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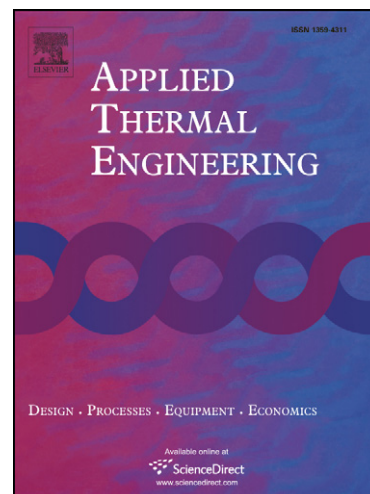
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Optimization of intercooling compression in CO₂ capture systems

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Abstract

Compression of CO₂ is an essential process in the development of carbon capture and storage (CCS) technologies. In spite of power requirements for CO₂ compression could be as much as 100 kWe per tonne CO₂, the minimization of energy requirements has received little attention in the literature. Although intercooling compression reduces power requirements, it introduces important cooling necessities that could be minimized.

The aim of this paper is the integration of intercooling compression into the low-pressure part of a steam cycle to take advantage of the intercooling heat and analyse the energetic and economical results under different assumptions. Simulation and optimization have been performed in order to evaluate the intercooling configuration, energy requirements and the most cost-effective integration. Results have shown reduction in compression power requirement around 40% and reduction of the incremental COE around 23%. Proposed integration could be used to increase the efficiency of CO₂ capture processes and, therefore, to reduce the CO₂ capture cost.

Keywords: intercooling compression, CO₂ capture, energy optimization

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Nomenclature

| | |
|----------------------|--|
| CF | capacity factor |
| COE | cost of electricity (€/MWh) |
| Cost _{HE} | heat exchanger cost (€) |
| Cost _{TEG} | TEG cost (€) |
| Cost _w | compressor cost (€) |
| F | annuity factor |
| f _{M&S} | Marshall and Swift index |
| h _{in} | inlet stream enthalpy from compressor (kJ/kg) |
| h _{out} | exit stream enthalpy from compressor (kJ/kg) |
| m | mass flow feeding the compressor (kg/s) |
| m _i | water mass flow (kg/s) |
| m _{co2} | CO ₂ mass flow through compression (kg/s) |
| m _{h2o} | water mass flow before compression (kg/s) |
| MWh | electricity production (MWh) |
| OF | objective function |
| pr | compressor pressure ratio |
| Q _{cooling} | intercooler heat (kW) |
| Q _{heating} | transferred heat to steam cycle (kW) |
| TCR | Total capital requirement |
| TEG | Tri Ethylene Glycol unit |
| T _{he} | first heat exchanger outlet temperature (°C) |
| T _{in} | compressor inlet temperature (°C) |
| T _w | condensate water outlet temperature (°C) |
| W | compressor power (kW) |
| W _{extraST} | extra power by steam turbine (kW) |

| | |
|------------------------|---|
| X_{water} | water content in flow to compression |
| η | compressor isentropic efficiency |
| ΔCOE | incremental cost of electricity (€/MWh) |
| ΔT_{lm} | logarithmic mean temperature difference (K) |

1. INTRODUCTION

Recently, there has been a growing interest in the study of CO₂ capture processes as an option to reduce greenhouse gasses emission and mitigate the global warming. Many recent studies have focused on the analysis of post combustion capture, oxyfuel and pre combustion capture schemes. In particular, the development of zero (or near zero) emissions power plant technologies is gaining rapid momentum worldwide and some large demonstration projects are expected in the coming decade for new plants [1]. Total capital requirement (TCR), cost of electricity (COE), CO₂ avoided and cost per ton of CO₂ avoided are the essential variables used to compare different options, configurations and new developments.

Although considerable research has been devoted to capture processes, rather less attention has been paid to CO₂ compression. As an intermediate and essential process between capture from flue gases and a safe storage, CO₂ treatment for transport plays an important role in efficiency penalty and total capital requirement associated with capture processes. It is required temperature and pressure over the critical point due to CO₂ specific volume at supercritical conditions is more than 500 times smaller compared with values at gas phase (0,8 m³/kg at 150°C and 1bar; 0,00158 m³/kg at 40°C and 100bar). Geologic storage requires a high density CO₂ stream with a suitable pressure for safe storage and capacity maximization. These temperature and pressure values are also necessary for pipeline transport because it reduces pipe diameter (although increasing thickness). For tank transport, conditions differ completely, 6.5 bar and approximately -50°C are needed in order to be economically feasible and remain in liquid phase [2].

