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Conceptual study of a high efficiency coal-fired power plant with CO_2 capture using a supercritical CO_2 Brayton cycle

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ABSTRACT

A concept of coal-fired power plant built around a supercritical CO_2 Brayton power cycle and 90% post-combustion CO_2 capture have been designed. The power cycle has been adapted to the coal-fired boiler thermal output, this boiler has been roughly designed in order to assess the power cycle pressure drop and its cost, an adapted CO_2 capture process has been designed and finally the overall heat integration of the power plant has been proposed. Due to the high complexity of such as plant, this paper does not intend to provide definitive evaluation of the concept but to explore its potential.

A coal power plant with CO_2 power cycle without carbon capture could achieve a net efficiency of 50% (LHV) with a maximal temperature and pressure of 620 °C and 300 bar, these performances has to be validated but the first results on pilot plant are encouraging. The CO_2 capture process use mono-ethanolamine as solvent and is equipped with vapor recompression systems in order to reduce the heat needed from the CO_2 cycle. It achieves around 2.2 GJ/t_{CO_2} of specific boiler duty with 145 kWh/t_{CO_2} of electrical auxiliary consumption including compression to 110 bar. The energetic evaluation of the overall power plant carried out highlights the promising potential of CO_2 supercritical cycle. A net power plant efficiency of 41.3% (LHV), with carbon capture and CO_2 compression to 110 bar, seem to be achievable with available or close-to-available equipment.

A technical-economic evaluation of the designed power plant has been performed. It shows a levelized cost of electricity reduction of 15%, and a cost of avoided CO_2 reduction of 45%, without transport and storage, compared to a reference supercritical coal-fired power plant equipped with standard carbon capture process.

1. Introduction

Parasitic load on power generation is the main concern about carbon capture technologies. On the one hand, this efficiency penalty leads to poor economic performance for these power plants; on the other hand, there is an increase in fuel consumption leading to supply problems and acceleration of resource depletion. Improvements of power plant efficiency due to higher steam condition are frequently foreseen for outbalancing the carbon capture penalty. But even with a temperature of 700 °C, a power plant design based on a steam cycle will have around 50 %-LHV theoretical efficiency [1]. Coupled with an, up-to-date, tightly integrated, MEA-based, post-combustion capture process, an overall efficiency of 42.5% could be achieved at best.

New power cycles based on helium or supercritical carbon dioxide are developed for future nuclear applications (fourth generation) [2–9]. These power cycles have very good efficiency. Helium cycle seems better adapted to high temperature reactor (800 °C) [7] whereas carbon dioxide seems adapted to low temperature reactor (450 °C) [7] but could be used in the range 400–750 °C [4]. Other high efficiency power cycle based on CO_2 are developed for gas turbine in oxy-combustion mode such as Graz [10] and Matiant cycle [11,12]. These cycles offer very high performance including CO_2 capture but are not adapted to coal-fired power plant: these cycles are based on an internal combustion in the cycle itself which is not possible with a solid fuel due to the lack of high temperature ash management and solid pressurization systems.

One main flaw of CO_2 power cycle is the cost and availability of large quantities of CO_2. With a CO_2 capture plant coupled with the power plant, a part of the captured CO_2 could be purified up to the boiler and turbine requirements (>99.99%) and feed into the power cycle [11].

This paper investigates the potential performance of a coal-fired power plant based on a supercritical CO_2 Brayton cycle and a post-

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combustion amine-based CO_2 capture process. An overall flow diagram of the plant is proposed and a simplified preliminary economic analysis is performed.

2. Adaptation of supercritical CO_2 Brayton cycle to coal-fired power plant

Supercritical CO_2 Brayton cycle has been extensively studied in literature since its conception [13] and especially these last years [2–7] as a suitable power cycle for sodium cooled nuclear reactor. Some preliminary design of a 750 MWe turbine has been carried out [5]. Main interests of such power cycle are their reduced volume, high safety and high efficiency even at low temperature source (450 °C). The high efficiency of this cycle is especially due to the strongly non ideal behavior of supercritical CO_2 near its critical point with respect to density and heat capacity. These turbines are in the developing stage and a few number have been tested at pilot scale (around 1 MWe) [2,4,14]. The main technological issues are the validation of turbine and compressor efficiencies estimation and the control-loop optimization of the power cycle due the sensitivity of the cycle performance to the operating parameters. Secondary concerns are linked to engineering developments: thrust load management, rotor cavity pressure control, advanced gas-seal operation, rotor windage [4].

2.1. Classical CO_2 cycle description

A Brayton supercritical CO_2 cycle is based on a regenerative Brayton cycle in within a fraction of CO_2 is pump at high pressure without prior cooling [2]. This arrangement maximizes the cycle performance and is named part flow cycle or recompression cycle. It is composed of (Fig. 1):

- A main heater where the CO_2 is heated to its maximal temperature.
- A turbine where the CO_2 enthalpy is converted to mechanical power, the minimal pressure being around 80 bars in order for the CO_2 to stay supercritical despite the pressure drop in the next heat exchangers.
- A main economizer where the cold, high pressure, CO_2 is pre-heated before the main heater cooling down the hot, low pressure, CO_2.
- An auxiliary economizer where a part of the cold, high pressure CO_2 is preheated before the main economizer cooling down the hot, low pressure, CO_2.
- A cooler where the low pressure CO_2 is cooled down to 30 °C.
- A main compressor where the most of the CO_2 is pumped to 200 bar.
- An auxiliary compressor where the rest of the CO_2 is pumped to 200 bar, the feed of auxiliary compressor is taken before the cooler and mixed back after the auxiliary economizer.

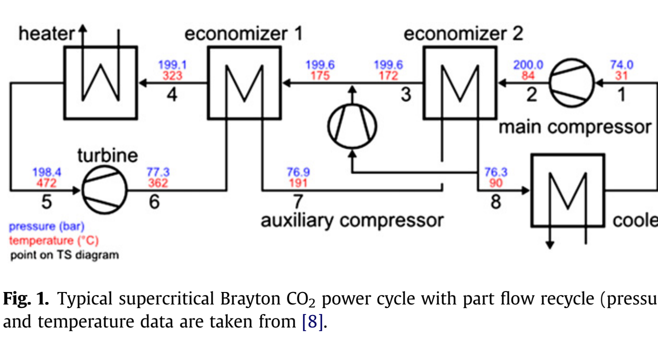

Fig. 1. Typical supercritical Brayton CO_2 power cycle with part flow recycle (pressure and temperature data are taken from [8].

The amount of CO_2 sent to the auxiliary compressor is adjusted in order to keep the pre-heated temperature constant. Numerous sensibility analysis on main operating parameter have been performed in literature [7,8], the most critical parameters are: the minimal pressure and temperature before the main compressor, the maximal pressure and temperature before the turbine and the split fraction sent to auxiliary compressor. Most supercritical CO_2 Brayton cycle have been designed for nuclear application with a heat source with constant temperature comprise between 450 and 600 °C with an overall cycle efficiency in the range of 39–45%. A high performance cycle without reheat and inlet temperature of 650 °C could achieve 47.1% overall efficiency [8].

The Fig. 2 shows the thermodynamic path of this power cycle, the point highlighted are the same than in Fig. 1. This figure has been calculated by the Lee-Kesler-Plocker equation of state for CO_2.

2.2. Cycle adaptation to coal-fired boiler

In coal-fired power plants, the temperature in the boiler is around 1400 °C; approximately half the total heat duty is delivered at this temperature through radiative heat transfer, the other half is the sensible heat of flue gas leaving the boiler. The CO_2 cycle heating must be redesign specifically for a coal-fired boiler coupled with a CO_2 capture process. Fig. 3 shows a simplified block flow diagram of the foreseen power plant.

The part-flow design [2,7,8] is not fully adapted: the heat needed for the cycle is delivered at high temperature (i.e. above 500 °C) consequently, with this configuration, a large part of the duty in the flue gas below 500 °C cannot be valorized in the CO_2 cycle or by air preheating. In order to circumvent this problem an additional CO_2 heater has been added in parallel to the primary air preheater and an additional economizer is foreseen to recover the CO_2 compression heat. The Fig. 4 describes the new flow arrangement, the thermodynamic diagram of the cycle is presented on Fig. 2 with the ′ points.

Moreover, in order to maximize efficiency and use the high heat transfer in the boiler, two reheats have been added. The maximal cycle temperature is 620 °C which is equal to the maximum temperature in most advanced supercritical steam/water power cycle (300 bar/600 °C/620 °C). Due to the high average temperature of cooling media (power cycle) in the boiler (around 590 °C) a high metal temperature (around 640 °C) is reached therefore a very high grade of steel (e.g. T91) must be used for flue gas corrosion protection. This grade of steel allows a high operating pressure of 300 bar (compared to a low pressure of 220 bar) at relatively few cost therefore such a high pressure has been chosen. A multiple reheat cycle has been chosen; the working fluid has high density (70–200 kg/m3) and allows compact heat exchanger arrangement for reheat. The multiple reheat cycle improve the overall cycle efficiency by 7–8 %pt. The reheat pressures have been chosen in order to have constant pressure ratio in each of the three turbines: 300/200/125 bar.

