an optimization step should be taken into account. Based on the finding from the analyses in this thesis, we could optimize the system in terms of the thermodynamic performance and the costs.
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[34] BERR Carbon Abatement Technologies Programme. Future CO₂ capture technology