Around 25% of the total capital requirements are due to compression equipment cost. Energy requirements for gas conditioning are typically 90-120 kWh/tonne CO₂ [2]. For an existing sub-critical power plant with 38% efficiency, 0.9 tonneCO₂ is emitted per MWh produced, so around 100-133 kWh are needed for compression per MWh produced. With COE around 35 €/MWh the additional

COE due to CO₂ compression power requirement is 3.5 €/MWh. These values highlight the necessity of detailed analysis in order to minimize both, energetic requirements of the compression and additional capital requirement for the capture system

Compression process includes condensation of water before CO₂ conditioning and generally dehydration in the CO₂ compression intercooling is also required [3-9]. A recent research also includes the removal of other components in CO₂ rich-stream [2]. This interesting work focused on the necessity of different process flow schemes for CO₂ compression depending on transport specifications, ship or pipeline, and took into account the gas quality (depending on capture process) before the removal of water, other liquids and volatile gases in the gas conditioning.

Gas conditioning in oxyfuel applications have deserved major effort at research [3,4,6,7]. Detailed descriptions of the compression process for a 900 MWe oxyfuel capture system have been previously presented [3,4]. Research was focused on system analysis, equipment sizing and economic evaluation. Due to the high moisture content in the gases (German lignite), a flue gas condenser was included to take advantage of condensation heat in the feed-water preheating. Compression was designed in some stages with intercooling, a two-stage compression increases pressure up to 30 bar, a Tri Ethylene Glycol (TEG) unit eliminates moisture and avoid sweet corrosion. Another compression stage increased the pressure up to 58 bar. The stream is then cooled down and CO₂ is liquefied to install a pump for increasing the pressure up to 100 bar. Cooling necessities for compressor intercooling were approximately as high as power demands, 53.3 MWth for cooling and 58 MWe for power. Moreover the CO₂ condensation demanded 35.2 MWth in a condenser with an important heat transfer area (mean logarithmic temperature difference of 3°C) and additionally 3.6 MWth for the sub cooling of CO₂ previous to the final compression. However, it remains unclear whether it is possible to recover energy from the compression process in order to reduce the COE of capture systems and hence the cost per ton of CO₂ avoided.

The aim of this paper is to analyze and optimize the design of a CO₂ intercooling compression system taking advantage of the low temperature heat duty in the low-pressure heaters of a steam cycle. The novelty of this work includes the separation of the intercooling in two heat streams, one to be used in a steam cycle and another to be released in a cooling system. Integration of this energy in a steam cycle has been previously reported [10] for amine scrubbing in an existing power plant. The objective functions for optimization are energy and exergy requirements, and total cost, including Total

Capital Requirement (TCR) plus operation and maintenance cost. The paper is organized as follows: A description of the intercooling compression is given in section 2. The equations and the optimization problem are described in section 3. Results and discussion are included in section 4 and, finally, conclusions are given in section 5.

2. CO₂ INTERCOOLING COMPRESSION

The basic configuration of a CO₂ intercooling compression system is illustrated in Figure 1. The CO₂-water stream coming from a post combustion capture process or from oxyfuel combustion is firstly cooled down in order to condensate and separate the water content. Compression process is divided in several stages. Assuming a maximum total pressure relation of five and a final CO₂ pressure of 100 bar, three or four radial compressor steps are necessary [2]. Compression up to 30 bar is usually designed with a three-stage process, compressors with the same pressure ratio and an isentropic efficiency around 80%. The net work required by each compressor can be reduced by means of multistage intercooling. Since the solubility of water in CO₂ gas decreases with pressure [11] a TEG is recommended for drying process at the pressure of 30 bar [3]. Adsorption with molecular sieves or silica gel have been also proposed in the literature [2]. A water concentration of 60 ppm can be reached using any of these systems. After the dehydration stage, one additional compression stage is required to make the CO₂ stream pressure reach 100 bar.

The positive effects of intercooling are the reduction of power needs for gas compression and therefore the compressor size and TCR. In general this heat is rejected to low temperature cooling equipment in order to reduce compression penalty. This strategy is beneficial for operation, especially in cold locations, but TCR could increase due to the necessity of larger heat exchangers for gas cooling in locations with higher temperatures.

Three variables influence the compressor power consumption and cooling necessities: compressor efficiency, pressure ratio (i.e. compression stages) and CO₂ inlet temperature. Figure 2 shows the effect of these variables in power and cooling demands. Approximately the effect of varying inlet temperature from 30 to 50°C is an increase of 10% in power requirements due to suction volume augmentation with temperature. Increment of outlet temperature is higher than inlet temperature, so cooling necessities also increases up to 30%. A reduction in the compressor isentropic efficiency, from 90 to 75%, causes an important increase (around 20%) in power and cooling necessities. The effect of

increasing pressure ratios, or reducing the number of compressor stages from 4 to 3 (increasing pressure ratio from 3.1 to 5.5), is more important than those previously mentioned for a single compressor. Evidently, with pressure ratio of 5.5, one compressor stage is eliminated.

