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Exergetic comparison of different oxyfuel technologies

Isabel Guedea[1], Carlos Lupiañez[2], Luis M. Romeo[*3]

[1,2,3] CIRCE (Research Centre for Energy Resources and Consumption),
Universidad de Zaragoza, Mariano Esquillor, 15, 50018 Zaragoza, Spain.

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Abstract

Carbon Capture and Storage (CCS) systems have relevant energy consumption associated with the CO_2 capture process. It causes an energy efficiency reduction that diminishes the economic interests and increases the technical uncertainty of these systems. With the objective of improving the system performance and reducing thermodynamic inefficiencies, the exergy analysis has been traditionally applied as a guide for design process. This work presents and compares energy and exergy analysis of two CCS systems based on pressurized oxyfuel combustion, a pressurized fluidized bed combustion working under oxyfiring conditions and a chemical looping combustion using coal as fuel. The aim is to calculate CCS energy and exergy penalties, detecting irreversibilities and proposing items for optimization. It is demonstrated that opposed to energy penalty, the exergy losses do extend neither in the same quantity nor in the same equipment, leading to outstanding conclusions for system improvements. As it will be demonstrated, the exergy penalty of additional equipment for CO_2 capture does not cause relevant losses and these irreversibilities are concentrated in several systems that should be redesigned or analysed in detail to reduce the losses.

Keywords

CO_2 capture, exergy analysis, pressurized combustion, fluidized bed, chemical looping combustion

1. Introduction

One of the main obstacles for the deployment of Carbon Capture and Storage (CCS) systems is the important energy consumption associated with the CO_2 capture process. This causes an increment of the operational cost that, joined to the capital cost uncertainties, reduces the attractiveness to step forward the commercial stage. In spite of these transient drawbacks, Intergovernmental Panel on Climate Change (IPCC) and European Technology Platform for Zero Emission Fossil Fuel Power Plants (ZEP) consider CCS one of the options to reduce greenhouse gasses emissions in medium-long term [1-2].

To increase the system performance and reduce the inefficiencies the exergy analysis has been traditionally used as a guide. It has demonstrated good results in the synthesis of complex systems and efficiency improvements in energy applications. This work presents an exergy analysis of two CCS systems based on pressurized oxyfuel combustion. The aim is to reduce CCS energy penalties, detecting irreversibilities and proposing items for optimization.

Due to the novelty of CCS, there is a lack of literature related to the exergy analysis of different capture systems. A pioneering application of this kind of analysis applied to CO_2 capture systems is the use of the exergy of liquid natural gas to reduce the energy consumption of oxygen production in an oxyfuel combined cycle [3]. Also,

[*] Corresponding author: Luis M. Romeo (e-mail: luismi@unizar.es)
Tel.+34-976-761863; Fax: +34-976-732078.)

exergy analysis has been used in amine scrubbing applications to conclude that the exergy losses in stripper and reboiler were smaller than losses in the flasher and that the pressure drop over the absorber had a strong influence on exergy destruction, and finally, based on the results, they proposed items for optimization [4-5]. Oxyfuel and chemical looping power cycles have also deserved attention on the literature [6-19]. The exergy regeneration performances were investigated for the chemical recuperation with CO_2-NG reforming [6-8] in oxyfuel power cycles. Chemical looping combustion with gas turbine cycles [9], comparing different oxygen carriers [10], H2 production [11-12], and power cycles based on oxyfuel+GT [13-18] and Ca-looping [19] have also been studied.

Aiming the efficiency maximization, precombustion, gasification and high pressure combustion systems have an undoubtedly interest and represent an interesting option. Looking for reducing energetic requirements, high pressure combustion could decrease CO_2 conditioning and compression demands, having an important effect on net efficiency and cost reductions of CCS power plants. The possibility to incorporate a gas turbine or even to take advantage of the water-condensing flue gases energy recovery are other possible advantages of these options. In the literature it can be found two main possibilities to include a pressurized coal combustor [20-22]. Research carried out by MIT and ENEL [20] is focused in oxyfuel combustion using pressurized coal technology. No details are described about the combustor design, but they show an intensive work with steam cycle integration and flue gas pre-treatment. The conclusions were interesting values of net power plant efficiency with carbon capture of 34.8% based on Low Heating Value (LHV) and a gross efficiency of 47.6%. The work of KTH [21-22] proposes a hybrid combined cycle with pressurized fluidized bed combustion (PFBC) and CO_2 capture. They join a gas turbine as a topping cycle with a PFBC as a bottoming cycle plus the CO_2 capture system with a hot potassium carbonate process with a regeneration heat of 1.85 MJ/kg CO_2. Results are excellent and encouraging with power plant efficiencies of 43.9% with CO_2 capture. However, the complexity of the process increases sharply and cost figures, not included in the study conducted, should increase in parallel.

