ANALYSIS OF OXY-FUEL COMBUSTION POWER CYCLE UTILIZING A PRESSURIZED COAL COMBUSTOR

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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 in which fuels are burned in an environment of oxygen and recycled combustion gases. In this paper, an oxy-fuel combustion power cycle that utilizes a pressurized coal combustor is analyzed. We show that this approach recovers more thermal energy from the flue gases because the elevated flue gas pressure raises the dew point and the available latent enthalpy in the flue gases. The high-pressure water-condensing flue gas thermal energy recovery system eliminates the low-pressure steam bleeding which is typically used in conventional steam cycles and enables the cycle to achieve higher efficiency. The pressurized combustion process provides the purification and compression unit with a concentrated carbon dioxide stream. For the purpose of our analysis, a flue gas purification and compression process including de-SO_x, de-NO_x, and low temperature flash unit is examined. We compare a case in which the combustor operates at 1.1 bars with a base case in which the combustor operates at 10 bars. Results show nearly 3 percentage point increase in the net efficiency for the latter case.

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1. INTRODUCTION

Most current energy conversion systems utilize fossil fuels and discharge large quantities of carbon dioxide into the atmosphere. 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]. There is a growing concern that the use of fossil fuels and the associated carbon dioxide emissions are contributing to global warming. 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].

As energy use grows, concerns over global warming may lead to imposing limits on greenhouse gas emissions from fossil fuel plants. This has stimulated extensive research on the subject of carbon capture and sequestration. The International Energy Agency estimates that carbon capture and sequestration could play an important role in decreasing carbon dioxide emissions [4]. 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.

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 recycled gases are used to control the flame temperature and replace the nitrogen separated prior to combustion [5]. 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 [6, 7]. 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 [8].

Recent research shows that pressurized oxy-fuel combustion systems 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, ISOTHERM®, and has already

performed several tests on a 5 MW$_{th}$ scale, working at 4 bars [9-11]. The pressurized combustion technology achieves high carbon dioxide purity in the flue gases and reduces the energy penalties [12]. CANMET and ThermoEnergy also conducted technical and economic studies on the pressurized oxy-fuel combustion system [13-19]. Their approach shows the net efficiency gain and the reduction in the capital cost and the cost of electricity when using high pressure oxy-fuel combustion.

In this study, 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. 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. 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 a high pressure level 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$_x$, de-NO$_x$, and a low temperature flash unit to prepare the carbon dioxide stream for transportation to an EOR or a sequestration site.

The methodology used in the analysis is described briefly in Section 2, and the detailed process descriptions and operating conditions are discussed in Section 3. In Section 4, the results of the cycle simulation are presented and discussed. Section 5 includes conclusions.

2. METHODOLOGY

Two commercial simulation packages, Thermoflex® and Aspen Plus®,[1] are used to model the pressurized

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[1] Thermoflex® and Aspen Plus® are registered trademarks of Thermoflow LTD and Aspen Technology, Inc., respectively.

oxy-fuel combustion power cycle. While the former focuses more on the modeling of steam generation and the power island, the latter is used to model the air separation unit and the flue gas treatment unit including the carbon dioxide purification and compression unit. Based on the fixed oxygen purity target of 95% and the mass flow rate of oxygen needed in the oxy-coal combustor (see Section 3.1), the power consumption and the heat balance of the air separation unit are estimated, using Aspen Plus®. Utilizing the oxygen stream information from Aspen Plus® and the coal analysis data given in Table 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® so as 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).

The proposed approach focuses on the base case at a 10 bars combustor operating pressure and compares it with the near atmospheric pressure combustor case. 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.

3. PRESSURIZED OXY-FUEL COMBUSTION POWER CYCLE

Figure 1 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. As shown in Figure 1, 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 1, 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 1.

After leaving the second feedwater pump, the feedwater is heated regeneratively to state 6. Next, it 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 and the exhaust stream, state 22 and state 23, respectively.

3.1. **Air Separation Unit**

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 [20-23]. 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 the two distillation column system, the cryogenic air separation unit delivers an oxygen stream with 95% purity (by volume) at a power consumption rate comparable to that of commercial air separation units. The specific energy of producing O_2 in the base case air separation unit is 0.245 kwh/kg-O_2, while the commercial air separation units consume 0.247 kwh/kg-O_2 [23] or 0.244 kwh/kg-O_2 [24]. Note that the reference values do not necessarily represent the most up-to-date technology of different gas producers. The air separation unit operates with air

compressed to 5.5 bars and delivers the pressurized gaseous oxygen-enriched stream at 10 bars to the combustor. The air separation process is shown in Figure 2.