2.3. Cycle performance discussion

The influence of maximum pressure on the plant efficiency has been investigated and the part of the CO_2 sent to the auxiliary compressor has been optimized for each pressure for CO_2 power cycle without the CCS parts (i.e. no CO_2 diverted to boil the solvent and no CO_2 compression heat integration) (Fig. 5). The plant net LHV efficiency is in the range of 47.4% for 200 bar cycle to 50.8% for 350 bar cycle. The increase in efficiency comes from one main factor: as the pressure increase, the fraction of CO_2 recycled back through the auxiliary compressor decrease. This observation

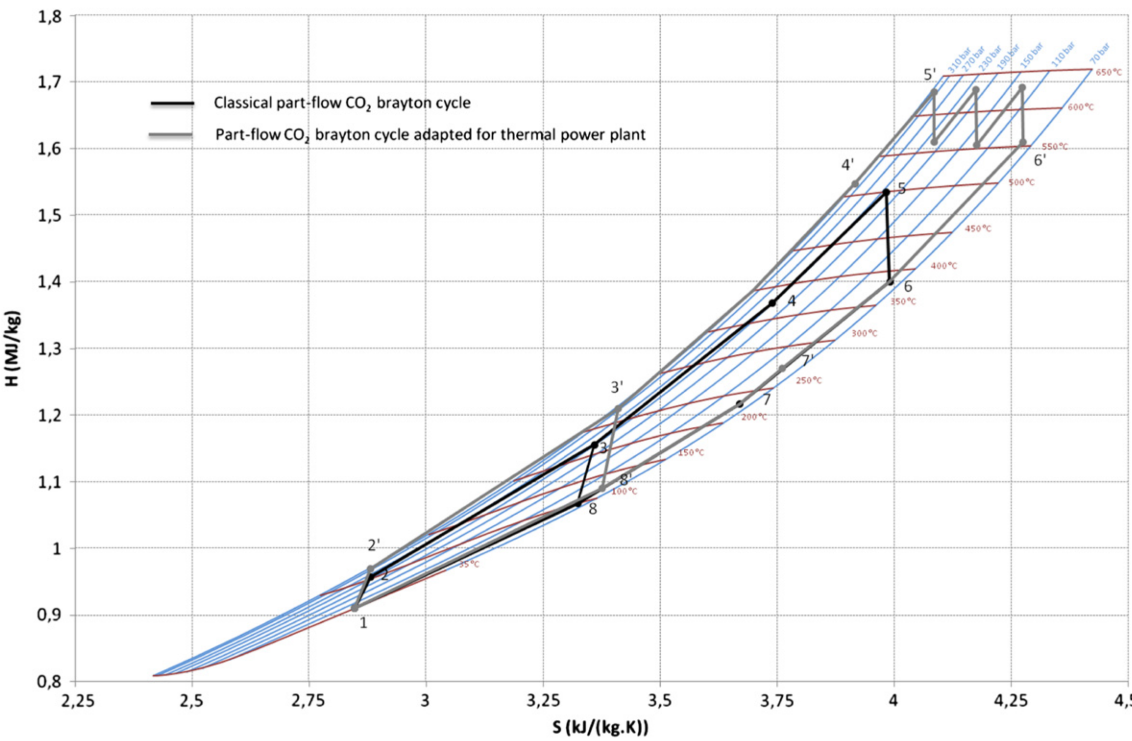

Fig. 2. H–S chart of supercritical CO_2 cycles.

illustrates the reduction of heat capacity of the CO_2 in the temperature range [90–200 °C] as the pressure increases.

CO_2 turbines isentropic efficiency of 93% has been chosen. This value is of the same order than the typical value of 92.9 [8] or 93.4 [2,5,15] % utilized in literature. It depends mostly of the number of turbine stages and varies from 90 (for a 2 staged turbine) to 94% (for a 10 staged turbine) [8].

CO_2 compressors isentropic efficiency of 90% has been chosen which is a typical value for a large heavy duty compressor. The efficiency range found in literature is from 89.1 to 95.5% for the main compressor and 87.5–94.8% for the auxiliary compressor [2,5,7,8,15]. The main compressor can be designed as a classical (centrifugal or axial) compressor or as a centrifugal pump due to its very high density of 600 kg/m^3. A conservative design study presented in Appendix 3 designs the main compressor as a 9 staged axial compressor and the auxiliary compressor as a 14 staged axial compressor (both are similar to gas turbine's compressors). Literature studies [8,15] design centrifugal compressor with 3–4 stages for the main compressor and 5 to 8 stages for the auxiliary compressor. Indeed, due to the high supercritical CO_2 density, the power cycle compressors can have very efficient and compact designs.

Pressure drop are a very important aspect in a Brayton cycle even if supercritical. A short cut design procedure of the boiler (Appendix 1) and main economizers heat exchangers (Appendix 2) has been performed in order to assess, quite accurately, the pressure drop along the CO_2 cycle. The calculated pressure drops are reported in Fig. 4.

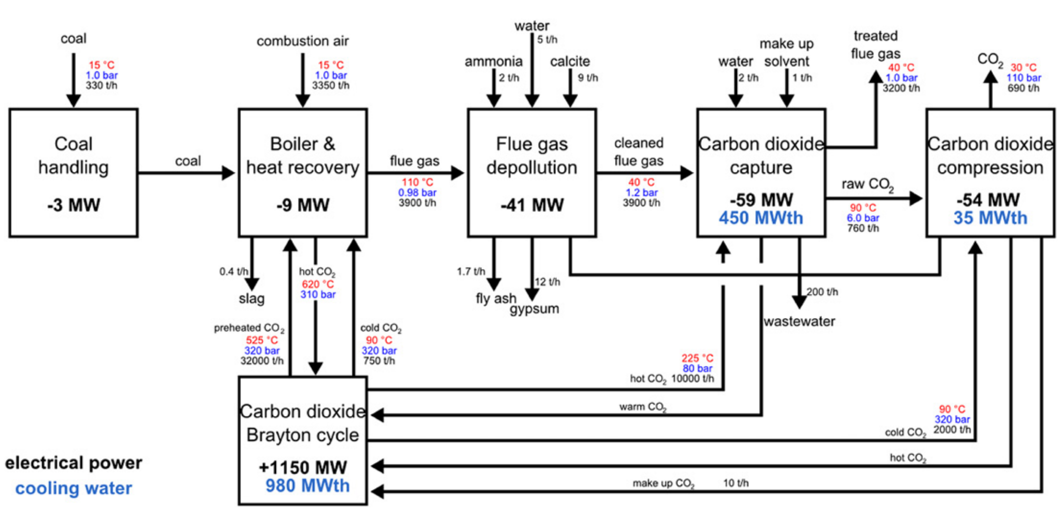

Fig. 3. Simplified block flow diagram of the power plant build around a CO_2 Brayton cycle.

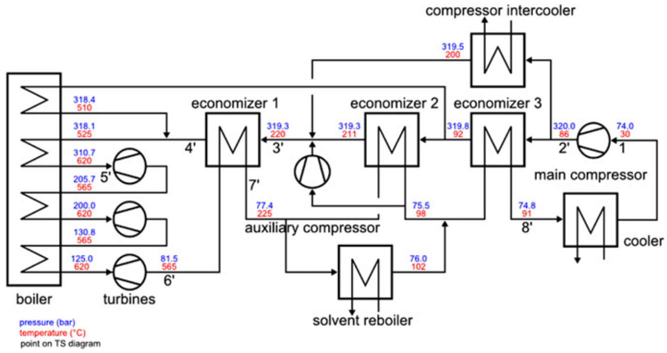

Fig. 4. Supercritical Brayton CO_2 power cycle adapted for coal-fired boiler with carbon capture.

These pressure drops are slightly higher than those reported in literature: 3.6 bar before turbine and 2.0 bar after [15], this is mostly due to the change in maximal cycle pressure (300 bar in this study and 200 bar in most previous publication [2,7,8,15] and the higher CO_2 temperature achieve at the first economizer cold outlet and the subsequent high heat exchanged between hot and cold flows). Pressure drop in boiler are high (more than 10 bar) but unavoidable in order to achieve the high heat transfer performance needed in the boiler furnace to cool down the furnace wall.

Pressure drop minimization would be one of the main sources of cycle efficiency improvement for instance, without pressure drop, net power plant efficiency would be 1.6 %pt better (for a 300 bar cycle).

3. CO_2 capture process

The absence of steam implies the absence of large quantities of heat at constant temperature which is the common mode of heating for CO_2 stripping; therefore the thermal integration of the CO_2 capture plant with the power plant must be adapted. There are two large source of sensible heat at the required temperature level: the flue gas between the SCR (Selective Catalytic Reduction of NO_x) and the ESP (Electrostatic Precipitator) and the CO_2 exiting the main economizer. The flue gas does not contain enough energy therefore a part of the CO_2 will be extracted at approximately 210 °C and 80 bar from the power cycle in order to boil the solvent. The

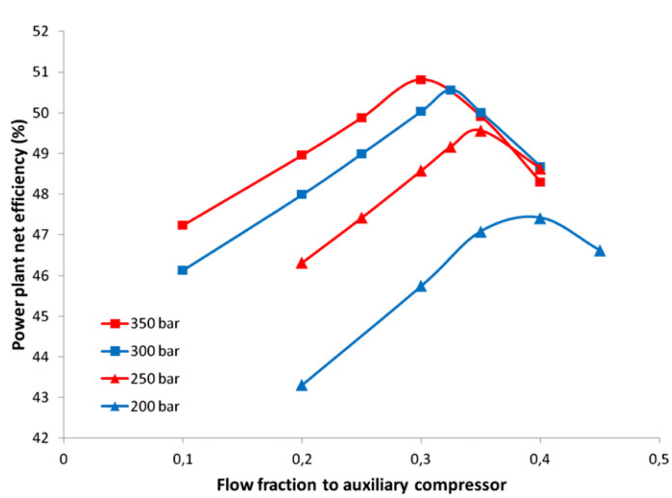

Fig. 5. Power plant efficiency with respect to the flow fraction going to recompressing compressor and turbine inlet pressure without the CO_2 capture parasitic load.

amount of CO_2 extracted is set in order to keep a 5 K minimal temperature pinch in the boiler. The conception of the capture process is based on anterior study on process modification [16].

3.1. Process description

The carbon capture process is based on a simple absorber (without intercooling or split flow) and a highly thermally integrated stripper. Both columns are equipped with 15 m structured packing bed. The CO_2 stripper is designed as follows:

- The stripper operates at 2 bar.
- A primary boiler fed with hot supercritical CO_2. This CO_2 is cooled down to 131 °C in order to boil the solvent.
- A primary economizer which pre-heat the rich solvent before its injection at the top of the stripper.
- A secondary boiler fed with the mix of gaseous CO_2 and H_2O exiting the stripper at its top and compressed to 6 bar in the overhead compressor (Stripper Overhead Compression: SOC). This stream is cooled down to 131 °C to boil the solvent, most of the water condenses.
- A knock-out drum which separate liquid water and CO_2 gaseous stream, the water is injected in the middle of the stripper. The pressure drop (from 6 to 2 bar) generates some steam which enhances the solvent regeneration (Stripper Overhead Vapor: SOV).
- A secondary economizer fed with the CO_2 gaseous stream exiting the knock out drum.
- A second knock out drum where the hot lean solvent is flashed under half the stripper pressure generating some steam (Lean solvent Vapor Compression: LVC). This steam is then compressed in the LVC compressor and injected back the bottom of the stripper.

Two solvents have been investigated: MEA at 30 %w and an MEA activated MDEA solvent at 45 %w MDEA and 5 %w MEA. The Fig. 6 show a simplified flow scheme of the process with some basic operational parameters and typical electrical and heat duties for MEA.