The CO₂ compressor outlet temperatures are shown in Figure 3. For a pressure ratio of 3.1 temperature ranges are 110-165°C depending on efficiency and inlet temperatures. Values increase with pressure ratio of 5.5 up to 160-225°C. This energy, with important temperature level, could be integrated into the capture process or a power plant steam cycle in order to reduce energy penalty associated with CO₂ compression. In this case, the main disadvantage is that the temperature level of the water coming from condenser is higher than cooling water temperature. As a consequence, the CO₂ compressor inlet temperature increases and also the power necessities, reducing or cancelling some of the benefits of energy integration.

With the aim of studying the configuration and values that optimize the intercooling compression in CO₂ capture system, i.e. minimization of the incremental COE associated with CO₂ compression, a proposed intercooling layout is presented in Figure 4. The novelty is that the cooling process is divided in two-stages. The heat extracted from the first one is a useful energy and it could be used in the low-pressure part of the steam cycle for water preheating (Q_{heating}). This strategy could reduce steam turbine bleedings mass flow and increases the turbine gross power output. The heat extracted from the second one has to be dissipated (Q_{cooling}), for example in the power plant cooling system (cooling tower). This stage would decrease CO₂ temperature in order to reduce compressor power requirements. The use of heat from intercooling in the preheating section of steam cycle enlarges the problem with new questions to solve:

- Firstly, regarding intermediate and compressor inlet temperatures. Intermediate temperature has a limitation depending water from condenser temperature plus a temperature difference in heat exchangers. It is evident that lower compressor inlet temperatures means a reduction of compressor power needs, but it increases the cooling requirements and heat transfer area.
- Secondly, the number of compressor stages and compressor isentropic efficiency. With four intercooler stages the compressor outlet temperatures are maintained in low-medium values and heat qualities are also low (i.e. it is only possible to transfer this energy to the very low temperature section of steam cycle that is condenser outlet). However, if the

number of stages is reduced, compressor power requirements increase but it is possible to take advantage of more energy (heat) with higher temperatures in a medium temperature section of steam cycle. The same effect is observed when reducing the compressor efficiency.

The independent variables that influence the system are the compressor efficiencies, pressure ratios (i.e. compression stages), CO₂ intermediate temperatures and compressor inlet temperatures.

3. OPTIMIZATION OF INTERCOOLING COMPRESSION

To study the performance of the system under different situations and implementing a optimization, a process simulation has been developed [12]. The objective function for optimization was the incremental COE associated with CO₂ compression. This function includes the capital investment for main components and net compression power consumption.

The effects of compressor efficiency, pressure ratio, intermediate temperature and compressor inlet temperature have been studied in order to minimize the incremental COE.

For reference purposes a COE of 36.5 €/MWh has been considered. It has been supposed a 400 MW(net) power plant with specific emissions of 720 kg/MWh and 80% of plant capacity factor (CF). CO₂ mass flow to compression is assumed 72 kg/s, plus a water content of 15% (w).

The objective function includes the capital investment and compressor consumption,

$$OF = \Delta COE = \frac{(COE_{ref} * MWh_{ref}) + F * \sum TCR}{MWh} - COE_{ref} \quad (1)$$

Where F is 0.1, the annuity factor for an interest rate of 8.78% and 25 year project life. The new net power production, MWh, depends on the reference production, compressor power requirements and extra power in steam turbine (equation 2). The total compression power as the sum of the work from different compression stages (equation 3). The effect of taking advantage of intercooling heat is the reduction of steam turbine bleedings in the low-pressure part of the steam cycle. Consequently, mass flow in this part increases and steam turbine could be designed with an extra power production. The compressor output temperatures-enthalpies are calculated with compressor isentropic efficiencies and assuming negligible heat transfer with the ambient and no appreciable kinetic and potential energy variations (equation 4).

$$MWh = MWh_{ref} - \left(\sum_{stages} W_i * 8760 * CF \right) + (W_{extraST} * 8760 * CF) \quad (2)$$

$$W_i = \dot{m} * (h_{out_i} - h_{in_i}) \quad (3)$$

$$h_{out_i} = h_{in_i} + \frac{(h_{out_i_s} - h_{in_i})}{\eta_c} \quad (4)$$

Total capital requirement (TCR) is the sum of main components cost: Compressor, TEG unit and heat exchangers. Cooling process has been divided in two-stages. Heat exchanger cost is separated in the cost for equipment that heat condensate from steam cycle (HEheat) and the cost of cooling heat exchanger (HEcool).