The oxygen delivery temperature to the combustor is controlled to prevent the 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 1, 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 needs to 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 raw flue gases exiting the combustor have 3% oxygen on a molar basis, as shown in Figure 3.

3.2. Pressurized Coal Combustor

The pressurized coal combustor used in this work is based on the technology developed by ITEA with the support of ENEL [10]. Combustion takes place at elevated pressures and at 1400 to 1600 ºC. Note that stoichiometric combustion of coal in pure oxygen reaches up to 3500 ºC [25]. While 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 1.

In the proposed oxy-fuel process, the combustor temperature is maintained at 1550 ºC, by premixing the oxygen stream with the recycled flue gases. 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$_e$, the combustor is assumed to lose 2% of the lower heating value of the fuel to the water-cooled walls of the combustor. The energy lost during combustion is used to heat the feedwater stream, as shown in Figure 1. 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. Steam from the high-pressure steam turbine

is also injected into the pressurized combustor to atomize the slurry particles. 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 ºC.

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 ºC at 250 bars and is delivered to the power island. The superheat temperature of 600 ºC 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 ºC each, as shown in Figure 4.

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 ºC 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 5.

Moreover, as shown in Figure 1, 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 a large amount of latent enthalpy of water.

3.4. Power Island

A supercritical Rankine cycle has been chosen for the power cycle. Within the boundary of the supercritical Rankine cycle which has an inlet steam flow at 250 bars and 600 ºC, the power island has two reheat steam flows at 620 ºC. The exhaust steam is condensed at 33 °C 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 6, 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 1, 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 an increase in the inlet temperature of 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 incoming feedwater to the deaerator.

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 [26]. 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 7. This process has been chosen only for the simulation purposes reported in this paper. Other processes can be considered and adopted in future research.

To remove NO_x and SO_x 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 NO_2-catalytic conversion of SO_2 to sulfuric acid with increasing pressure [27], SO_x removal process operates in the water-wash column at 15 bars. It separates out all the SO_2 and SO_3 as sulfuric acid, as well as almost half of the remaining water content in the flue gases. After the de-SO_x unit, the flue gases are compressed to 30 bars and introduced into the next water-wash column. Here, more than 90% (by mass) of NO_x is removed as nitric acid. Most NO_x produced from the high temperature

combustion is in the form of NO and it must be converted to NO_2 in order to remove it as nitric acid. The reaction rate for converting NO to NO_2 increases as the pressure is raised and the temperature is lowered [26]. Hence, the removal of NO_x is attained in the water-wash column at 30 bars. The corresponding reactions are as follows:

$$NO + \frac{1}{2}O_2 \leftrightarrow NO_2 \tag{1}$$

$$NO_2 + SO_2 + H_2O \leftrightarrow NO + H_2SO_4 \tag{2}$$

$$2NO_2 + H_2O \leftrightarrow HNO_2 + HNO_3 \tag{3}$$

$$3HNO_2 \leftrightarrow HNO_3 + 2NO + H_2O \tag{4}$$

After cleaning up NO_x and SO_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 ºC which is close to the carbon dioxide triple point, -56 ºC. This cooling process produces two different streams, a capture-ready carbon dioxide stream 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.

4. PROCESS EVALUATION

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

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, 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 1, is not compressed and thermal energy sources are not

9

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, the 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 1. Consequently, the atmospheric oxy-fuel power cycle is based on a 100 °C oxygen delivery temperature. Moreover, in the atmospheric pressure combustor system, the steam from the low-pressure turbines is used to heat up the feedwater leaving the second feedwater pump. The Table 3 shows the conditions for each of the major states for both cases.

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) [23], 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. It can be explained through the outlet temperature of the acid condenser represented as stream 21 in Table 3. With the same inlet thermal energy sources, the acid condenser of the atmospheric case yields an outlet temperature, 73.19 °C, higher than that of the pressurized case, 60.49 °C.

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 Table 3 and Figure 1. 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.

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 8. A significant reduction in the steam bleeding increases the power generation from the turbines. As shown in Figure 1 and Figure 8, 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 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 [16], the proposed approach gives a 7 percentage point higher gross efficiency.