3.2. Process simulation

The process has been modeled with rate-based calculations and the electrolyte-NRTL thermodynamic model implemented in ASPEN-Plus. Two different stripper pressures have been investigated for MEA solvent in order to find the optimal setting: 1 and 2 bar at the stripper top. The overall plant net efficiency (LHV) has been calculated with respect to solvent CO_2 lean loading. The results of this parametric study are presented in Fig. 7. It appears the optimal lean loading is dependant of stripper pressure: 0.225 for 2 bar stripper and 0.325 for 1 bar stripper. High stripper pressure gives better overall efficiency: around 41.4% compared to 40.7% for atmospheric pressure. A higher pressure can possibly gives better efficiency but the boiler temperature will be so high (>130 °C) that the thermal degradation of solvent will be unmanageable.

The 15 m packing height in the stripper and the 5 K pinch in the economizer lead to a specific heat duty of 3.65 GJ/t_{CO_2} for a simple process with MEA solvent and 2.85 GJ/t_{CO_2} for activated MDEA solvent. Due to the high complexity of the process regeneration part direct analysis of boiler heat duty is not possible. Indeed, a part of the boiler duty is produced by compressed stripper overhead cooling and condensation. Moreover both hot fluxes exiting the boiler are cooling down further in order to preheat the solvent before its injection at the top of the stripper and the

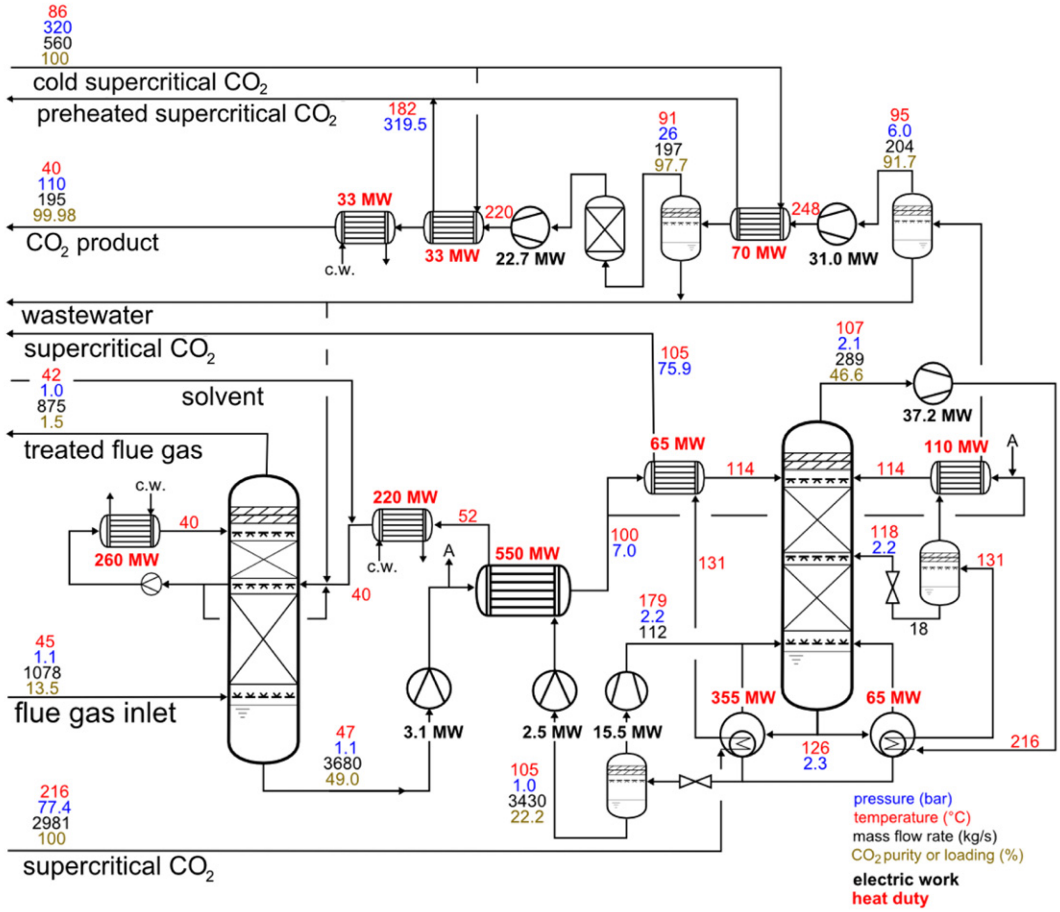

Fig. 6. Simplified flow scheme of the capture process for MEA case.

additional stripping steam provided by both the LVC and the SOV in the stripper reduce further its heat duty. The Table 1 summarizes the contributions of the four heat exchangers around the stripper (both boilers and both preheaters) and the equivalent heat provided by additional stripping steam for both processes and both solvents.

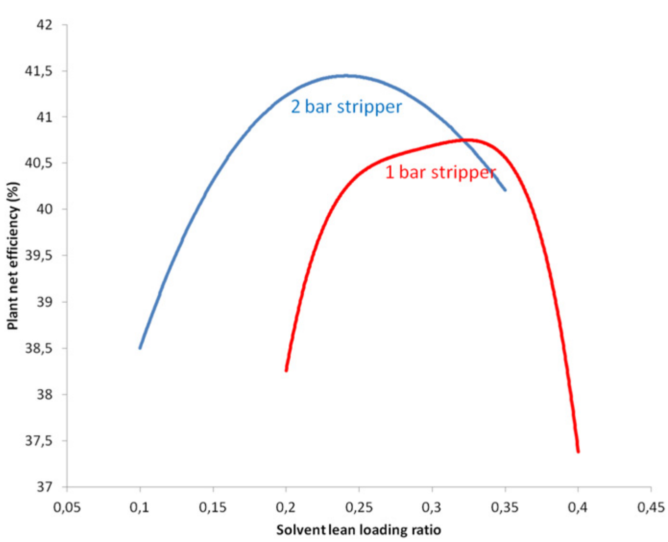

Fig. 7. Net power plant efficiency with respect to solvent CO_2 lean loading for 1 and 2 bar stripper operating pressure.

For the MEA case, the highly integrated process presented in this study reduces the heat from power cycle by 40% from 3.65 to 2.16 GJ/t_{CO_2} at the expense of an increase in auxiliary consumption of 45% from 100 to 145 kWh/t_{CO_2}. In other words, 45.4 electrical kWh/t_{CO_2} are converted into 414 thermal kWh/t_{CO_2}. The net efficiency increase for the whole power plant is presented in Fig. 11.

For the MDEA case, the integrated process reduces the heat needed from the power process by 50% from 2.83 to 1.42 GJ/t_{CO_2} with an increase in auxiliary consumption of 44% from 104 to 152 kWh/t_{CO_2}.

4. Integration of power plant and CO_2 capture process

The whole power plant has been modeled with ASPEN-Plus in order to assess its overall efficiency. Main assumptions are summarized in the following section.

4.1. Modeling assumption

4.1.1. Coal handling and combustion modeling assumptions

Properties of the coal are:

- Coal heat value: HHV (dry): 28891 kJ/kg et LHV (raw): 25739.4 kJ/kg,
- Immediate analysis: moisture: 7.06%, volatile matter: 32.28%, ash: 16.64%, fixed carbon: 51.07%,

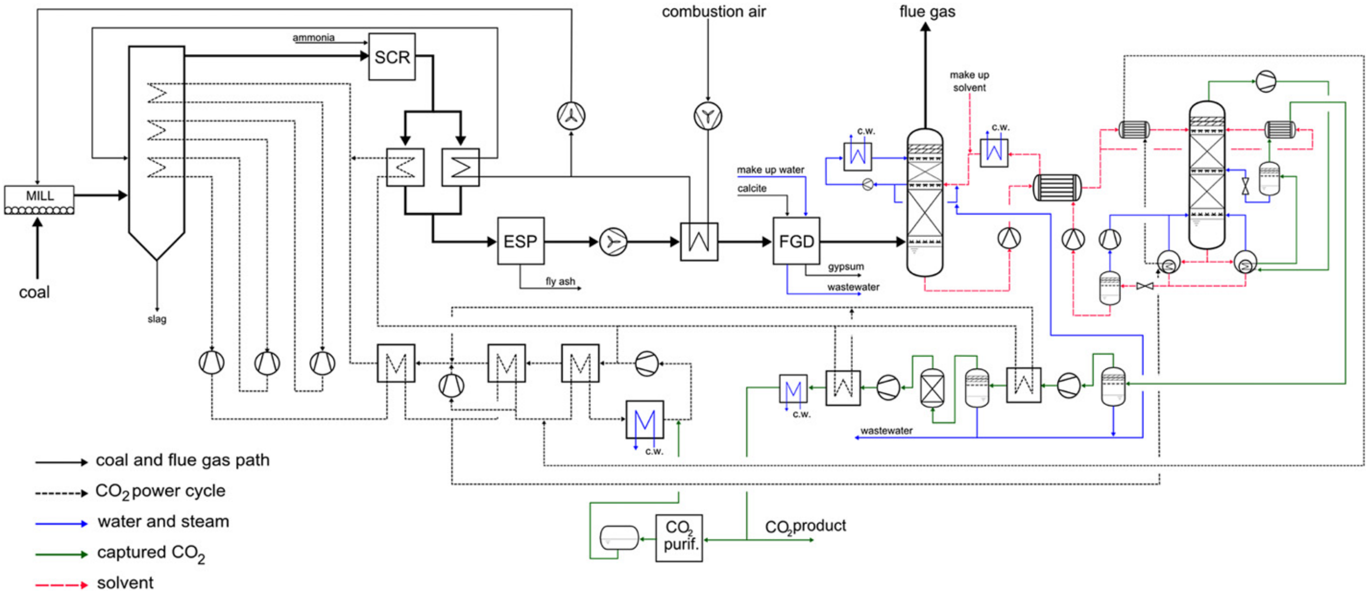

Fig. 8. Overall simplified flow scheme of the coal-fired power plant.

- Elementary analysis (dry): carbon: 69.35%, hydrogen: 4.45%, nitrogen: 1.04%, sulfur: 1.0%, oxygen: 7.51% and ash: 16.65%.

The coal is pulverized in a bowl and pillar mill, the specific work of the mill is 15 kWh/t_{coal}, including coal handling, which is an average value for such technology.

Coal combustion is modeled by an equilibrium reactor, its temperature is set to 1300 °C, and excess heat is extracted and represents the radiative transferable heat. An unburt carbon ratio of 1% is considered and 0.2% heat losses are supposed in the boiler. Coal flow rate is adjusted in order to keep the flue gas temperature at 110 °C at the outlet of the first air heater (just before the flue gas fan). 20% of the ashes are extracted at the outlet of the boiler and their heat is lost.