$$\sum \text{Cost} = \text{Cost}_W + \text{Cost}_{TEG} + \text{Cost}_{HE} \quad (5)$$

$$\text{Cost}_{HE} = \text{Cost}_{HEheat} + \text{Cost}_{HEcool} \quad (6)$$

For compressors cost estimation, the experience of manufactures in calculating this capital cost as a function of their capacity and efficiency is essential. Such experience is not always available or reliable in open literature. In some cases, there are important differences in investment costs equations [13-15]. One of the recent proposals for the function cost for compression depends on the power requirements, compressor type and Marshall & Swift index [16] and it is given by [13], equation (7) assuming a centrifugal, reciprocating compressor.

$$\text{Cost}_{Wi} = 517,5 * 2,11 * W_i^{0.82} * f_{M\&S} \quad (7)$$

The function cost for heat exchangers is taken from [17] adapted from [18], equations (8,9)

$$C_{HEcool} = 3650 \left(\frac{Q_{cooling}}{\Delta T_{lm}} \right)^{0.7948} f_{M\&S} \quad (8)$$

$$C_{HEheat} = 3650 \left(\frac{Q_{heating}}{\Delta T_{lm}} \right)^{0.7948} f_{M\&S} \quad (8)$$

Where Q is the heat exchanged between cold and hot streams in kW, ΔT_{lm} the logarithmic mean temperature difference in K.

The cooling necessities ($Q_{cooling}$) and heat to the steam cycle ($Q_{heating}$) are given by equations (9) and (10):

$$\begin{aligned} \sum_j Q_{cooling} &= Q_{cooling_1} + Q_{cooling_2} + Q_{cooling_3} + Q_{cooling_4} + Q_{cooling_5} = \\ &= \dot{m}_{co2} * (h_3 - h_4) + \dot{m}_{co2} * (h_7 - h_8) + \dot{m}_{co2} * (h_{10} - h_{11}) + \\ &+ \dot{m}_{co2} * (h_{13} - h_{14}) + \dot{m}_{co2} * (h_{17} - h_{18}) \end{aligned} \quad (9)$$

$$\begin{aligned} \sum_j Q_{\text{heating}} &= Q_{\text{heating}_1} + Q_{\text{heating}_2} + Q_{\text{heating}_3} + Q_{\text{heating}_4} + Q_{\text{heating}_5} = \\ &= [\dot{m}_{\text{CO}_2} * (h_1 - h_3) + (\dot{m}_{\text{H}_2\text{O}} * (h_1 - h_2))] + \dot{m}_{\text{CO}_2} * (h_6 - h_7) + \\ &+ \dot{m}_{\text{CO}_2} * (h_9 - h_{10}) + \dot{m}_{\text{CO}_2} * (h_{12} - h_{13}) + \dot{m}_{\text{CO}_2} * (h_{16} - h_{17}) \end{aligned} \quad (10)$$

The cost function for active dehydration with TEG, equation (11), is derived from the values in [3]. Where $C_{\text{TEG,ref}}$ is 2 million euro and CO_2 mass flow at reference is 205.86 kg/s.

$$C_{\text{TEG}} = C_{\text{TEG,ref}} \cdot \left(\frac{\dot{m}_{\text{CO}_2}}{\dot{m}_{\text{CO}_2,\text{ref}}} \right)^{0.6} \cdot f_{\text{M\&S}} \quad (11)$$

In order to compare the thermal energy that is possible to take into account with the proposed process diagram, it is suggested to evaluate the electrical energy produced by means of the integration in the low-pressure section of a steam cycle. Two or three low-pressure heaters could be eliminated and, consequently, will increase net power output. As an example of a general power plant, it is supposed that the pressure of these extractions are 1.5 and 0.7 bar of superheated steam at 160°C and 110°C, and 0.3 bar of saturated steam (69.1°C). It is also supposed that extractions have the same mass flow, which is approximately correct as last supercritical cycle designs suggest. With the energy balance shown in equation (12) it is possible to calculate the reduction in extractions mass flow.

$$Q_{\text{heating}} = \dot{m}_1 * \left(h_{1.5\text{bar}, 160^\circ\text{C}} - h_{0.3\text{bar}, \text{sat_liq}} \right) + \dot{m}_2 * \left(h_{0.7\text{bar}, 110^\circ\text{C}} - h_{0.3\text{bar}, \text{sat_liq}} \right) + \dot{m}_3 * \left(h_{0.3\text{bar}, \text{sat_ste}} - h_{0.3\text{bar}, \text{sat_liq}} \right) \quad (12)$$

Left-hand side of the equation is the heat taken from the intercooling compression, and the final enthalpy of the extraction is saturated liquid at 0.3 bar (lowest pressure heater outlet). Extra power in steam turbine, equation 13, is the power produced in the expansion of these steam bleedings from the original conditions to the condenser inlet (assuming a condenser pressure of 0.045 bar).