Details of the power generated and consumed in the atmospheric and the pressurized oxy-fuel power cycles are shown in Figure 9. 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 in the air separation unit than the atmospheric oxy-fuel power cycle by 4 percentage point of the gross power output. If we consider the power demand without the oxygen compression work, it becomes consistent with the atmospheric case or previous oxy-fuel research [24].

Figure 9 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 1, are already at high pressure, reducing the compression work required for liquefaction. This feature is advantageous for the overall performance. Table 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 save 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 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 for the recycled flue gases, described as fan compression work in Figure 9. Energy requirement for the fan drops by 7.6 MW_e 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 10 shows the contributions of the different units to the improvement of the net efficiency.

5. CONCLUSION

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 a 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 MW_e. 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.

6. ACKNOWLEDGEMENTS

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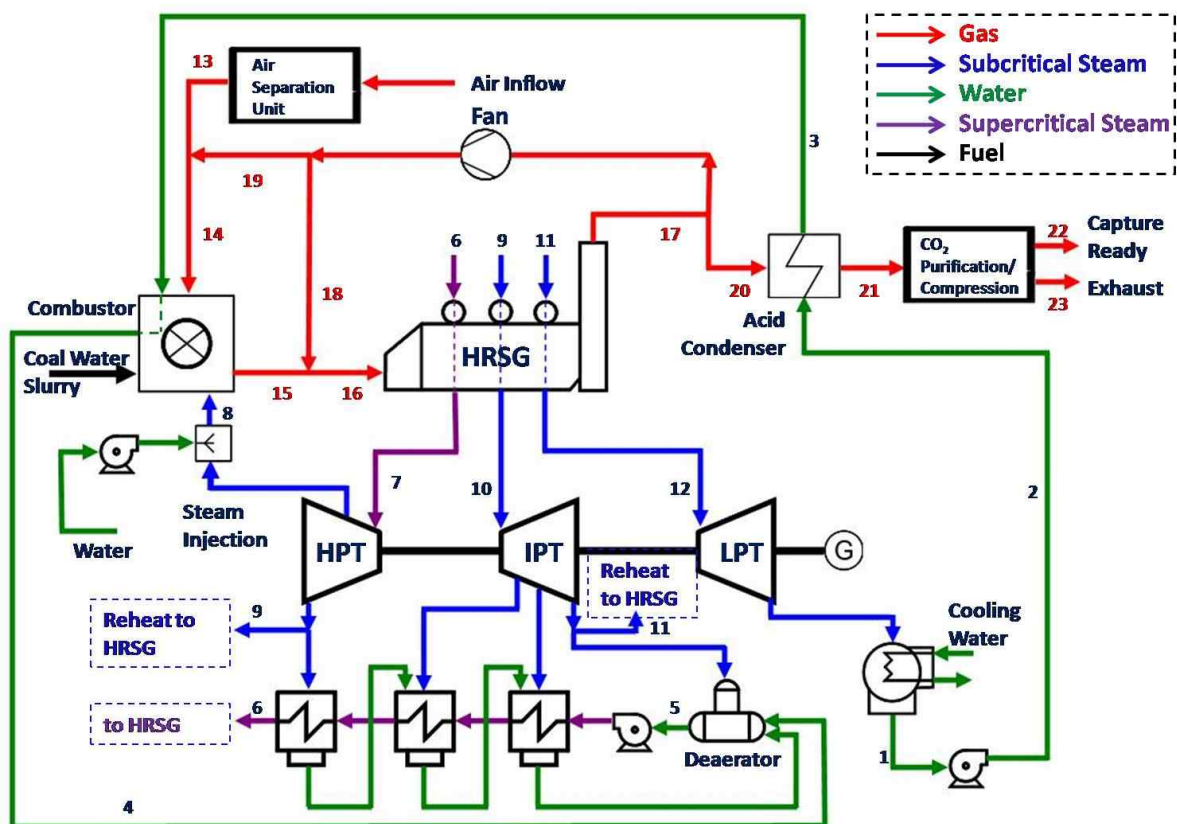
List of Figures