Recently, the pressurized chemical looping combustion with combined cycle has been proposed as a possible solution to increase thermal efficiency when coal is used as fuel in chemical looping combustion [23-25]. The use of solid fuels in chemical looping combustion has deserved attention by outstanding researchers in the field [26-30], although nowadays, the main interest is focused in the use of gaseous fuels [31-33]. For some coals, operating temperature should be limited to avoid ash melting in the reactor; the use of pressurized reactor should increase system efficiency and would enable higher fuel conversion efficiency [34]. Evidently, there is a lack of literature on pressurized chemical looping combustion of solid fuels [35], but the effect of pressure on several variables of CLC has been presented elsewhere [35-36].

The aim of this work is to simulate and propose system improvements of pressurized fluidized bed oxyfuel combustion (PFBC) through an exergy analysis, comparing energy and exergy efficiencies of different equipment. Firstly, an oxyfuel pressurized fluidized bed is described and simulated, and later a pressurized chemical looping combustion (CLC) for coal is proposed and modelled.

2. PFBC cycle

2.1. Description

The first option uses a pressurized fluidized bed working under oxyfiring conditions to produce a concentrated CO2 stream. This option has the advantage of the reduction of system size that could reduce the capital investment. For analysing and comparing this technology with pressurized CLC in literature, a 550 MW_{th} pressurized fluidized bed under oxy-firing conditions has been simulated. The software used for simulation is EES [37] that allows the solution of a set of algebraic equations, providing many built-in mathematical and thermophysical property functions useful for engineering calculations. The layout of this O_2/CO_2 PFBC system is schematically illustrated in Fig.1. Oxy-PFBC cycle is roughly divided into five main sections: the air separation unit (ASU), the pressurized fluidized

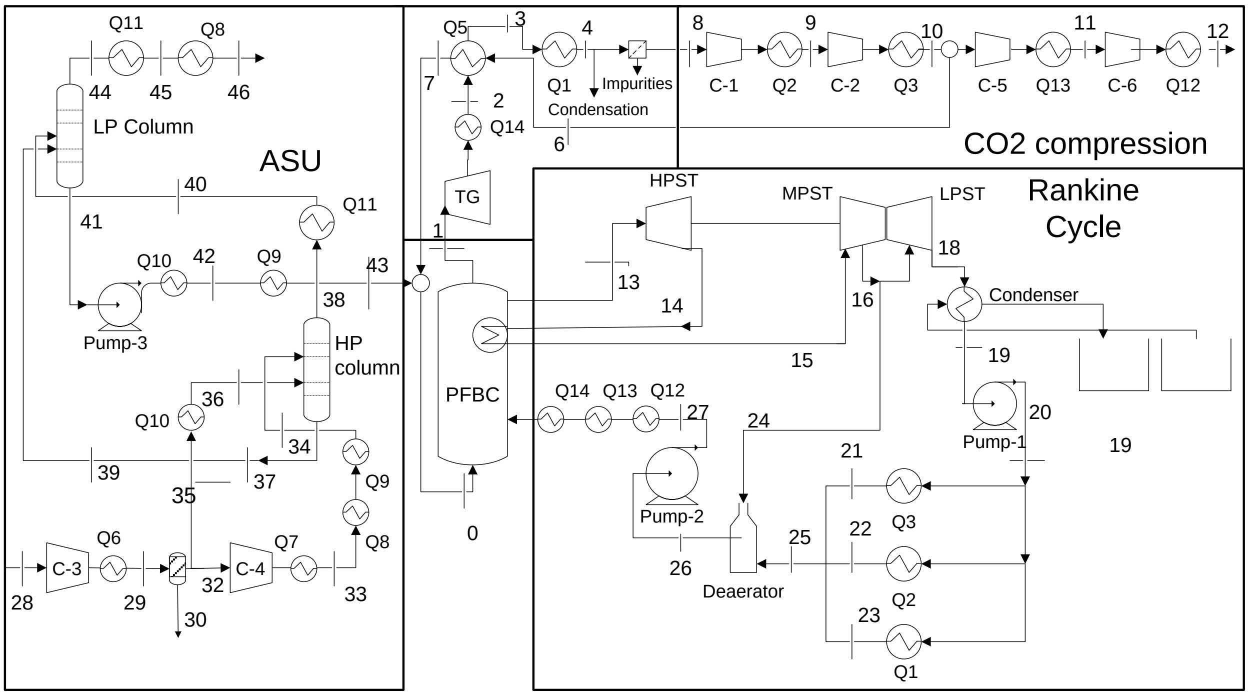

Fig. 1: PFBC cycle scheme.

bed combustor (PFBC), the CO_2 turbine, the Rankine cycle and the CO_2 compression system.

PFBC operates at 12 bar and between 800 and 850ºC. Modelling is based on a previous work [38] using data of a semi-commercial power plant for validation at air conditions. For this case, the main objective is the evaluation of maximum boiler capacity (live steam production), maintaining a suitable bed temperature profile. The model inputs presented are the fed fuel (550 MW_{th}), a reasonable fluidization velocity (1.1 m/s), the Ca/S ratio (1.8) and the particle size distribution inside the bed (mean diameter 700 µm). The simulation calculates the boiler heat transfer coefficients and, with several energy balances along the bed height, the steam production and bed temperature are obtained. In PFBC the largest influence on heat transfer is the contribution of particle convection. Literature shows that similar values for this variable are obtained when fluidization changes from air to O_2/CO_2 mixtures [39]. Owing to the facts that the fluidized bed temperature has been considered the same for both cases, air and oxy-firing conditions, and the signifier contribution of the particles in the heat transfer in fluidized bed boilers, it is reasonable to assume that literature correlations for heat transfer could be used for first estimations under oxy-firing conditions at PFBC. The bed-to-tube heat transfer coefficient results to be around 380 W/m^2K, with slight variations depending on the bed height, which operates at an average temperature of 820ºC.