$$W_{\text{extraST}} = \dot{m}_1 * \left(h_{1.5\text{bar}, 160^\circ\text{C}} - h_{0.045\text{bar}} \right) + \dot{m}_2 * \left(h_{0.7\text{bar}, 110^\circ\text{C}} - h_{0.045\text{bar}} \right) + \dot{m}_3 * \left(h_{0.3\text{bar}, \text{sat_ste}} - h_{0.045\text{bar}} \right) \quad (13)$$

The configuration requires the constraints equations displayed in table 1. Constrains include the water content in CO_2 stream (X_{water}) and CO_2 properties (enthalpy, pressure and temperatures) at first compressor inlet, TEG and last compressor outlet. Constraints also include energy and mass balances for each equipment and negligible pressure drop through intercoolers.

The process simulation was implemented with one objective function, 252 optimization variables and 252 constrain equations. Independent variables include: output temperature from first heat exchanger in each stage ($T_3 = T_7 = T_{10} = T_{13}$), compressors inlet temperature ($T_5 = T_8 = T_{11} = T_{15}$) and compressors pressure ratios and efficiencies.

For a specific integration of this scheme into a power plant an additional constraint is needed, that is, the total heat transferred to the steam cycle must not exceed the maximum specified. In parallel with solving the optimization problem, it is necessary to know the possibility of integrating the energy from compression in a steam cycle. An analysis using pinch methodology has been implemented to calculate the final condensate temperature.

The grand composite curves for some relevant cases are shown in Figure 5. The heat integration only presents a pinch point when compressor inlet temperature is lower than steam cycle condensate temperature plus a temperature difference. Additional cooling is only required in this situation, Figure 5a. The excess of heat above the pinch shows the possibility of integrating the low-pressure part of a steam cycle. The hot stream is the CO_2 coming from compressor, the heat is exchanged with low pressure water substituting some low pressure heaters.

Final water temperature depends on integration and variables affecting compressor performance. In order to have the same reference to compare results, an external heat source maintains this temperature in a constant value (T_{ref}). The heat source is the 1.5 bar pressure bleeding of the steam turbine. Equation 14 shows the bleeding mass flow necessary to obtain a constant water temperature T_{ref} .

$$\Delta \dot{m}_1 = \frac{\dot{m}_1 * c_p * (T_{ref} - T_w)}{\left(h_{1,5\text{bar}} - h_{0,3\text{bar}} \right)} \quad (14)$$

160°C sat_liq

4. RESULTS AND DISCUSSION

Under usual optimization objectives, compression work should be minimized. That leads to reduce inlet compressor temperatures with the limit of cold reservoir plus a temperature difference in cooling equipment (e.g. 10°C). For example, for cold reservoir temperature (ambient temperature or water) around 20°C and a temperature difference of 10°C the minimum compressor inlet temperature could be 30°C . For the reference case, the energy from CO_2 intercooling is not integrated into the steam cycle, therefore the compressor inlet temperature is 30°C . Assuming four compression stages

with an isentropic efficiency of 80% the compression total cost is 28.1 M€, approximately 90% due to compressors (25.1 M€), 1.8 M€ of heat exchangers and 1.2 M€ for gas treatment. Compressor power necessities are 26.7 MW and the COE is 40.04 €/MWh. Therefore, the incremental COE due to CO₂ compression is 3.54 €/MWh.

When energy is integrated into the steam cycle, the minimum intermediate temperature depends on the water temperature coming from condenser and a heat exchanger temperature difference. Assuming 15°C as temperature difference and 30°C the water from condenser, this minimum intermediate temperature is 45°C. If lower compressor inlet temperatures are required it is necessary to reduce this temperature down to 30°C in the second heat exchanger. The investment cost increases due to the new heat exchanger but operational cost will be reduced due to lower power requirements. Higher compressor inlet temperatures could be selected with smaller equipment, reducing investment cost and increasing operational cost. The aim of this work is to minimize the sum of investment and operational cost.

Tables 2-3 show a comparison between the base case (cooling without integration) and the results of integration for intermediate/compressor inlet temperatures of 45/30, 45/45 and 60/60°C for compressor efficiency of 80% and four compression stages.

The first option needs the second heat exchanger for cooling the CO₂ from 45°C down to 30°C but the cooling requirements have been reduced from 64.3 MW to 4.1 MW. Compressor power consumption is equal to the base case, 25.4 MW, because the compressor inlet temperature does not change, 30°C. The integration of intercooling takes advantage of 59.6 MWth into the steam cycle. It reduces the steam turbine bleedings producing an extra power of 8.4 MW and reducing the net power necessities to 17.0 MW. To obtain these values, higher heat exchanger equipment is needed because temperature difference between CO₂ and water from condenser is lower than in reference case. The heat exchanger equipment doubled its cost up to 3.55 M€, table 3, but COE is reduced from 40.04 to 39.23 €/MWh. This is a reduction of 22% in the incremental COE due to compression from 3.54 to 2.73 €/MWh.