Figure 1 Overall Process Layout for Oxy-Fuel Combustion Power Cycle Utilizing a Pressurized Coal Combustor (edited from [12])
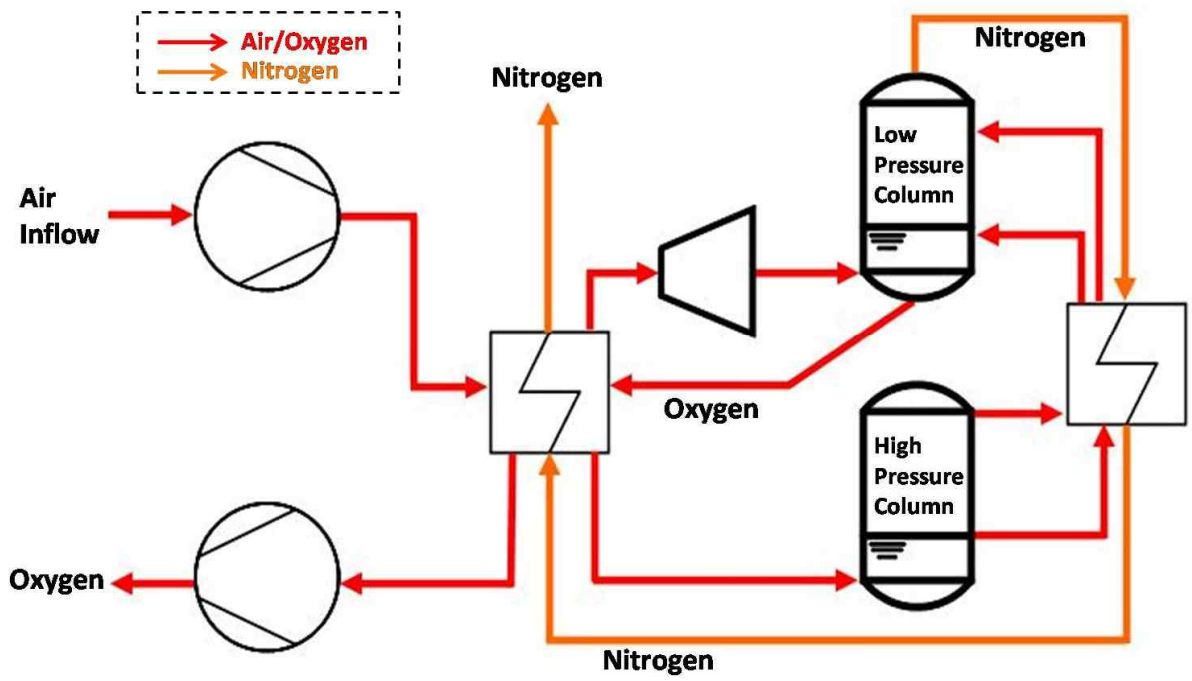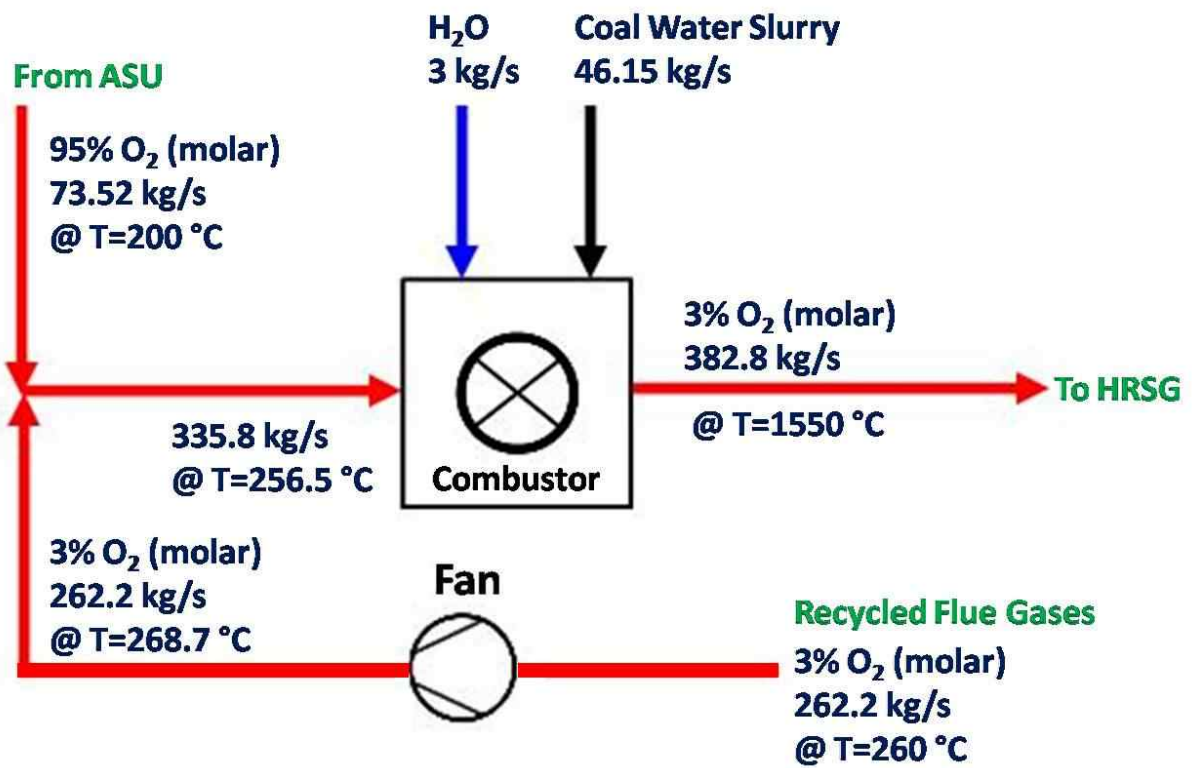
Figure 2 Air Separation Unit