Coal composition is shown in table 1. Due to the fact that the combustion time is much larger than the mixing time of particles, char combustion is modelled as being uniform along the bed height. Its contribution is evaluated as the oxidation of carbon, whereas the combustion of volatiles is calculated by subtracting the char energy content from the coal heating value. Volatile matter combustion is also considered to be uniform along the bed height.

For oxyfiring conditions, it is supposed a 35% oxygen concentration at PFBC inlet (0). This stream is mixed with a fraction of the exhaust gases that are recycled (57 %). The required oxygen is provided by an air separation unit (ASU) consisting on a double distillation column. The air flow entering the ASU (28) is calculated to obtain 10% of oxygen excess above stoichiometric. In order to obtain a high purity CO2 stream

Table 1. Coal composition.

Fuel Composition	
C	64.46 (% d.b.)
H	2.32 (% d.b.)
N	1.20 (% d.b.)
O	6.20 (% d.b.)
S	1.04 (% d.b.)
Moisture	2.05 (% d.b.)
Ashes	26.00 (% d.b.)
LHV	
24009 (kJ/kg)	

the pressure level at ASU has been selected as usual, 11 bar, to obtain an oxygen purity of 95%.The ASU layout reference [40] is modified to adequate the heat transfer to the stream temperatures of the plant and to optimize the intercooling compression system. Distillation columns operate at low temperature, between -167ºC and -180ºC in the high pressure column (11 bar) and from -182ºC to -202ºC in the low pressure column (5.1 bar). Pure and pressurized oxygen (43) is diluted with recirculated CO2 (7) to increase the fluidization velocity and it is injected in the Oxy-PFBC.

After the PFBC, flue gases (1) at 820ºC mainly composed by CO2 are expanded in a gas turbine and then cooled down in three heat exchangers for steam reheating, increasing the temperature of flue gas recirculation and low-pressure water preheating. After these heat exchangers, impurities as excess of oxygen or nitrogen from the ASU are eliminated. Minimum temperature difference in Q14 and Q5 is around 40 ºC and flue gases intermediate temperatures result in 390ºC and 235ºC. After the water condensation in Q1 (4), part of the flue gases at 60ºC are conducted to a 2-stage intercooled compression (8). When the fluidized bed pressure is achieved in the compression system, a fraction of the CO2-stream (6) is recirculated as explained above and then preheated (7) until 354ºC. Intercooling compression reduces the CO2 temperature at compressor inlet to 60ºC, leading a decrease of the power necessities. Waste energy from Q2 and Q3 is integrated as low pressure heaters in the steam cycle [41]. The two-stage compression produces a stream (12) with 92.2 % CO2 concentration at 120 bar and 60ºC.

Table 2. Simulation Assumptions.

Air properties	***1 bar, 15ºC***
CO_2 compressor pressure ratio (CLC)	***3.30***
CO_2 compressor pressure ratio (PFBC)	***3***
Compressors isoentropic efficiency	***0.91***
Air turbine isoentropic efficiency	***0.91***
HP Turbine isentropic efficiency	***0.92***
MP Turbine isentropic efficiency	***0.91***
LP Turbine isentropic efficiency	***0.86***
Pumps isentropic efficiency	***0.75***
Generator efficiency	***0.98***
Ancillaries consumption	***5 %***
Heat exchanger pressure drop (water-side)	***2 bar***
Evaporator pressure drop	***5 bar***

Live steam (13) is produced in the PFBC boiler at 180 bar and 550ºC. The main assumptions of the steam cycle are exposed in Table 2.Steam is expanded in three stage turbine at 48.5 bar with reheating in Q4 (14-15), 19.4 bar to deaerator (24) and 0.05 bar to condenser (18). Due to the integration of low-temperature waste energy in CO2 compression (Q1 to Q3) only one steam bleeding (24) is needed for deaerator.

3. Chemical Looping Cycle

3.1. Description

Chemical looping cycle (CLC) may be also considered as an oxyfuel technology, fuel reacts with oxygen transported by an oxygen carrier (usually a metal oxide) that previously has separated the oxygen from an air stream. Thus, one of the most important disadvantages of oxyfuel technologies, the elevated ASU power consumption is removed. For comparison purposes, the same thermal power and assumptions illustrated in Table 2 were selected for the CLC simulation. $NiO/NiAl_2O_4$ (40%/60% wt.) is used as oxygen carrier, due to its excellent performance at high temperatures [29,30]. A layout of the system is illustrated in Fig. 2 Air (1) is compressed and introduced at 10 bar in the CFB air reactor (2) with metal particles from fuel reactor (7) that arrives at 950ºC. Metal oxidizes in an exothermic reaction at 1050ºC producing an oxygen depleted air stream (3) that is introduced in a gas turbine producing power.