Inlet air is modeled as a mixture of nitrogen (79%) and oxygen (21%) at 15 °C and 1.03 bar, with a water mole fraction of 0.01%. The air flow rate is adjusted in order to keep the oxygen molar ratio at 4% in the flue gas, before air ingress. The air flow rate is preheated to 100 °C in a first air heater. 10% of this flow rate is send to the mill in order to transport the coal and is considered as the primary air, the complement is heated to 500 °C in a second air heater and delivered to the boiler. Air ingress of 2% of the total air flow rate is injected in the electrostatic precipitator. The secondary air forced draft fan compensates the pressure drop in the air heater and the wind box. This pressure drop is estimated at 40 mbar. The primary air forced draft fan compensates the pressure drop in the coal mill and the burners. This pressure drop is estimated at 120 mbar. The isentropic efficiency of fans is 65% with a mechanical efficiency of 99%.

Table 1
Main results of carbon capture process simulation.

Process		Standard	Fig. 7	Standard	Fig. 7
Solvent		MEA	MEA	a-MDEA	a-MDEA
Concentration	%Mass	30	30	45 + 5	45 + 5
Lean loading		0.225	0.225	0.10	0.10
Rich loading		0.49	0.49	0.24	0.24
Solvent flow rate	t/t_{CO_2}	17	17	36	36
Stripper pressure	Bar	2	2	2	2
Specific boiler duty ($S-CO_2$)	GJ/t_{CO_2}	3.45	1.82	2.66	1.18
Specific preheater duty ($S-CO_2$)	GJ/t_{CO_2}	0.20	0.34	0.17	0.24
Specific side boiler duty	GJ/t_{CO_2}	0.0	0.35	0.0	0.26
Specific side preheater duty	GJ/t_{CO_2}	0.0	0.58	0.0	0.31
Total specific heat duty	GJ/t_{CO_2}	3.65	3.09	2.83	2.00
Equivalent heat produced by LVC and SOV steam	GJ/t_{CO_2}	0.0	0.56	0.0	0.83
Specific heat duty provided by power cycle	GJ/t_{CO_2}	3.65	2.16	2.83	1.42
SOC auxiliary	kWh/t_{CO_2}	0.0	56.7	0.0	48.5
LVC auxiliary	kWh/t_{CO_2}	0.0	22.1	0.0	32.9
Pump auxiliary & Cooling system	kWh/t_{CO_2}	8.3	7.7	12.3	11.5
CO_2 products compression to 110 bar	kWh/t_{CO_2}	92.0	59.2	92.0	59.2
Total auxiliary	kWh/t_{CO_2}	100.3	145.7	104.3	152.0

4.1.2. CO_2 supercritical power cycle modeling assumptions

The maximum pressure and temperature of the CO_2 is assumed to be 620 °C and 300 bar, the same conditions as water in most advanced supercritical steam/water cycle. In order to maximize the cycle efficiency, the CO_2 is reheated at 620 °C twice: at 195 bar and 125 bar as detailed in Section 2.2.

Compared to the previously described supercritical CO_2 cycle, some process modifications are necessary to adapt it to a pulverized coal power plant with CO_2 capture:

- Part of the cold, high pressure, CO_2 is pre-heated against low pressure flue gas, between the SCR and the ESP equipment.
- Another part of the cold, high pressure, CO_2 is used to cool down the CO_2 in the compression train.
- A part of the hot, low pressure, CO_2 is used in order to boil the solvent used for CO_2 capture. The amount of CO_2 drawn off is adapted in order to keep the CO_2 capture ratio at 90%.

Seven convective and three radiative heat exchangers transfer the flue gas heat to the supercritical CO_2 cycle. An enthalpy loss of 3% is considered in each convective heat exchanger. The CO_2, preheated at 515 °C, then heated to 600 °C in the radiative heat exchanger and finally heat to 620 °C in two convective heat exchangers. After each turbine, the CO_2, which is at 565 °C, is reheated through a radiative and two convective heat exchangers at 620 °C. Design details are available in Appendix 1.

The compressors and turbines isentropic efficiencies have been set according to the literature as it is detailed in Section 2.3.

The CO_2 is cooled down at 30 °C. The needed cooling water flow rate is specified to avoid a cooling water exit temperature above 25 °C. The pressure drop in the cooling water system is 4 bar which is compensated by a pump with 65% isentropic efficiency.

4.1.3. Air control quality systems modeling assumptions

The selective catalytic NO_x reduction is not modeled since it does not have significant impact neither on flue gas composition nor on the energy balance of the plant. The electrostatic precipitator is modeled through an ideal separator which removes 100% of fly ashes. The flue gas desulphurization process is modeled with a multi-stage column feed by warm water (45 °C) and coupled with a separator. The column represents the flue gas cooling and humidification and the separator the sulfur removal. A pump raises the pressure of the warm water from 1 to 8 bar in order to take its work into account on the overall plant efficiency. The warm water flow rate is adjusted in order to keep a liquid to gas ratio of 12 L/m^3. The pump has an isentropic efficiency of 80% and a mechanical efficiency of 99%.

The induced draft fan compensates the pressure drop in the flue gas path with an isentropic efficiency of 65% and a mechanical efficiency of 99%. This pressure drop is estimated at 200 mbar including the CO_2 absorber column.

4.1.4. CO_2 capture and compression modeling assumptions

The CO_2 absorber is modeled with a five equilibrium stages column and the stripper with a fifteen equilibrium stages. The absorber is keep simple because the equilibrium stages assumptions does not allow for correct calculation of intercooling exchangers, multiple feed injections and so on.

The chosen solvents are an aqueous solution of MEA at 30% mass and an MEA activated MDEA at (45 + 5%). The optimal lean CO_2 loading is 0.225 for the first solvent and 0.10 for the second. Pumps and compressors have 90% isentropic efficiency and 99% mechanical efficiency.

The CO_2 is compressed in two stages (after SOC) from 6 bar to 110 bar. The CO_2 is intercooled at 90 °C after each compressor. The final CO_2 product is cooled further to 30 °C before transportation. Both compressors have a 90% isentropic efficiency.

4.2. Efficiency evaluation and pinch analysis

The complete power plant is represented on Fig. 8. Numerous process parameters need to be adjusted simultaneously in order to reach an appropriate operating point: supercritical CO_2 flow rate, fraction sent to the auxiliary compressor, fraction sent to the flue gas economizer, fraction sent to the CO_2 compression heat economizer, fraction sent to the CO_2 capture main boiler, pinches in each heat exchangers.

All these parameters have been set in order to achieve the targeted minimal pinch considered during the design procedure (Appendix 1 and 2):

- The pinch temperature in the three CO_2 power cycle economizers has been set to 5 K. The pinch temperature of all cooler has been set to 10 K.
- The total supercritical CO_2 flow rate of the power cycle (Q_{CO_2}) has been adjusted together with the fraction sent to the auxiliary compressor ($x_{aux.comp}$) in order to cool down the flue gas to 540 °C in the boiler and to maximize the overall plant efficiency. The pinch temperature in the boiler is located at the cold side of the first flue gas economizer, it has been set at 40 K.
- The fraction of supercritical CO_2 (cold, high pressure) sent to the second economizer ($x_{(sec.eco.)}$) has been set in order to achieve a 110 °C flue gas temperature at the exit of both secondary air preheater and second economizer and to obtain a 40 K pinch temperature at the hot side of both the second economizer and the secondary air preheater.
- The fraction of supercritical CO_2 (cold, high pressure) sent to the compression heat economizer ($x_{comp.eco}$) has been set to cool down the compressed CO_2 to 90 °C with a minimal pinch temperature of 10 K in these heat exchangers.
- The fraction of supercritical CO_2 (hot, low pressure) sent to the CO_2 stripper reboiler ($x_{reboiler}$) has been set in order to regenerate the solvent to the targeted lean loading (0.225 for MEA, 0.1 for activated MDEA) with a minimal pinch temperature of 5 K in the reboiler. The supercritical CO_2 drawn off from the power cycle is further cooled in a solvent preheater with a minimal pinch temperature of 5 K. It is then sent back to the power cycle between second and third economizer.

It could be noted effects of this setting interact with each other. For example, an increase of $x_{reboiler}$ reduces the total heat recoverable in power cycle economizers and reduces the final temperature of the preheated supercritical CO_2 before the entering the boiler. Therefore, Q_{CO_2} and $x_{aux.comp}$ must be tuned in order to maintain the maximal achievable power plant efficiency.

Fig. 9 shows the temperature–enthalpy diagram of the boiler part of the plant. As in a typical boiler, the heat transfer driving force is very high in the furnace part (horizontal flue gas line) and in the high temperature part of the flue gas path (>550 °C). A constant 40 K pinch is achieved in the low part of the flue gas path (<550 °C).

Fig. 10 shows the temperature–enthalpy diagram of the regenerative part of the power cycle, the CO_2 capture process and the balance of plant (miscellaneous cooling operation). The difference in calorific capacity of high pressure and low pressure CO_2 is highlighted (red and blue lines) as a 5 K pinch in the cold part of the heat exchanger leads to a 40 K pinch in its hot part. The average pinch in the CO_2 capture process is low (approximately 5 K in the solvent economizer) but reach up to 100 K in the reboiler due to the

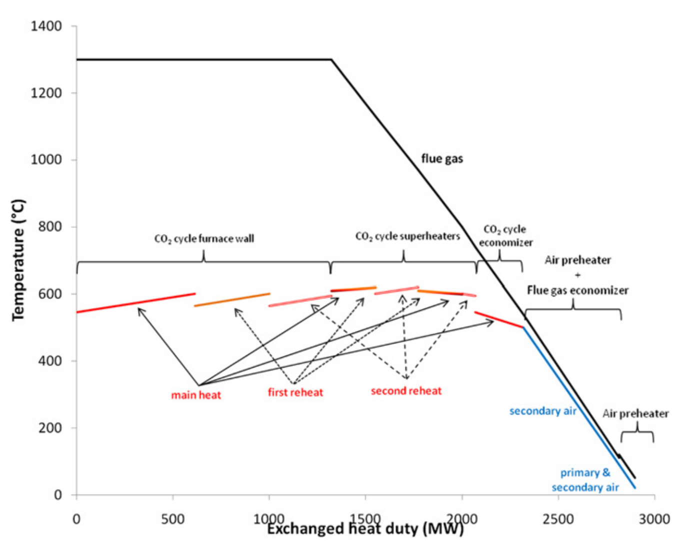

Fig. 9. Temperature–enthalpy diagram for the boiler: CO_2 power cycle and air preheating for the integrated MEA case.