Figure 3 Mass Flow Rate and the Temperature of Streams across the Pressurized Coal Combustor
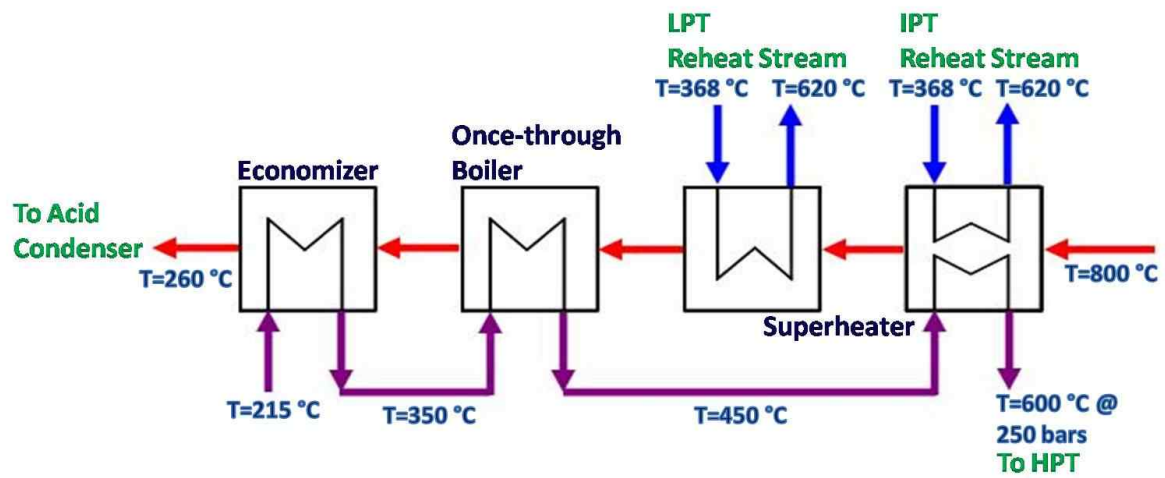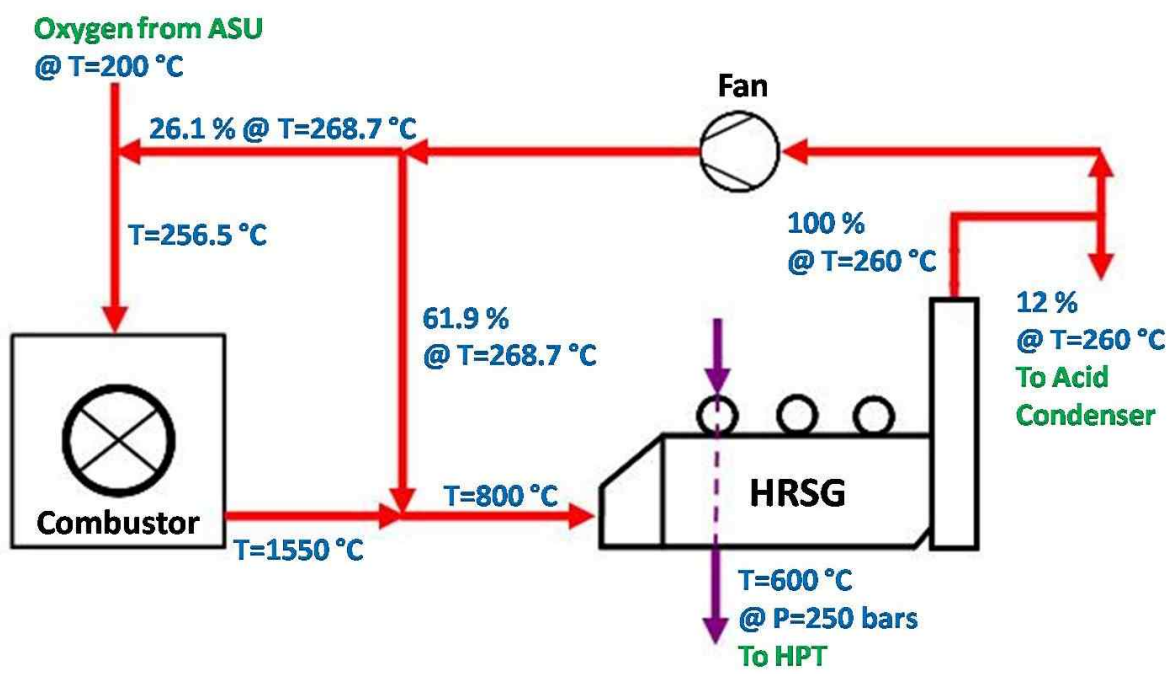
Figure 4 Steam Path within the HRSG