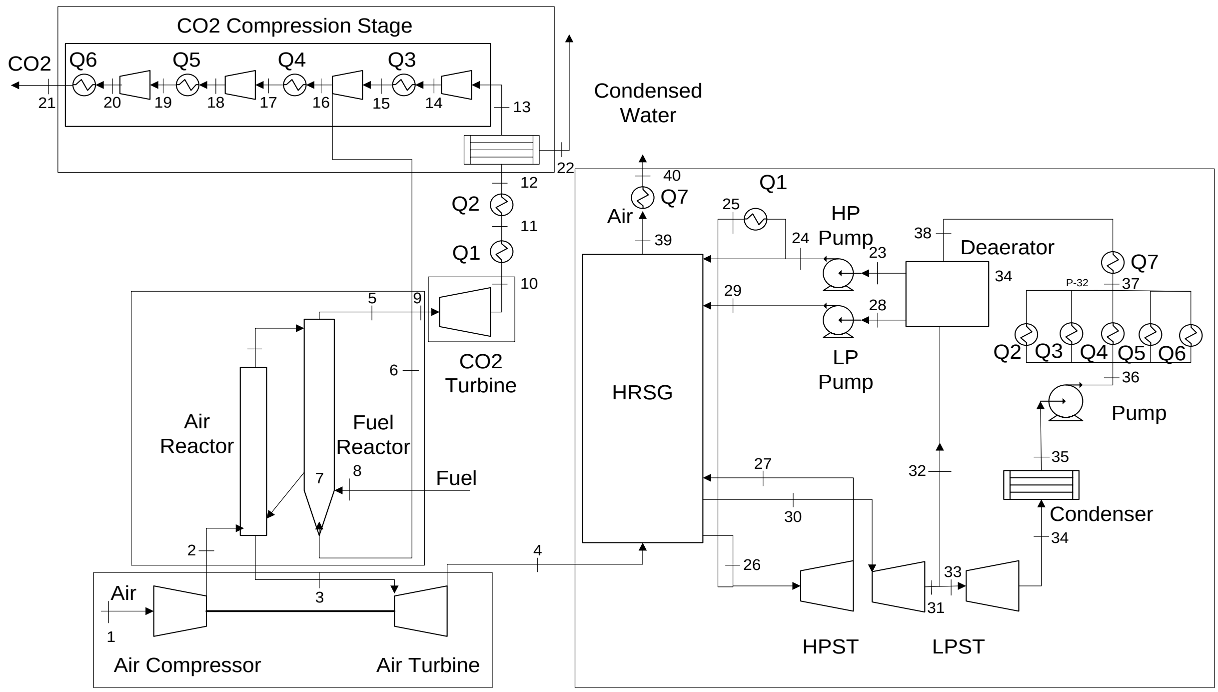

Fig. 2: Chemical Looping Combustion scheme.

As previously presented, table 1, a low sulphur coal has been selected. The presence of sulphurous compounds that can degrade the oxygen carrier particles, or form metallic sulphides or sulphates that reduce the melting point [31,32] is avoided. Coal is fed at the bottom of the reactor (8) with a CO_2/H_2O stream recycled from compression stage (6). Although it is not taken into account in simulation, it should be necessary a gasification stage [28] to improve the metal carrier reduction efficiency and eliminate the need for larger residence times. The reduction reaction is exothermic [42] and it takes place at 950ºC, which coincides with the flue gas stream temperature (5).

After a CO_2 turbine that produces power and reduces its temperature to 597ºC, a small HRSG produces HP steam (25) in Q1 (490ºC) that is led to steam turbine together with the main steam flow (26). In this heat recovery steam generator (HRSG) the minimum temperature difference is 20ºC and the simulation calculates the steam mass flow produced. Q2 takes advantage of the waste energy before water vapour condensation to preheat low pressure water. Pure CO_2 (13) is compressed up to 120 bar and 60ºC (21) through intercooling compression (Q3 to Q6) taking advantage of waste energy to preheat water in Rankine cycle.

The oxygen depleted air after the gas turbine (4) at 543ºC is used in the HRSG to produce steam. There are two steam pressure levels: 120 bar and 20 bar. Live (26) and reheat (30) temperature is 490ºC. There is only one steam bleeding for deaerator (32) at 3 bar, and steam is exhausted to condenser pressure 0.05 bar.

4. Results and Exergy Analysis

4.1. PFBC with CO_2 capture

Table 3 shows the main thermodynamics variables of the simulation of the PFBC with CO_2 capture. The net output power of the PFBC under oxy-firing conditions decreases from 247.5 MW_e without CO_2 capture, to 196.9 MW_e including CO_2 capture and compression system (table 4). In terms of net efficiency, the penalty is from 45.1% to 35.8%. There are 9.3 percentage points of difference compared with PFBC power plant without CO_2 capture. The efficiency reduction is mainly caused by ASU consumption, 7.5 percentage points; and the other significant contribution comes from the CO_2 compression system, 6.5 percentage points.