There is a minimum COE when the second heat exchanger for cooling is eliminated. The compressor inlet temperature is equal to the intermediate temperature and higher than the base case. This causes an important increase in power demand from 25.4 to 26.9 MW but there is also an important increase in available energy (see Figure 1) that increases the extra power in steam turbine

up to 10.4 MW. The net power consumption is 16.5 MW, slightly lower than previous case but 10.2 MW lower than base case. Compressor cost is higher than analysed cases due to higher consumption and specific volume. Heat exchangers costs are lower than before due to higher temperature difference. As a consequence the COE is 39.21 €/MWh, 0.02 €/MWh lower than previous case and a reduction of 23.5% with the base case.

Increasing temperatures in compressor inlet with the objective to reduce heat exchanger cost do not improve results. For a compressor inlet temperature of 60°C the COE is 39.34 €/MWh higher than previous values. In this case the reduction of heat exchanger cost is not compensated by the increase in compressor cost and compressor power necessities.

Tables 4-5 illustrate the effect of isentropic efficiency and pressure ratio on the incremental COE. In table 4 a compressor efficiency of 90% has been assumed. The base case has a compression total cost of 25.7 M€ mainly due to compressors (22.8 M€). Compressors power necessities are 28.0 MW and the COE is 39.65 €/MWh. Therefore, the incremental COE due to CO₂ compression is 3.15 €/MWh. This value is 0.39 €/MWh lower than less efficient compressors. For this efficiency, differences in the incremental COE between options increase. The COE for the best option is 38.86 €/MWh, 0.04€/MWh lower than the case of cooling CO₂ down to 30°C. This represents a reduction of 25.1% with the base case of 90% efficiency. For compressor efficiency of 70% heat integration is not able to compensate the increase of compressor power when low efficiency equipment is used. Consequently, for this efficiency there are no differences in the incremental COE between the options of cooling CO₂ down to 30°C or eliminate the second stage heat exchanger, 3.17 €/MWh.

The effect of increasing the pressure ratio (reducing the number of stages from four to three) is showed in table 5. In general COE values increase, but the best option is reducing the compressor inlet temperature to 30°C with the second heat exchanger. COE is 0.02 €/MWh lower than the option of 45°C and 0.89 €/MWh lower than base case.

Figure 6 shows a possible integration scheme of intercooling compression and low-pressure section of a steam cycle for the case of 45/45°C, 80% efficiency and four stages.

With data from previous optimization, table 7, the reduction in each steam extraction mass flow is 8.7 kg/s for the 1.5 bar pressure bleeding and 9.0 kg/s for the 0.7 and 0.3 bar pressure bleedings, and therefore the extra power in steam turbine is 10.4 MWe.

Present results provide support for the proposed integration, with a reduction of the incremental COE of compression from 3.54 to 2.71 €/MWh. Anyway, specific studies are needed in order to fully integrate the intercooler heat streams into the capture system or steam cycles and to obtain a detailed compressor cost data that improve the cost analysis of the overall system.

5. CONCLUSIONS

CO₂ compression introduces an important energy and efficiency penalty in CCS technologies. Intercooling compression is used to reduce compression power requirements. This heat is usually dissipated in the cooling equipment.

A CO₂ intercooling compression process flow diagram has been proposed. Intercooling has been divided in two-stages. The heat transferred in the first one is a useful energy and it could be used in the low-pressure part of the steam cycle for water preheating. The second one has to be dissipated in the cooling system. The second stage increases the cost of heat exchangers but compressors cost and power necessities decrease. A thermal simulation has been presented to calculate the minimum incremental COE due to CO₂ compression under several assumptions.

Results from energetic and economical analysis of different compression options have been suggested: firstly, the integration of CO₂ intercooling waste energy into the steam cycle reduces 23% the incremental COE associated with compression (approximately 0.8 €/MWh), also there is a TCR augmentation around 2 M€ but power necessities reduce around 40% (10 MW as average); secondly, the compressor efficiency has an important influence in economical results, it has been calculated a reduction of 0.3-0.4 €/MWh when results of 90% and 80% compressor efficiency are compared; thirdly, the reduction of compressor stages does not improve energetic or economic results; and finally the interest of proposed two-stage intercooling has been demonstrated for high pressure ratio compressors (three compressor stages). With four compressor stages it is not necessary the second heat exchanger to obtain a minimum incremental COE.

Proposed integration results could be used to reduce the energy and efficiency penalty of CO₂ capture processes and, therefore, to reduce the CO₂ capture cost. However a detailed integration and optimization is required for each singular CCS application.