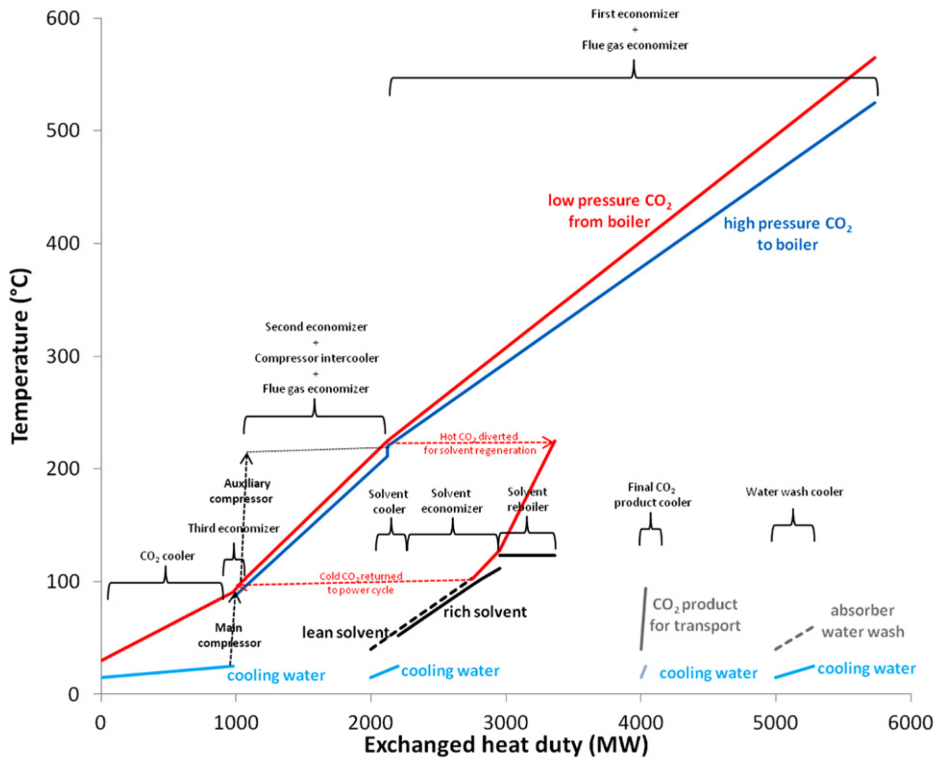

Fig. 10. Temperature–enthalpy diagram for the CO_2 power cycle regenerative part, the CO_2 capture process and the balance of plant.

use of sensible heat from the CO_2 drawing off from power cycle which represent approximately 30% of the total supercritical CO_2 flow rate.

With these setting procedures, the net plant performance could be evaluated. The main auxiliary consumption is presented in Appendix 4. Three different capture plant cases have been considered: a standard capture plant, an integrated capture plant without compression waste heat recovery and an integrated capture plant with compression waste heat recovery. The Fig. 11 summarizes the efficiencies of the different configurations investigated. The net LHV efficiency of the plant, for a 620 °C maximal cycle temperature, without CO_2 capture is 50.3%, this is 4.8 %pt higher than a state-of-the-art supercritical steam power plant.

A classical MEA based CO_2 capture process reduces this efficiency by 11.9 %pt down to 38.4%. This loss of efficiency is similar to the observed loss for steam plant [16]. The integrated process detailed in this study reduces the loss of efficiency to 9.7% leading to a 40.6% efficiency power plant. The further integration of the compression waste heat increases the plant efficiency to 41.4%.

The substitution of the MEA with activated MDEA solvent reduce the CO_2 capture penalty by 1.6 %pt for the standard process and by only 0.3 %pt for the integrated process, such a little efficiency gain is probably linked to the choice of carbon capture process (detailed in Section 3). Another process configuration could probably improve this design such as intercooling, a partial flash regeneration and a staged stripper feed. With the compression

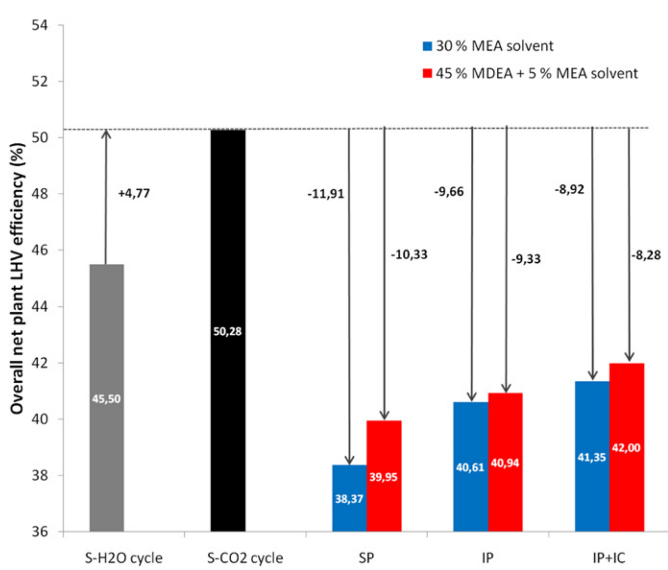

Fig. 11. Plant net efficiency for supercritical CO_2 cycle (620 °C maximal temperature) with CO_2 capture by the two studied solvents (MEA and activated MDEA) (SP: standard capture process, IP: integrated capture process, IC: integrated compression waste heat).

Table 2
Main economic assumption.

Project date	2010
Plant start up	2015
Currency	€ (2010)
Plant availability	7500 h/year
Plant lifetime	40 years
Project scope	Brown field
Levelization factor	8.4%
Financial cost	20%
License cost	5%
Contingencies	10% (20% for S–CO_2 cycle)
Fuel cost	2.75 €/GJ
MEA cost	4 €/kg
MEA consumption	1.5 kg/t_{CO_2}
Fixed O&M cost	2% CAPEX/year
Variable O&M cost	1 €/MWh (3 €/MWh for CCS)

Table 3
Techno-economic preliminary study results.

Power cycle		Steam	Steam	S–CO_2	a-S–CO_2
CO2 capture		–	MEA	MEA	MEA
LHV efficiency	%	45.5	36.5	41.4	44.5
CO_2 emission	t_{CO_2}/MWh	729	99	79	74
Capital cost	€/kW	2050	3000	2550	2370
LCOE	€/MWh	51.2	79.7	66.5	62.4
Capital part	€/MWh	22.9	34.0	28.8	26.9
Fuel part	€/MWh	21.8	27.1	23.8	22.0
Solvent part	€/MWh	–	4.3	4.2	4.2
O&M part	€/MWh	5.9	10.2	9.1	8.7
$LCACO_2$	€/t_{CO_2}	–	39.7	23.7	17.2

waste heat integration the plant efficiency reaches 42.0% with an activated MDEA solvent.

Finally, a maximal cycle temperature of 700 °C increases the calculated efficiency to 44.5% for an integrated case with MEA solvent. 700 °C is the targeted temperature for the so-called ultra-supercritical steam power plant planned for 2020–2030 (with a maximal pressure of 350 bar).

4.3. Economic evaluation

CO_2 turbines are very specific compared to steam turbine. A rough turbine design has been discussed with internal EDF experts, considering an internal diameter of 1 m and an exit Mach number of 0.15 (sonic velocity of 470 m/s at the outlet of high pressure turbine). Due to the very low density drop in the turbine (less than 30%), only one stage could be needed in each turbine. A blade height of 25 cm is needed for the high pressure stage (from 300 to 198 bar), 36 cm for the intermediate pressure stage (from 195 to 128 bar) and 50 cm for the low pressure stage (from 125 to 82 bar). Practically, in order to improve the turbine efficiency 5 to 8 stages could be needed [8].

The base cost of supercritical CO_2 turbine has been established based on the cost of the high pressure steam turbine corrected by the ratio of turbine volume. Due to large uncertainties on the cost of supercritical CO_2 turbines, a contingencies cost of 100% has been applied to the CO_2 turbines base costs. This leads to a cost of CO_2 turbine of approximately 40 €$_{2010}$/kW for the machine, of 1 GWe range, and 102 €$_{2010}$/kW including the civil works, piping, etc. Dostal et al. [8], have also analyzed the potential cost of such turbomachine and found a cost of 61 \$$_{2004}$/kW (i.e. 57€$_{2010}$/kW) for a 700 °C turbine of 1 GWe range but it is not clear if the secondary equipments, civil works and indirect costs are included.

Columns, drums and heat exchangers have been roughly sized with ASPEN-Plus model. The design of this equipment is reported in Appendix 5. The cost for columns, drums, tanks and heat exchangers have been calculated by the Pre-Estime method [17], the cost of ESP from Garret et al. [18], the cost of pumps and fans from Perry & Green [19] and the cost of coal mills, boilers, SCR, turbines and compressors from in house expertise. Table 2 summarizes the main economic assumptions concerning the project scope, plant lifetime and availability, the owner's cost calculation and the fuel and chemical cost. A brown field installation has been considered: the plant is installed on an industrial site already equipped with common utilities such as roads, rails, car parks, administrative building, high voltage connection, etc. No additional cost has been taken into account for additional health and safety systems concerning CO_2 cycle such as additional fan and specific building. These assumptions are in accordance to the EBTF guidelines [20].

Two reference cases have been added to CO_2 supercritical power cycle for comparison: one base case of 1000 MW (net electrical power) without carbon capture build around an advanced supercritical steam cycle (600/620 °C at 280 bar, 45.5% LHV efficiency), one base CCS case based on the same power plant with integrated MEA post-combustion capture and CO_2 compression train heat integration leading to 9 %pt of efficiency loss and a 800 MW (net electrical power) power plant (1000 MW minus the parasitic load). The base case economic evaluation gives a levelized cost of electricity (LCOE) of 51 €/MWh for the base case without capture and an LCOE of 80 €/MWh for CCS base case. The resulting levelized cost of avoided CO_2 ($LCACO_2$) is 40 €/t. These results are in good agreement with carbon capture international economic evaluation [21–23], they are summarized in Table 3.