Table 3. Properties of oxy-PFBC streams.

Stream	Temperature (ºC)	Pressure (bar)	Mass flow (kg/s)
1	820.0	12.0	145.0
2	390.0	1.1	145.0
3	235.0	1.1	145.0
4	60.0	1.1	145.0
5	60.0	1.1	140.2
6	60.0	12.2	81.8
7	354.2	12.2	81.8
8	60.0	1.1	140.2
9	60.0	3.5	140.2
10	60.0	12.2	140.2
11	60.0	37.5	53.4
12	60.0	120.0	53.4
13	550.0	180.0	146.4
14	351.3	50.0	146.4
15	549.5	50.0	146.4
16	231.6	4.5	146.4
17	231.6	4.5	139.5
18	32.9	0.05	139.5
19	32.9	0.05	139.5
20	32.9	6.0	139.5
21	121.2	4.5	37.1
22	121.2	4.5	37.1
23	121.2	4.5	65.4
24	231.6	4.5	6.9
25	121.2	4.5	139.5
26	147.9	4.5	146.4
27	151.4	185.0	146.4
28	15.0	1.0	192.3
29	29.9	5.1	192.3
30	29.9	5.1	4.1
31	29.9	5.1	188.2
32	29.9	5.1	131.7
33	29.9	11.0	131.7
34	-164.1	11.0	131.7
35	29.9	5.1	56.45
36	-175.2	5.1	56.45
37	-180.1	3.6	109.1
38	-175.8	3.6	79.0
39	-180.1	3.6	109.1
40	-182.2	2.4	79.0
41	-202.1	1.5	44.1
42	-202.1	12.2	44.1
43	15.0	12.0	44.1
44	-191.2	1.5	144.1
45	-180.4	1.5	144.1
46	-3.2	1.5	144.1

The last quantity includes recirculation compression. There is an efficiency gain due to the absence of the air compression. As a consequence 37.6 MW_e and 6.8 efficiency points are saved in oxy-PFBC. From an energetic point of view, the main energy losses are located in the ASU. Oxygen production involves a large amount of energy due to the compressors used to obtain the appropriate pressure for the distillation columns operation, about 41.3 MW_e. It represents the 21.1% of the net power output. In literature, the energy needed for the oxygen production is between 7-10% of the total system input [18]. In this case the value calculated is 7.5% with a final consumption of 246.9 kWh/tO_2. This value is similar to other values found [18, 24, 25, 26], for the current purity of the O2 stream, although the outlet pressure is 12.2 bar, larger than in other CCS applications. Evidently the oxygen production consumption should be decreased and manufacturers attempt to achieve values around 180 kWh/tO_2. It would mean that the penalty could reduce from 7.5 to 5.5 efficiency points.

Flue gases at 820ºC and 12 bar are expanded in a CO_2 turbine after the pressurized bed. The power produced by this turbine is 55.7 MW_e. This quantity is bigger than the reduction in compression necessities when the CO_2 at 12 bar is directly led to compression and storage. After the CO_2 turbine, the stream is cooled down to 390ºC in a steam cycle high-pressure heater working as economizer.

Table 4. Analysis of performance of energy conversion in oxy-PFBC.

Energy input	***550.0(MW_{th})***
Steam turbine output	***222.7 (MW_e)***
CO_2 turbine output	***55.7 (MW_e)***
ASU consumption	***41.3(MW_e)***
CO_2 compression consumption	***36.1(MW_e)***
Net power output	***196.9(MW_e)***
Net overall efficiency	***35.81(%)***
Steam production	***146.1(kg/s)***
Intercooling waste heat	***36.0(MW_{th})***
Heat recovery after CO_2 turbine	***17.4 (MW_{th})***

CO_2 final conditions after four compression stages are 120 bar and 60ºC. The power consumption for compression is limited by intercooling between compressors. Intercooling compression reduces the power penalty and waste energy is integrated as low pressure heaters in the steam cycle.

The total compression necessities are 36.1 MW_e . On the other hand, there are 36 MW_{th} that could be integrated in the low pressure section of the steam cycle, avoiding the necessity of steam bleeding and increasing the power output in this turbine.

A live steam flow of 146.1 kg/s (180 bar/550ºC) drive the turbines producing 223.2 MW_e. Reheating section (50 bar/549ºC) is located inside the bed combustor. A small bleeding mass flow (7.2 kg/s) is necessary for deaerator. The low pressure section is preheated by the compression intercooling as it has been already described.

The flue gases moisture content after the combustion is around 4%. After CO_2-stream is dehydrated, the species concentrations in the recycled and capture stream are: CO_2 92.92 %; H_2O 0.71 %; O_2 2.8 %; SO_2 0.034 %; N_2 3.9 % (w).

The exergetic efficiency of the cycle is 45.35%. For this study the plant is divided in five subsystems: Air Separation Unit (ASU), pressurized boiler, CO_2 turbine, steam cycle and CO_2 compression system, fig.1.