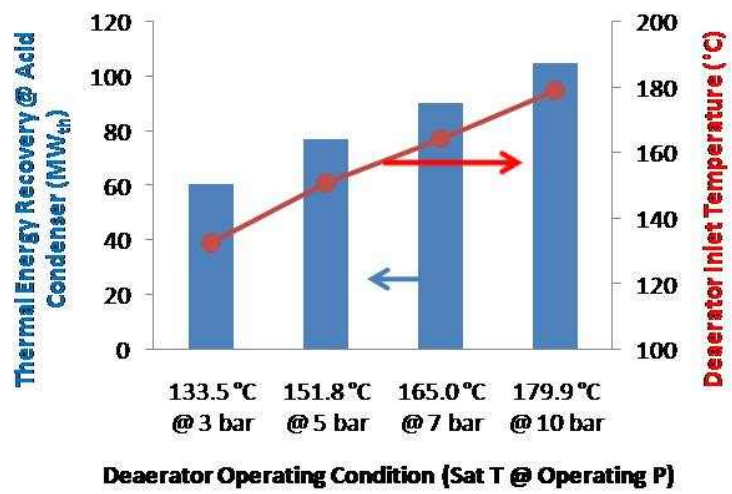**Figure 5 Stream Information across the Pressurized Coal Combustor and the HRSG (patent pending)**