Economic calculations have been performed for two different cases: one with a CO_2 supercritical cycle (620 °C, 300 bar, double reheat) with an MEA capture process and one with an advanced CO_2 supercritical cycle (700 °C, 400 bar, double reheat), again, with MEA capture process. The a-MDEA capture process has not been studied due to its not so different energetic performance and the lack of quantitative data on MDEA solvent degradation rate at industrial scales.

The 620 °C, MEA case shows a significant improvement on LCOE with a reduction of 15% and, consequently, a $LCACO_2$ reduction of 45% at 24 €/t_{CO_2}. These improvements are mainly due to the high efficiency of the plant with CCS: 41.3% and the lowest cost of CO_2 turbines (half the cost of a steam turbine) which compensate part of the cost of the capture plant. Indeed, a steam turbine installed cost is approximately 320 €/kW_{gross}, including alternator, and a supercritical CO_2 turbine installed cost 105 €/kW_{gross}; the net electrical cost is 350 €/kW_{net} for steam and 160 €/kW_{net} for CO_2 due to larger auxiliary consumption for the CO_2 case (35% instead of 5% for steam turbine). The detailed breakthrough of the capital cost is presented in Fig. 12. Consequently, this leads to a moderate increase on the specific capital cost compared to the case without carbon capture: only +26%.

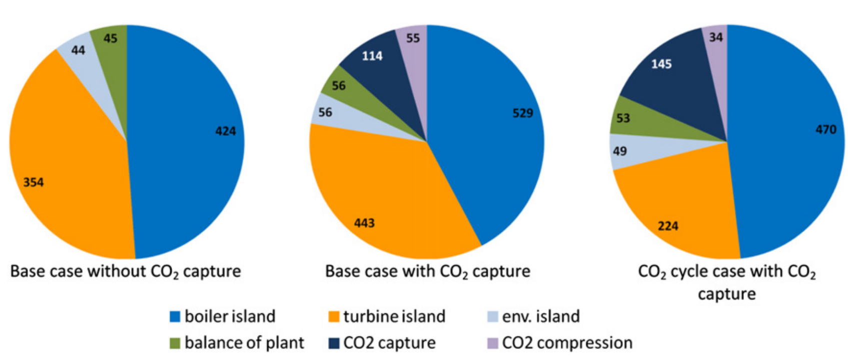

Fig. 12. cost breakthrough of main equipment in €/kW_{net} (excluding owner's cost, financial cost and contingencies) for the base case without CO_2 capture, the base case with CO_2 capture and the CO_2 cycle with CO_2 capture.

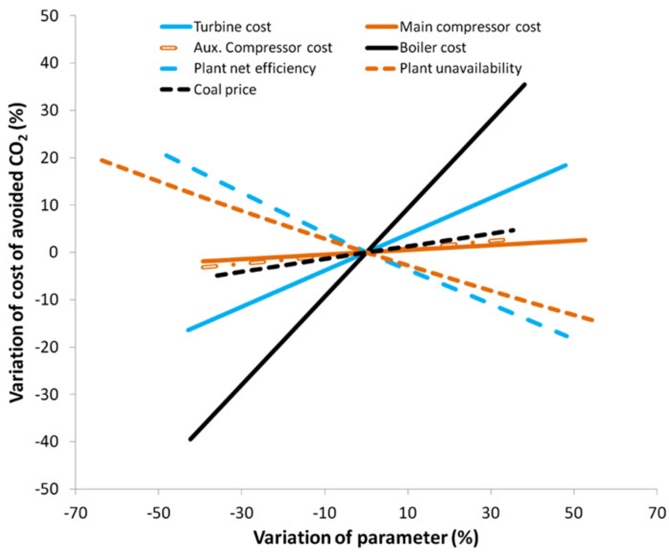

Fig. 13. Sensitivity analysis of main technical economic evaluation parameter on the cost of avoided CO_2.

A sensitivity analysis on the parameter with the most uncertainties has been carried out. These parameters are the supercritical CO_2 turbine and compressors cost, the supercritical boiler cost, the plant net efficiency, the plant unavailability and the coal price. The results are reported in the Fig. 13. It appears that the cost of boiler is the most important factor governing the cost of avoided CO_2 a variation of +/− 50% lead to a variation of the same magnitude on the cost of avoided CO_2. Following the boiler cost, the turbine cost and the plant net efficiency are the second most important. Plant availability is also important: a poor availability can hinders the good economic performance of the supercritical CO_2 Brayton power plant. Finally, cost of compressors and coal price have not significant impact on the cost of avoided CO_2.

5. Conclusion

A concept of coal-fired power plant build around a supercritical CO_2 Brayton power cycle and 90% post-combustion CO_2 capture have been designed. The power cycle has been adapted to the coal-fired boiler thermal output, this boiler has been roughly designed in order to assess the power cycle pressure drop and its cost, an adapted CO_2 capture process has been designed and finally the overall heat integration of the power plant has been proposed. Due to the high complexity of such a plant, this paper does not intend to provide definitive evaluation of the concept but to explore its potential. Indeed, the power plant concept proposed in this study allows a significant increase in net plant efficiency of 41.3% (LHV) (+5 %pt) with MEA-based carbon capture and CO_2 compression to 110 bar, a cost of electricity of 67 €/MWh and a reduced cost of avoided CO_2 of 24 €/t_{CO_2} (approximately 45%). A higher power cycle temperature (700 °C) yields a 44.5% net efficiency and a cost of avoided CO_2 of 17 €/t.

Numerous options can be investigated to further improve the plant efficiency, notably: the coupled optimization of the CO_2 capture flow scheme and solvent in order to benefit of the newly developed solvent, the integration of a fourth economizer in the CO_2 power cycle in order to reduce exergetic losses due to solvent boiling, the utilization of a global thermoeconomic optimization tool [24] in order to improve the overall plant economics or utilizing oxy-combustion CO_2 capture (no integration is needed between the CO_2 cycle and the carbon capture process and the CPU could provide the CO_2 purity required for the power cycle).

The CO_2 cycle make-up is a specific issue not tackled in this study, which raises two specific concerns:

- The CO_2 must be pure enough for the power cycle, a dedicated purification device could be necessary. This additional cost has not been considered. The rectification of almost pure CO_2 (99.98%) to pure CO_2 (>99.99%) can be performed at 60 bar in order to have CO_2 in liquid phase. A this pressure, the CO_2 boil at 22 °C, the heat needed for rectification can be provided by power plant waste heat. The pure CO_2 is pumped back to the main compressor inlet at 80 bar. The overall energy consumption for the CO_2 rectification is approximately 2 kWh/t_{CO_2} and has not significant impact on the plant efficiency.
- The CO_2 leak must be tightly controlled, the cycle flow rate is around 30,000 t/h and the captured CO_2 flow rate around 700 t/h consequently the turbine leakage need to be around 0.1% in order to keep an acceptable CO_2 capture rate. This leakage range is achievable with a combination of labyrinth, pressure and abradable seals.

Nevertheless, the first evaluation carried out in this paper highlights the promising potential of CO_2 supercritical cycle and the need of pilot plant validation of supercritical CO_2 Brayton cycle performance and CO_2 capture plant integration feasibility in order to consolidate the results.

Appendix 1. Boiler design short cut method

A1.1. Short cut methodology

A large number of parameter have to be taken into account for a boiler design: coal properties, plant load, type of power cycle, etc. This appendix describes a short cut method for tower (or one pass) boiler design in four steps: combustion calculation, furnace sizing, furnace wall sizing and suspended heat exchangers sizing. It is based on the open literature [25–29]. A boiler design is an iterative procedure, most of the design parameter must be adjusted to maximize heat transfer between flue gas and working fluid and reduce the volume and mass of the boiler and the grade of used steel in each heat exchange area: water wall, economizer, superheater and reheater. The short cut method presented in this appendix can be used in order to produce feasibility analysis of new boiler design with respect to main design parameters. This short cut could be improved in numerous ways such as zone model for radiative heat transfer, detailed modeling of the flue gas temperature decrease in the suspended heat exchangers, full and part load heat transfer performance calculation, etc.

A1.1.1. Combustion calculation

In this short cut method, combustion is treated as an instantaneous and near complete reaction. The fuel lower heating value is released in the furnace producing high temperature flue gases and radiative emission. Main hypothesis for this step are the unburnt fuel percentage (x_{unburt}, 0% for gas, 0.3% for heavy oil, 1% for coal) and the flue gas temperature exiting the furnace $T_{furnace\ exit}$ (between 1150 and 1350 °C, preferably around 1250 °C for large utility boiler [26]). Thus, a simplified heat balance on the furnace can be formulated:

$$H_{air} + H_{fuel} + H_{combustion} = H_{flue\ gasses} + H_{wall} + H_{loss}$$

with H_{air} the enthalpy coming with the preheated combustion air, H_{fuel}, the enthalpy coming with the preheated fuel, $H_{flue\ gasses}$, the enthalpy leaving the furnace with the hot flue gases, H_{wall}, the heat exchanged with the water wall, H_{loss}, the heat loss in the boiler (around 0.2% for a large utility boiler) and $H_{combustion}$, the reaction enthalpy of combustion which can be expressed as :

$$H_{combustion} = \dot{m}_{fuel}(1 - x_{unburt})\mathrm{LHV}_{fuel}$$

The coal composition (proximate and ultimate analysis) allows the calculation of flue gases compositions through a material balance. The overall quantity of air must be adjusted in order to have a 20% air excess (or 4 %vol. of O_2 in flue gases). With previous hypothesis, flue gases flow rate and heat transferred to the water wall could be calculated.

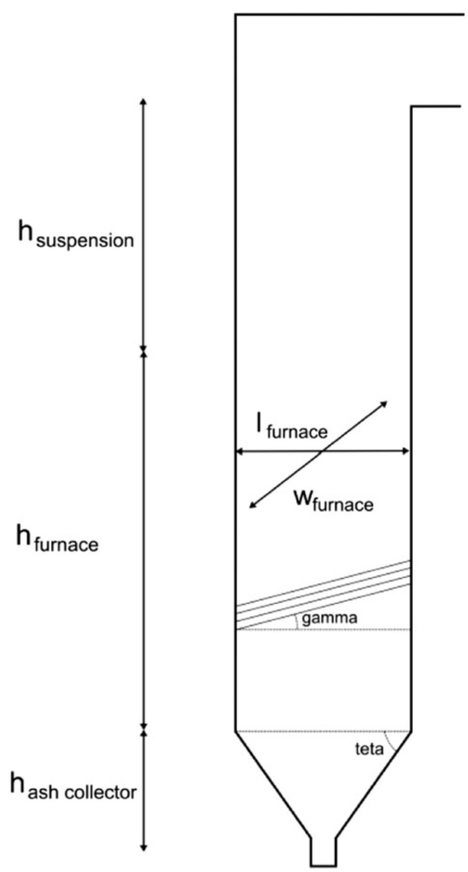

Fig. 14. Simplified geometry of a tower boiler.