The lowest exergy efficiency of the system is located in the ASU (table 5), where the efficiency is similar to previous references [43], around 28.5 %. It is caused mainly by oxygen separation and the heat exchangers irreversibility. The exergy input to this subsystem is electricity consumption and the output is a pressurized flow of oxygen.

Any other stream (nitrogen) is included as exergy losses, although it should be used elsewhere in the system. ASU irreversibility represents a 6% of the exergy fuel input, similar values are found in the literature [8]. It highlights again the importance of this equipment in order to achieve an efficient CCS system. Efforts should be done to optimize energetically and exergetically the ASU.

As expected, the highest irreversibility is associated with the combustion and heat transfer to steam cycle. Despite the low combustion temperature, usual values, around 63%, are obtained for this equipment. Exergy input is formed by the fuel, 522.5 MW; the oxygen produced in the ASU, 8.1 MW; recycled CO_2 stream, 9.4 MW; and the water preheated before being fed to the reactor, 27.4 MW. This exergy input is divided in 229.7 MW for steam cycle, 89 MW for flue gases directed to CO_2 turbine and 209.5 MW of exergy losses.

Exergy losses are much lower in CO_2 turbine, compression system and steam cycle. Gas turbine irreversibility is 12.1 MW, which represents 13.6 % of flue gas exergy.

There is a high exergy efficiency of CO_2 compression 85.3 %, due to the final CO_2 stream is considered a “product”. If this CO_2-stream is considered a loss, the efficiency drastically decreases to 46.3%. Regarding to the steam cycle, there is not a large scope for improvements as irreversibility amounts a 10% of total exergy input to this system.

As a conclusion, it should be noticed that the exergy penalty is not as important as energy penalty for the CO_2 capture equipment. In quantity, the main cause of irreversibility is fuel combustion with approximately the same percentage as air combustion.

CO_2 compression does not seriously affect energetic penalty and definitely the efforts should be developed to improve the oxygen production process as ASU exergy efficiency is significantly low. This point is essential to the success of the oxy-fuel technology.

In order to quantify the boiler pressure as the influential variable in oxy-PFBC, a sensitivity analysis has been performed. Without taking into account mechanical difficulties of operating under high pressure, calculations show (fig. 3) that an increase of the pressure in the reactor lead to important augmentation of energy and exergy efficiencies when the pressure is in the range 12-18 bar.

Above this pressure, both efficiencies have only a slight increase. The optimum point could be calculated if these results are complemented by an economic analysis. The main reasons are the power production in CO_2 turbine and the increase in intercooling waste energy that could be used in steam cycle.

Table 5. Analysis of performance of exergy conversion in oxy-PFBC.

PFBC	*Exergy*	*Efficiency*
Reactor		63 %
Exergy input	540.0 MW	
Irreversibility	209.5 MW	
Steam cycle		81.7 %
Exergy input	310.3 MW	
Irreversibility	56.9 MW	
CO_2 Compression		96.7%
Exergy input	21.7 MW	
Irreversibility	0.7 MW	
CO_2 Heat Exchangers		96.7%
Exergy input	87 MW	
Irreversibility	2.8 MW	
Gas Turbine		83.2 %
Exergy input	89.02 MW	
Irreversibility	14.9 MW	
ASU		28.5%
Exergy input	45.9 MW	
Irreversibility	32.8 MW	
PFBC Exergetic Efficiency		***45.4 %***

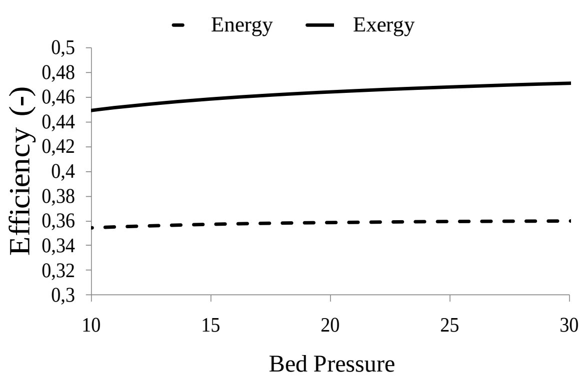

Fig. 3: Exergy and energy efficiencies vs pressure boiler.

4.2. Chemical Looping

In this case, table 6 show the thermodynamics variables, the net power output is 279.8 MW_e (table 7) for the same coal input as the example described before, 550 MW_{th}. Net system efficiency amounts 50.9%.

This value is lower than other reported in literature, 52.2% [44], but the use of gaseous fuel in CLC should results in higher efficiency. Nevertheless, it remains higher than other CO2 capture systems.

Table 6. Properties of CLC streams.