Figure 6 Dependence of Thermal Energy Recovery on the Deaerator Operating Condition
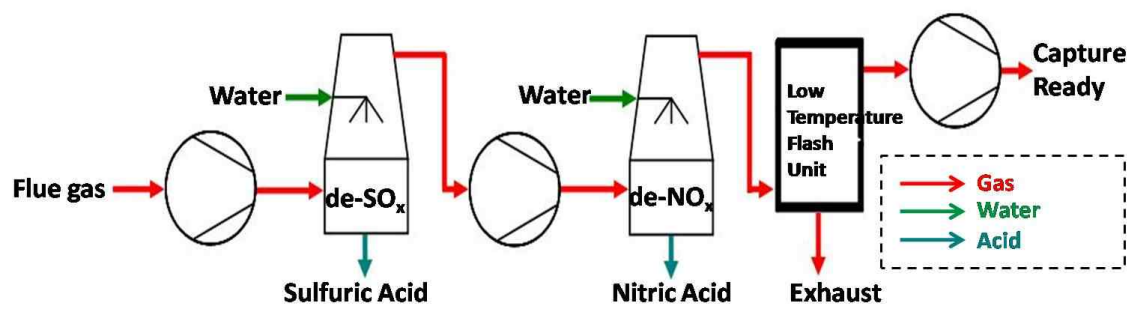
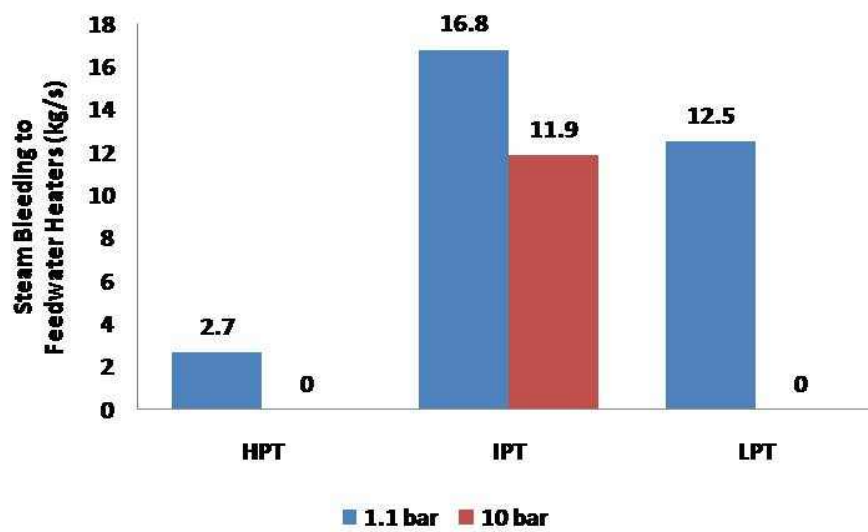**Figure 7 Carbon Dioxide Purification and Compression Unit**
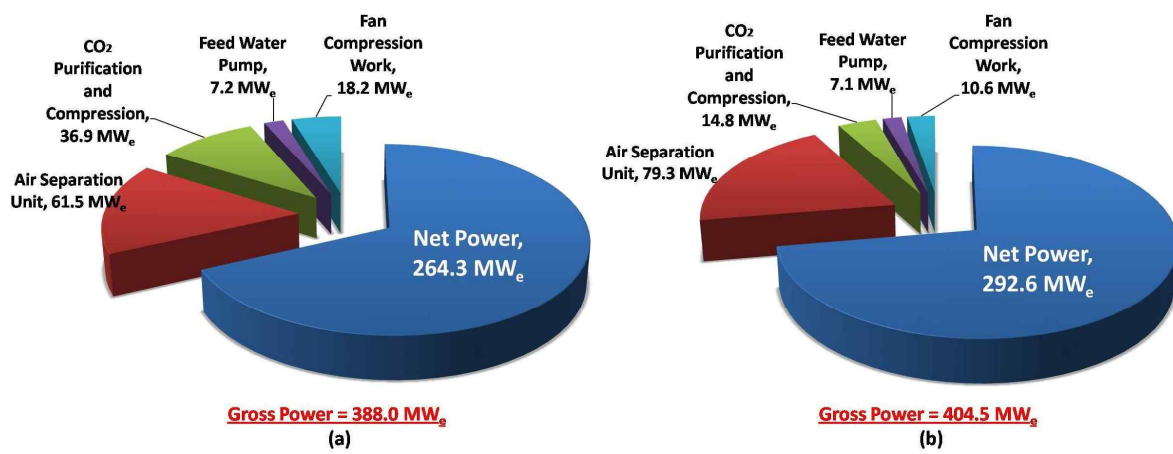
Figure 8 Steam Bleeding from Turbines

Figure 9 Net Power and Parasitic Power Demand:
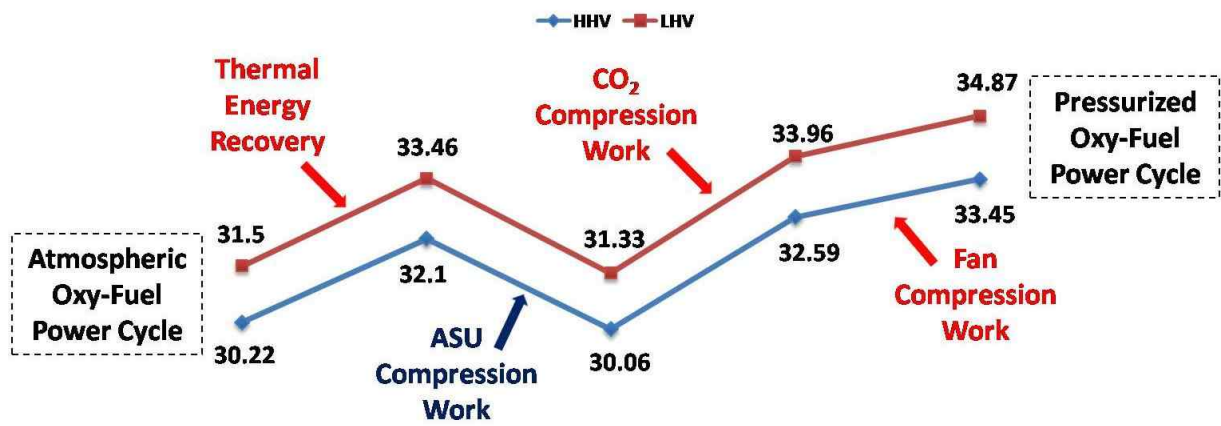**(a) the Atmospheric Oxy-Fuel Power Cycle and (b) the Pressurized Oxy-Fuel Power Cycle**