A1.1.2. Furnace sizing

Two main characteristics govern the sizing of the furnace: the combustion residence time t_r and the flue gases furnace exit velocity. The furnace volume $V_{furnace}$ can be calculated from the following equation:

$$V_{furnace} = t_r Q_{flue\ \ gases}\left(T_{furnace}\right)$$

with $Q_{flue\ \ gases}(T_{furnace})$ the flue gases volume flow rate at the average furnace temperature which could be approximated at 1500 °C. Due to the high temperature and low pressure in an utility boiler furnace, flue gas behavior can be approximate as an ideal gas behavior. The typical combustion residence time is 0.2–0.5 s for gas and 1–2 s for coal. This residence time musty include the residence time of the disengagement zone above the fireball. The furnace section area $S_{furnace}$ can be calculated from the following equation:

$$S_{furnace} = \frac{Q_{flue\ \ gases}\left(T_{furnace\ \ exit}\right)}{u_{flue\ \ gases}\left(T_{furnace\ \ exit}\right)}$$

with $T_{furnace\ exit}$, the temperature of the flue gas exiting the furnace (defined in step 1) and $u_{flue\ gases}$, the maximal allowable flue gas velocity which depends on the solid content of the flue gas and the nature of these solids. Typically, a value of 30 m/s for gas combustion and 10–15 m/s for coal combustion can be chosen depending of the ash content. An overdesign of 10–20% is often considered for both volume and section of furnace.

With the furnace volume and section, its height $h_{furnace}$, length $l_{furnace}$ and width $w_{furnace}$ can be calculated (Fig. 14). For a tangential fired boiler the length equals the width and for a frontal fired boiler, the width depends of the deployed flame length and therefore of the burner characteristic (fuel and air flow rate, swirl, air staging). Knowing the width of the furnace, the ash collector height can be calculated considering a slope θ of approximately 60° (Fig. 14).

A1.1.3. Furnace wall sizing

Approximately half of the total heat transferred to the working fluid is transferred to furnace wall mostly through radiative heat transfer. In evaporative power cycle, ebullition is performed in the furnace wall tubes whereas in supercritical power cycle the key design parameter is the temperature of the furnace wall metal which dictates the choice of metal for the tube and the cost of the boiler. Nowadays most of the furnace walls tubes are made in T22 steel for evaporative and most supercritical power cycle and higher grade steel for new advanced supercritical power cycle such as T91 steel. With good heat transfer characteristic, the tube temperature is approximately 50 °C higher than working fluid temperature.

There are two main parameters to consider for furnace wall sizing: the tube inside diameter $d_{t,i}$ and the angle γ between the tube and horizontal (Fig. 14). Vertical arrangement ($\gamma = 90°$) requires a large number of tubes whereas typical spiral arrangement ($10° < \gamma < 30°$) reduces significantly the number of tube allowing higher mass flux in each tube and therefore better heat transfer. Both $d_{t,i}$ and γ must be chosen in order to evacuate the heat transferred to the wall by the furnace in the working fluid (H_{wall}), calculated in step 1, with the minimal mass of metal and the minimal pressure drop:

$$Q_{wall} = US^{eff}_{water\ \ wall}\Delta Tml$$

The total furnace wall area of the furnace can be deduced from the furnace design:

$$S_{water\ \ wall} = 2h_{furnace}\left(l_{furnace} + w_{furnace}\right) + w_{furnace}\left(\frac{l_{furnace}}{\cos\theta} + \frac{w_{furnace}\tan\theta}{2}\right)$$

The total effective area is: $S^{eff}_{water\ \ wall} = \pi/2S_{water\ \ wall}$ Convective heat transfer coefficient outside and inside the tube [29]:

$$Nu = 0.023Re^{0.8}Pr^{0.4}$$

Radiative heat transfer coefficient from the furnace [27]:

$$h^r_f = \frac{\sigma E_w}{\Delta Tml}\left(E_f T^4_f - A_f T^4_w\right)$$

with σ, the boltzman constant, E_w the emissivity of the wall assumed equal to 0.8, E_f, the emissivity of flue gas also assumed equal to 0.8 and A_f, the absorptivity of flue gas : ($A_f = 1 + E_f/2$) [26,27].

$$\frac{1}{U} = \frac{1}{h_f^r + h_f^c} + \frac{1}{h_{\text{ash}}} + \frac{2d_{t,e}e_t}{\lambda_m(d_{t,s}+d_{t,i})} + \frac{1}{h_{\text{dep}}} + \frac{1}{h_t}$$

with h_f^r and h_f^c, respectively, the radiative and convective flue gas heat transfer coefficient, h_t, the water film heat transfer coefficient, h_{ash}, the ash deposit heat transfer coefficient (around 300 $W/m^2/K$ for a 5 mm ash deposit), h_{dep}, the water side deposit heat transfer coefficient (around 10,000 $W/m^2/K$ for demineralized water),$d_{t,e}$ and $d_{t,i}$, respectively the tube outside and inside diameter and e_t, the tube thickness calculated as follow:

$$e_t = \frac{d_{t,e}P}{2\sigma_m + 0.5P}$$

The tube side pressure drop is calculated with the semi empirical Colebrook formula [29]:

$$\Delta P = 8f\left(\frac{l}{d_{t,i}}\right)\frac{\rho u^2}{2}, \frac{1}{\sqrt{f}} = -2\log\left(\frac{r_t}{3,7d_{t,i}} + \frac{2.51}{Re\sqrt{f}}\right)$$

The furnace wall above the furnace is always constituted of vertical tube, the overall heat transfer in these tubes is, most of the time, not really important. Their role is to ensure adiabatic behavior of the boiler. Nevertheless, if needed, a detailed calculation of the heat transferred could be performed with the previously described equations coupled with heat transfer calculation of the suspended heat exchangers (step 4).

A1.1.4. Suspended heat exchanger surface sizing

The role of suspended heat exchangers is to superheat, or reheat, the working fluid in the superheater and reheater and to reduce the flue gas temperature in the economizer. The number of suspended heat exchanger varies with the working fluid, the number of reheat and the cycle maximal temperature. As a rule of thumb, in order to ensure correct operability of the boiler, a couple of superheater is needed each time the working fluid temperature is raised at the maximum cycle temperature. This allows an intercooling between these superheaters and allows precise control over the final working fluid temperature. The main parameters governing the suspended heat exchanger sizing are the heat transferred, the tube diameter and the configuration of the heat exchanger. Each of these heat exchangers are constituted of a number N_e of parallel elements, each of these elements are constituted of N_t tubes of length l_t arranged in N_p parallel vertical passes.

The first step of the design procedure is to determine a heat exchangers arrangement and to perform a detailed heat balance on each of these heat exchangers in order to calculate the amount of heat transferred Q_{SHX} and the logarithm mean temperature ΔTml_{SHX}. The heat exchanger suspended in the flue gas à temperature higher than 900 °C have concurrent flow pattern whereas those suspended in colder flue gas have countercurrent flow pattern. The heat transferred can be expressed as follow:

$$Q_{\text{SHX}} = U_{\text{SHX}}S_{\text{SHX}}\Delta Tml_{\text{SHX}}$$

U_{SHX} can be calculated with equations presented in step 3 and S_{SHX} can be calculated as follow:

$$S_{SHX} = \pi d_{t,s}l_tN_tN_e$$

Generally, one tube outside diameter is chosen for each pressure level of the power cycle, the lowest the pressure, the larger the tube. This is a compromise between performance (tube diameter impact heat transfer, pressure drop and cost) and maintainability (numerous different tube diameters make maintenance more complicated). During design, the most dense heat exchanger arrangement must be placed at the top of the boiler tower in order to allow a minimal slag deposition and to promote heat transfer thanks to the flue gas higher velocity. The void fraction α can be calculated as follow:

$$\alpha = 1 - \frac{d_{t,e}\left(\frac{l_t}{N_p}\right)N_e}{w_{\text{furnace}}, l_{\text{furnace}}}$$

The height of the suspended heat exchanger can be calculated as follow:

$$h_{\text{SHX}} = 1.5d_{t,e}N_tN_p$$

The total height of the boiler is:

$$h_{\text{boiler}} = h_{\text{furnace}} + \frac{w_{\text{furnace}}\tan\theta + l_{\text{furnace}}}{2} + \sum h_{\text{SHX}} + 3$$

Pressure drops are calculated with the same method as in step.

A1.2. Short cut design of the coal-fired boiler

The boiler has been designed as a tangentially fired tower boiler. The combustion residence time combined with the disengagement zone residence time has been set to 2 s with a maximal flue gas velocity of 13 m/s at the furnace outlet. A 10% overdesign has been added. The proposed design shows a 35 m high furnace with a 13 m high disengagement zone and a 12 m high slag collector. The boiler section is 400 m^2: a 20 × 20 m square. Considering corrosion and creep strength, furnace wall tube and economizer tube are made of T91 martensitic steel which seems adequate for CO_2 in such condition, superheater are made of super 304H austenitic steel [30]. Due to the high amount of radiative heat released in the furnace (approximately 1300 MWth) and the high average temperature of the supercritical CO_2, the furnace wall are used for the three heat paths (main heat, first and second reheat) in the temperature range from 540 to 600 °C. Consequently the pressure drop in the furnace wall has to be kept to minimum value necessary to ensure correct heat transfer. A 4 passes arrangement has been chosen, two passes for main heat in the hottest part of the furnace (WALL 1a and 1b), one pass for the first reheat in the lower furnace zone (WALL 2) and one pass for the second reheat in the disengagement zone (WALL 3). A 30° spiral arrangement has been preferred in order to homogenate the tube outlet temperature. The Table 4 summarizes the main design parameters of these heat exchangers. The transfer coefficients are equal because of the simple zone model used to estimate radiative heat transfer.

Table 4
Details of furnace wall heat exchange areas design.