Stream	*Temperature (ºC)*	*Pressure (bar)*	*Mass flow (kg/s)*
1	15.0	1.0	655.2
2	316.6	10.0	655.2
3	1050.0	10.0	614.7
4	543.8	1.3	614.7
5	950.0	10.0	109.5
6	177.5	10.0	54.3
7	950.0	10.0	69.1
8	15.0	1.0	22.8
9	950.0	10.0	109.5
10	600.3	1.0	109.5
11	152.0	1.0	109.5
12	90.0	1.0	109.5
13	34.0	1.0	106.9
14	146.0	3.3	106.9
15	60.0	3.3	106.9
16	177.5	10.9	106.9
17	60	10.9	52.6
18	177.5	35.9	52.6
19	60.0	35.9	52.6
20	178.8	120.0	52.6
21	60.0	120.0	52.6
22	34.0	0.06	2.6
23	128.5	3.0	43.7
24	132.0	128.0	43.7
25	132.0	128.0	19.9
26	490.0	120.0	23.9
27	257.8	21.5	43.8
28	128.5	3.0	59.3
29	131.0	31.0	59.3
30	490.0	20.0	103.0
31	261.3	3.0	103.0
32	261.3	0.05	0.9
33	261.3	3.0	102.1
34	32.9	0.05	102.1
35	30.9	0.05	102.1
36	30.9	3.0	102.1
37	106.4	3.0	102.1
38	123.5	3.0	102.1
39	150.0	1.3	614.7
40	138.3	1.3	614.7

For this system the main penalty is associated to the CO_2 compression up to 120 bar, 29.9 MW_e and 5.4 efficiency points. The oxygen depleted air turbine after the air reactor (AR) produces 356.7 MW_e and 614.8 kg/s of flue gases at 543.8ºC. This power is partly used to drive the air compressor (199.8 MW_e). As a result, the net gas turbine power is 156.9 MW_e which represents the 48.1% of the gross power output.

Before being compressed, CO_2/H_2O stream (109.5 kg/s) is expanded in CO_2 turbine generating 47.6 MW_e, and at the same time, gas temperature is reduced to 600.3ºC. After this stage, gases are cooled down in a small HRSG recovering 54.9 MW_{th} and exhausting the gases at 152.0 ºC. Another heat exchanger reduces gas temperature while the condensate water is preheated. Finally, an additional heat exchanger works as condenser with ambient as cold reservoir in order to separate water vapour from the CO_2 stream before compression (2.6 kg/s).

HRSG after air turbine generates 83.1 kg/s of live steam at 490ºC and the small HRSG after the CO_2 turbine produces 19.9 kg/s in a parallel branch. As a result, steam turbine power output is 119.8 MW_e. Steam cycle is completed using the waste energy of intercooling compression that provides 25.8 MW_{th} that are used to preheat condensate water before entering in the deaerator, to increase condensate temperature from 30.9ºC to 106.4ºC and to minimize the steam bleeding between the two low-pressure steam turbine stages.

Table 7. Analysis of performance of energy conversion in CLC.

Energy input	***550.0 (MW_{th})***
Steam turbine output	***119.8 (MW_e)***
Gas turbine output	***156.9 (MW_e)***
CO_2 turbine output	***47.7 (MW_e)***
CO_2 compression consumption	***29.9 (MW_e)***
Net power output	***279.8 (MW_e)***
Net overall efficiency	***50.9 (%)***
Steam production	***103.0 (kg/s)***
Intercooling waste heat	***25.8 (MW_{th})***
Heat recovery steam generator	***314.5(MW_{th})***

In order to analyse the irreversibilities of the power plant, it has been divided in six subsystems: gas turbine, reactors, CO_2 turbine, CO_2 exchangers, rankine cycle and CO_2 compression to perform a CLC exergy analysis. Again, as it was expected, the main irreversibility is produced in the air and fuel reactors, 83.9 MW (table 8). This value implies an exergy efficiency of 88.4%. As before, it is needed to highlight the importance of the exergy streams considered as product. Generally, exergy associated with work is taken as a product, as well as heat exergy flows in the heat exchangers. In order to calculate this exergy efficiency, the CO_2/H_2O flow and the oxygen depleted air were considered as products that are going to be used in other equipment. This is not the unique way to define the exergy and another definition can take as a product the balance of exergy of the air current, reducing the exergy efficiency to 84.2%. In both cases, it is evident that chemical reactions are the main cause of exergy losses. They highlight the conclusion above presented, CO_2 capture does not imply as high exergy penalties as energetic penalties.

Table 8. Analysis of performance of exergy conversion in CLC.

Chemical Looping	***Exergy***	***Efficiency***
Gas turbine		87.8 %
Exergy input	178.8 MW	
Irreversibility	21.9 MW	
Reactors		88.4 %
Exergy input	722 MW	
Irreversibility	83.9 MW	
CO_2 Turbine		96.7 %
Exergy input	49.3 MW	
Irreversibility	1.7 MW	
CO_2 Heat Exchangers		66.3 %
Exergy input	48.7 MW	
Irreversibility	16.4 MW	
Compressing stage		96.6 %
Exergy input	21.2 MW	
Irreversibility	0.7 MW	
Steam cycle		70.2%
Exergy input	170.8 MW	
Irreversibility	50.9 MW	
CLC Exergetic Efficiency		***64.2 %***

higher, 96.65% due to the absence of a compressor.