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| | | |
|-----------------------------------|--|-----------------------------|
| $h_{\text{co2_i}} = f(p_i, T_i)$ | $T_2 = T_3$ | $P_{14} = 30\text{bar}$ |
| $T_1 = 110^\circ\text{C}$ | $T_4 = T_5$ | $T_{14} = T_{15}$ |
| $P_1 = 1\text{bar}$ | $T_{3,7,10,13,17} \geq 45^\circ\text{C}$ | $P_{18} = 100\text{bar}$ |
| $X_{\text{water_1}} = 0.15$ | $T_{5,8,11,15} \leq 45^\circ\text{C}$ | $T_{18} = 60^\circ\text{C}$ |

Table 1. Optimization constraints

| T_{he} (°C) | T_{in} (°C) | W (kW) | $Q_{heating}$ (kW) | $Q_{cooling}$ (kW) | T_w (°C) | W_{TV} (kW) | $W-W_{TV}$ (kW) |
|------------------|------------------|-------------|-----------------------|-----------------------|---------------|------------------|--------------------|
| Base case | 30 | 25368 | - | 64344 | - | - | 26737 |
| 45 | 30 | 25386 | 59590 | 4077 | 109.0 | 8369 | 17017 |
| 45 | 45 | 26852 | 65151 | 0 | 109.0 | 10389 | 16463 |
| 60 | 60 | 28298 | 65919 | 0 | 109.0 | 10668 | 17630 |

Table 2. Energetic comparison between base case and integration for different intermediate/compressor inlet temperatures (80% compressor efficiency and 4 stages)

| T_{he} (°C) | T_{in} (°C) | C_W (10 ⁶ €) | C_{HE} (10 ⁶ €) | TCR (10 ⁶ €) | COE (€/MWh) | Δ COE (€/MWh) |
|------------------|------------------|------------------------------|---------------------------------|----------------------------|----------------|-------------------------|
| Base case | 30 | 25.10 | 1.75 | 28.06 | 40.04 | 3.54 |
| 45 | 30 | 25.10 | 3.55 | 29.94 | 39.23 | 2.73 |
| 45 | 45 | 26.30 | 3.26 | 30.76 | 39.21 | 2.71 |
| 60 | 60 | 27.50 | 2.39 | 31.09 | 39.34 | 2.84 |

Table 3. Economical comparison between base case and integration for different intermediate/compressor inlet temperatures (80% compressor efficiency and 4 stages)

| T_{he} (°C) | T_{in} (°C) | C_W (10 ⁶ €) | C_{HE} (10 ⁶ €) | TCR (10 ⁶ €) | COE (€/MWh) | Δ COE (€/MWh) |
|------------------|------------------|------------------------------|---------------------------------|----------------------------|----------------|-------------------------|
| Base case | 30 | 22.80 | 1.72 | 1.20 | 39.65 | 3.15 |
| 45 | 30 | 22.80 | 3.44 | 1.20 | 38.90 | 2.40 |
| 45 | 45 | 23.90 | 3.16 | 1.20 | 38.86 | 2.36 |
| 60 | 60 | 25.00 | 2.29 | 1.20 | 38.98 | 2.48 |

Table 4. Economical comparison between base case and integration for different intermediate/compressor inlet temperatures (90% compressor efficiency and 4 stages)

| T_{he} (°C) | T_{in} (°C) | C_W (10 ⁶ €) | C_{HE} (10 ⁶ €) | TCR (10 ⁶ €) | COE (€/MWh) | Δ COE (€/MWh) |
|------------------|------------------|------------------------------|---------------------------------|----------------------------|----------------|-------------------------|
| Base case | 30 | 26.20 | 1.58 | 1.20 | 40.23 | 3.73 |
| 45 | 30 | 26.20 | 2.98 | 1.20 | 39.34 | 2.84 |
| 45 | 45 | 27.40 | 2.68 | 1.20 | 39.36 | 2.86 |
| 60 | 60 | 28.60 | 2.37 | 1.20 | 39.51 | 3.01 |

Table 5. Economical comparison between base case and integration for different intermediate/compressor inlet temperatures (80% compressor efficiency and 3 stages)