	Main heat	First reheat	Second reheat
Tube	2400	1200	1200
Tube diameter (m)	0.05	0.07	0.07
Specific flow rate (kg/s/m^2)	1780	1820	1820
Transfer coefficient (W/m^2/K)	150	150	150
Effective transfer area (m^2)	4900	2900	2200
Heat duty (MW)	660	385	280
Pressure drop (bar)	3.65	3.31	3.88

Considering the flue gas path, an economizer (ECO) and three superheaters are needed in order to achieve the maximal supercritical CO_2 temperature and cool down the flue gas to 540 °C. Each superheaters have been divided in two parts in order to increases the average temperature difference between flue gas and supercritical CO_2 and to allow a simple procedure to control the final CO_2 temperature (with the possibility to bypass the first superheater

(SH1, SH2 or SH3) to regulate the inlet temperature of the second superheater (FSH1, FSH2 or FSH3) and therefore its outlet temperature). Finally, a total of 7 suspended heat exchangers have been considered for the flue gas path. The Table 5 summarizes the main design parameters of these heat exchangers. Due to the low total heat duty exchanged in the flue gas path (approximately 1000 MW, 30% less than in a classical supercritical water boiler), the total height of the suspended exchangers zone is reduced to approximately 20 m. A second economizer (ECO 2) is used to preheated a fraction of CO_2 exiting the main compressor in parallel to the primary air preheater (rotary air preheater type). The Fig. 15 shows a representation of the designed boiler.

Table 5
Details of suspended heat exchangers design.

	FSH1	FSH2	FSH3	SH1	SH2	SH3	ECO	ECO_2
# element	75	75	75	125	125	100	150	420
# tube/element	8	8	8	8	8	8	8	8
# pass/tube	2	2	1	2	2	1	4	4
Tube diameter (m)	0.07	0.07	0.09	0.07	0.07	0.09	0.07	0.09
Area/element (m^2)	21	21	54	26	26	68	250	250
Transfer coefficient ($W/m^2/K$)	120	120	120	100	100	100	80	60
Transfer area (m^2)	1575	1575	4050	3250	3250	6800	37500	100000
ΔTml (K)	600	600	450	300	300	180	70	20
Heat duty (MW)	110	105	208	110	105	115	225	121
Void fraction (%)	60	60	50	35	35	35	25	15
Pressure drop (bar)	0.8	0.4	1.2	0.6	1.1	0.8	2.6	0.5

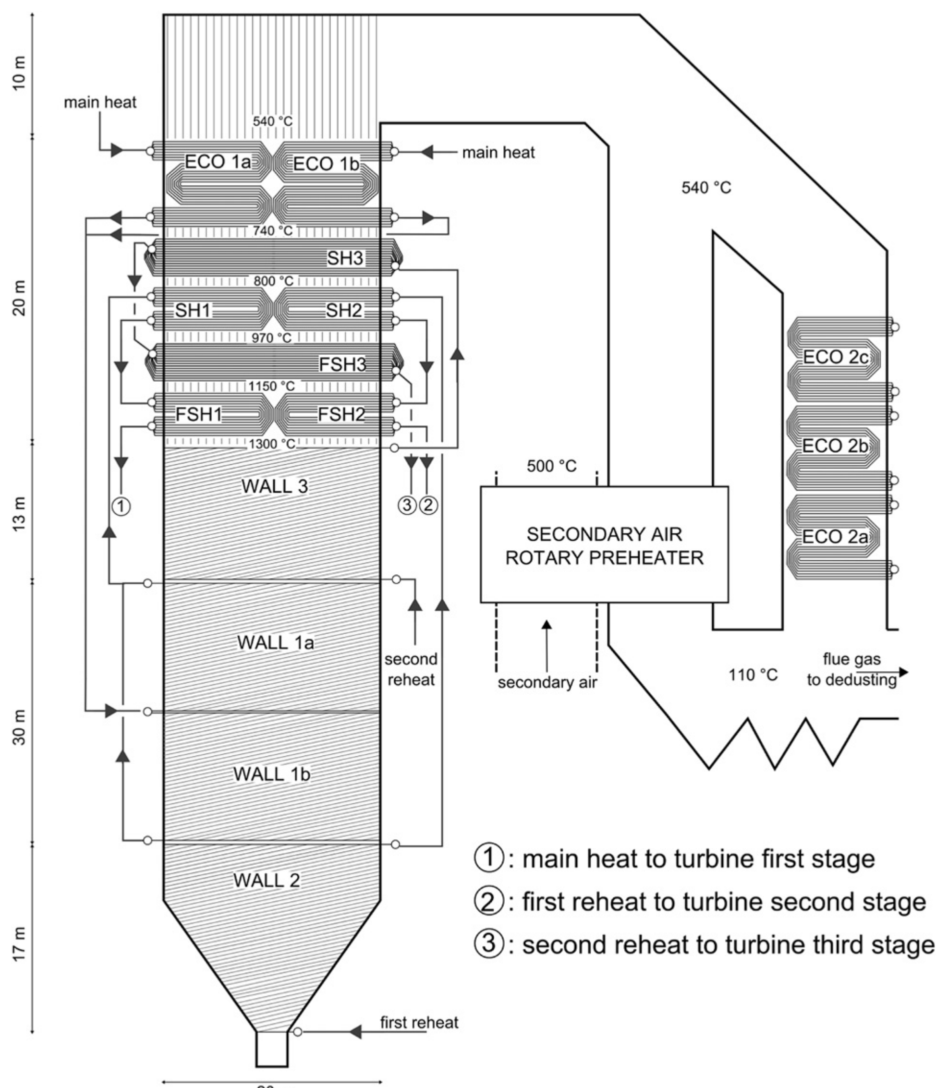

Fig. 15. Simplified representation of the designed boiler adapted to CO_2 supercritical Brayton cycle

Appendix 2. CO_2 cycle heat exchangers design and pressure drop evaluation

The three CO_2 cycle economizers are designed as a plate and frame heat exchangers, the following correlation [29] has been used for heat transfer coefficient evaluation:

$$Nu = 0.41Re^{0.65}Pr^{0.40}\left(\frac{\mu_{(T)}}{\mu_{H_2O(T)}}\right)^{0.14}$$

For turbulent flows (Re > 400 for a flow between parallel plates), pressure drop calculation can be calculated as follow [29]:

$$\Delta P = 8f\left(\frac{L}{2e}\right)\frac{\rho u^2}{2}, f = 0.125Re^{-0.3}$$

For simplification purpose, plate and frame heat exchangers have been designed as cuboids volumes. Due to the high pressure in these heat exchangers, a cylinder casing will be preferred for mechanical considerations; nevertheless the parallelepiped simplification allows a satisfactory overall heat exchange area and pressure drop evaluation. The Table 6 summarizes the design details of the three CO_2 cycle economizers.

Table 6
Summary of CO_2 cycle economizers design.

	CO_2 cycle economizer 1	CO_2 cycle economizer 2	CO_2 cycle economizer 3
Duty (MW)	200	900	3200
ΔTml (K)	8	10	20
h (W/m^2/K)	900	1200	1500
Area (m^2)	29,000	80,000	120,000
Type	Plate & Frame	Plate & Frame	Plate & Frame
Length (m)	8	10	12
Width (m)	6	8	12
Gap between plates (mm)	7.5	5.0	6.0
Number of plate	600	1000	900
Specific area (m^2/m^3)	115	210	170
Cold side ΔP (bar)	0.12	0.52	1.22
Hot side ΔP (bar)	0.73	1.92	4.07

Appendix 3. Evaluation of number of compression stages

The number of compression stage of a compression is function of fluid properties (molar mass, density, heat capacity and gamma) and compressor design (height of blade, blades design). The following expression [31] allows the calculation of enthalpy variation in a compression stage:

$$\Delta h = UV(\tan\alpha_1 - \tan\alpha_2)$$

with U the blade velocity, V, the axial gas velocity and α_1 and α_2 specific angles of the blade design [31]. For a given blade design ($\alpha_1 = \pi/3$ and $\alpha_2 = 0$), a first evaluation of the number of stage needed for CO_2 compression is possible. With conservative dimension for shaft diameter, the main CO_2 compressor can be design with 9 compression stage which is similar to classical gas turbine compressor. A larger shaft (and a higher blade velocity) allows a significant reduction of compression stage, a 3 staged compressor could be designed. Moreover, due to very high CO_2 density in the main compressor, it could be replaced by one staged centrifugal pumps [7]. This change will improve the cost and the availability of the plant. For the hot CO_2 compressor, a larger number of stages are needed due to the larger enthalpy and density variation. Nevertheless, 14 compression stages are sufficient for this compression. A larger shaft diameter could reduce this number of stage down to 8. The Table 7 summarized the compressor sizing. Study [8] have carried out a more detailed and optimized design and found that a 5 stages main compressor and 8 staged auxiliary compressor is acceptable.

Table 7
Summary of CO_2 compressor design.

	Main CO_2 compressor	Hot CO_2 compressor
Power (MW)	250	370
Isentropic efficiency	90	90
Mass flow rate (kg/s)	5510	3240
Δh (kJ/kg)	45.1	114.4
Inlet density (kg/m^3)	625	135
Outlet density (kg/m^3)	669	322
Shaft diameter (mm)	800	900
Inlet blade height (mm)	30	50
Outlet blade height (mm)	25	20
Rotation (Hz)	60	60
Maximal CO_2 axial velocity (m/s)	120	160
Average blade velocity (m/s)	25	28
Number of stage	9	14
Approximate length (m)	2.2	3.5

Appendix 4. Main auxiliary consumption

Table 8
Main plant auxiliary consumption.

Name	Work (MW)	Head (bar)	Efficiency (%)
Coal mill	5.0	–	–
Cooling water pumps	8.7	5.0	85
FGD pumps	9.8	8.0	80
Air main fan	2.1	1.05	65
Air second fan	1.3	1.15	65
Flue gas fan	28.8	1.20	65
Rich solvent pump	3.1	8.0	85
Lean solvent pump	2.5	7.0	85
Cooling water pump	4.5	5.0	85
LVC compressor	15.5	2.0	90
Overhead compressor	39.7	6.0	90
CO_2 compressor 1	31.0	26.0	90
CO_2 compressor 2	22.7	110	90

Appendix 5. Design details of drums and columns

Table 9
Main characteristics of drums and columns.

Name	Number	Diameter (m)	Height (m)	Type
CO_2 cycle storage drum	2	6	34	Horizontal
FGD column	2	15	40	4 spray banks, 2 perforated plates, direct contact cooling water on top, demister
CO_2 absorber column	2	17	40	15 m structured packing, two stages water wash on top, high efficiency demister
CO_2 Stripper column	2	6	40	15 m structured packing
Condenser drum	2	3	8	Horizontal, demister
LVC drum	2	4	12	High efficiency demister
SOC drum	2	4	12	High efficiency demister

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