CO_2 heat exchangers have relatively low exergy efficiency, 66.25%, due to the necessity of water condensation before the CO_2 compression. Then, the irreversibility of compression stages is low, 0.72 MW (96.59% exergy efficiency). In this block, there are several criterion to carry out the exergy analysis. Compressed CO_2 and heat obtained from the intercooling exchangers are considered the products of the block. When the CO_2 outlet is taken as a loss of the subsystem, the efficiency is reduced to 51.52%. Finally, if the CO_2 flow is an exergy product, but the heat exchanged is a loss, the efficiency decreases to 75.6%, which reveals the high importance of heat recovering after each compression stage. Finally, steam cycle has an exergy efficiency of 70.16% which is influenced by mechanical losses, steam condensation and heat loss in the condenser, and irreversibility in the HRSG.

To analyse the influence of pressure in CLC energy and exergy performance it has been assumed that the oxygen carrier and fuel combustion are not affected. Increasing oxygen partial pressure should promote oxygen transportation but the influence of higher CO_2 and O_2 partial pressure in CLC of coal have not been experimentally proved. Some researchers have reported that increasing pressure does not imply the improvement that was expected [36]. In our calculations, the pressure augmentation slightly increases the energy efficiency of the overall CLC, from 49.5% at 8 bar to over 54% at 20 bar. Higher pressure does not show any beneficial effect, fig. 4.

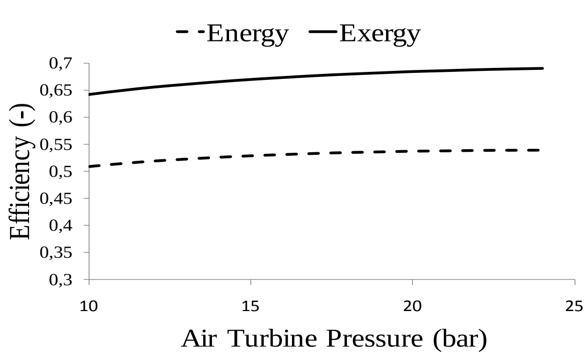

Fig. 4: The influence of inlet air turbine pressure on the energetic efficiency.

Similar tendency is observed for exergy efficiency in the range 8 bar to 20 bar, but near 20 bar, this value follows increasing whereas energetic efficiency seems to reach a steady value.

5. Conclusions

This work has compared the energy and exergy analysis of two oxyfuel pressurized combustion for CO_2 capture. An oxyfuel PFBC and a CLC using coal as fuel have been simulated and exergetically analysed to contrast energy and exergy penalties, to detect irreversibilities and to propose items for optimization. Although the energy penalty is high, exergy analysis show that CO_2 capture equipment does not have large irreversibility. In the analysed cases, fuel combustion remains to be the main cause of irreversibility with approximately the same percentage as in air combustion. For oxy-PFBC the exergy losses are 209.5 MW that represents a 37 %, and for CLC with coal the irreversibility is 83.9 MW that represents 11.6%.

CO_2 compression could affect the energetic penalty with reduction of 7 points in oxy-PFBC and 5.4 points in CLC. This difference is mainly produced by impurities of nitrogen and oxygen that have been considered in the fluidized bed, on the contrary, the combustion in CLC takes please by the stoichiometric oxygen without nitrogen impurities. Nevertheless, the exergy penalty is low in both cases, around 0.72 MW, it is due to the use of intercooling waste energy that minimize the losses and the CO_2 final stream that has been considered as a product. It is worthy to notice that the exergy efficiency of oxygen production is low, 28.5%, and is should be a candidate to propose items for optimization of system redesign. One of the advantages is CLC compared with oxy-PFBC is that the ASU energy and exergy penalty is avoided. Steam cycle losses are not affected by the CO_2 capture and values are similar to systems in the state-of-the-art power plants. Both systems show energy and exergy efficiencies augmentation when the operational pressure is increased. The influence is stronger for lower pressures (12-18 bar) both in oxy-PFBC and CLC and then the increment is smooth. In all cases CLC shows larger energetic and exergy efficiencies than oxy-PFBC.

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Biographies

Isabel Guedea (Ph.D. student Mechanical Engineer), is a Chemical Engineer Researcher that has been working on CO2 capture within the Thermal Division of Circe Foundation during the last four years. Her research is focused in design and built oxy-firing fluidized bed, instrumentation and automation of pilot plants and CO_2 capture cycles simulation.

Carlos Lupiáñez (Ph.D. student Mechanical Engineering), is a Mechanical Engineer Researcher in CIRCE during last five years. His research is focused in CO_2 capture systems. He took part in CIRCE oxy-firing fluidized bed design and nowadays he is studying SO_2 and NOx emission under oxy-firing conditions.

Luis M. Romeo (Ph.D. Mechanical Engineer), is a Project Manager in CIRCE and lecturer of “Thermal System Simulation” and “Zero Emissions Technologies, Carbon Capture and Storage” at the University of Zaragoza. His research is focused in energy integration and CO_2 capture systems (oxyfuel and CFB looping)