Figure 10 Net Efficiency Gain and Loss through Each Unit

List of Tables

Table 1 Coal Analysis Data

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

Table 2 Base Case Design Variables

Design Variables	Value
1. Air Separation Unit	
Oxygen Purity (mol %)	95%
Oxygen in the Flue Gases (mol %)	3%
Oxygen Delivery Temperature	200 °C
2. Pressurized Coal Combustor	
Combustor Pressure	10 bars
Combustor Temperature	1550 °C
Combustor Thermal Energy Loss	2%
Slurry Water (wt %)	35%
Steam Injection (wt %)	10%
3. Steam Generation	
Inlet temperature of HRSG	800 °C
Outlet Temperature of HRSG	260 °C
4. Power Island	
Turbine Inlet Pressure	250 bars
Turbine Inlet Temperature	600 °C
Reheat Temperature	620 °C
Deaerator Pressure	10 bars
Condenser Pressure	0.05 bars
5. Carbon Dioxide Purification and Compression	
CO_2 Compression Pressure	110 bars

Table 3 Stream Results for (a) the Atmospheric Oxy-Fuel Power Cycle and (b) the Pressurized Oxy-Fuel Power Cycle

#	Pressure (bar)		Temperature (°C)		Mass Flow Rate (kg/s)	
	(a)	(b)	(a)	(b)	(a)	(b)
1	0.41	0.41	32.6	32.6	195.1	198.1
2	11.2	11.2	32.7	32.9	195.1	198.1
3	11.2	11.2	102.5	158.7	195.1	198.1
4	11.2	11.2	121.9	177.2	195.1	198.1
5	11.2	11.2	179.9	179.9	211.9	210
6	250	250	218	215	211.9	210
7	250	250	600	600	211.9	210
8	70	70	316	316	3	3
13	1.1	10	105	201.2	73.52	73.52
14	1.1	10	246.7	256.5	334.4	335.8
15	1.1	10	1549.3	1549.7	381.4	382.8
16	1.1	10	800	800	1011.1	1004.7
17	0.983	9.351	260.6	259.7	1011.1	1004.7
18	1.1	10	275.8	268.7	629.6	621.9
19	1.1	10	275.8	268.7	260.9	262.2
20	0.983	9.351	260.6	259.7	120.6	120.6
21	0.983	9.351	73.2	60.51	107.7	87.7
22	110	110	30	30	72.5	72.5
23	1.2	1.2	30	30	16.9	16.9

Table 4 Overall Performance of (a) the Atmospheric Oxy-Fuel Power Cycle and (b) the Pressurized Oxy-Fuel Power Cycle

Performance Parameters	Unit	(a)	(b)
Thermal energy Input (HHV)	MW$_{th}$	874.6	874.6
(LHV)		839.1	839.1
Gross Power Output	MW$_e$	388.0	404.5
Net Power Output	MW$_e$	264.3	292.6
Gross Efficiency (HHV)	%	44.4	46.2
(LHV)		46.2	48.2
Net Efficiency (HHV)	%	30.2	33.5
(LHV)		31.5	34.9
Fuel Demand	kg/s	30	30
Steam Demand	kg/s	211.9	210
Oxygen Demand	kg/s	73.52	73.52
Flue Gas Flow Rate (Into the HRSG)	kg/s	1011.1	1004.7
Recirculation Ratio:	%	88.1	88
Combustor		(25.8)	(26.1)
HRSG		(62.3)	(61.9)
Flue Gas Flow Rate (Into the Purification and Compression Unit)	kg/s	107.7	87.7