Fig.1

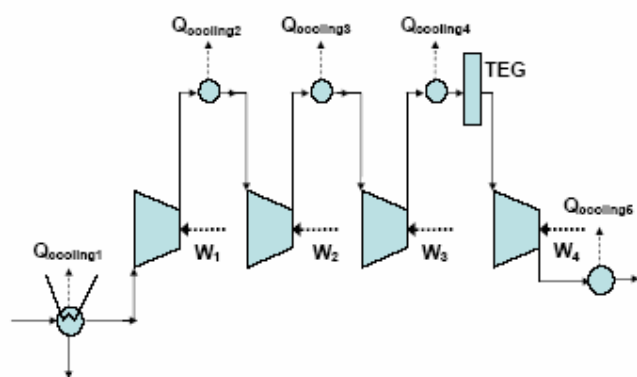


Fig.2

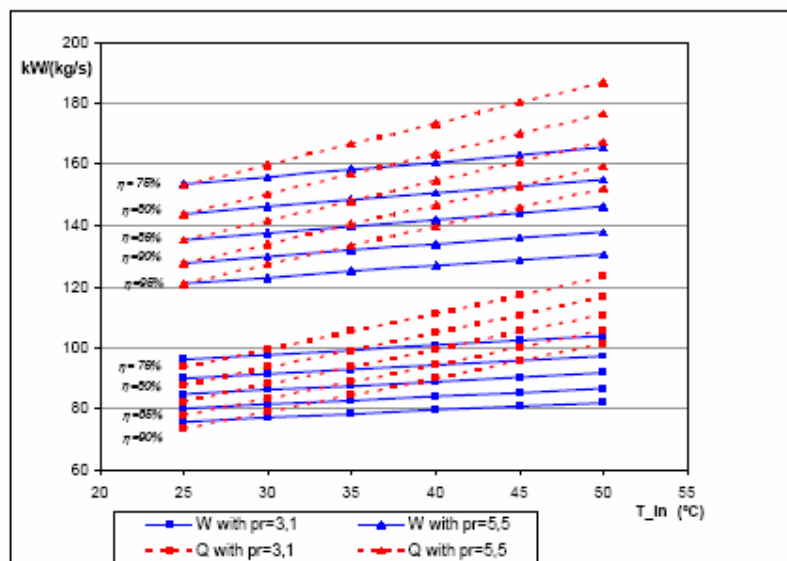


Fig.3

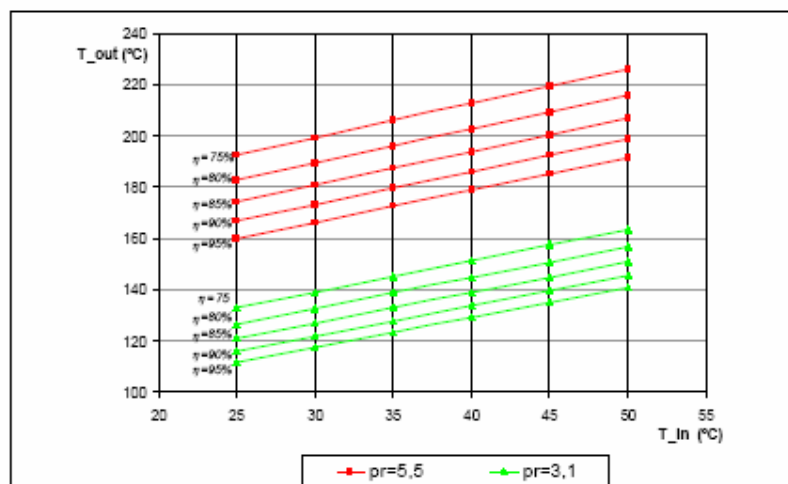


Fig.4

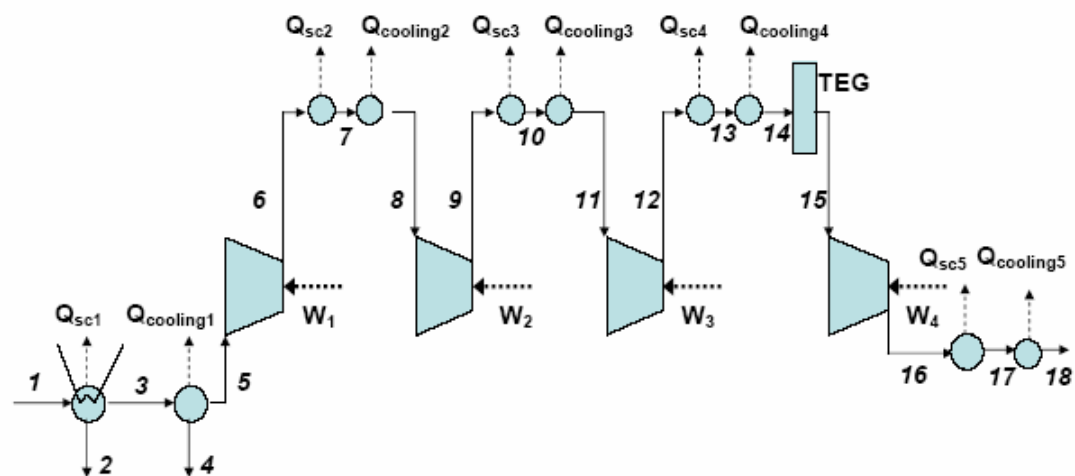


Fig.5

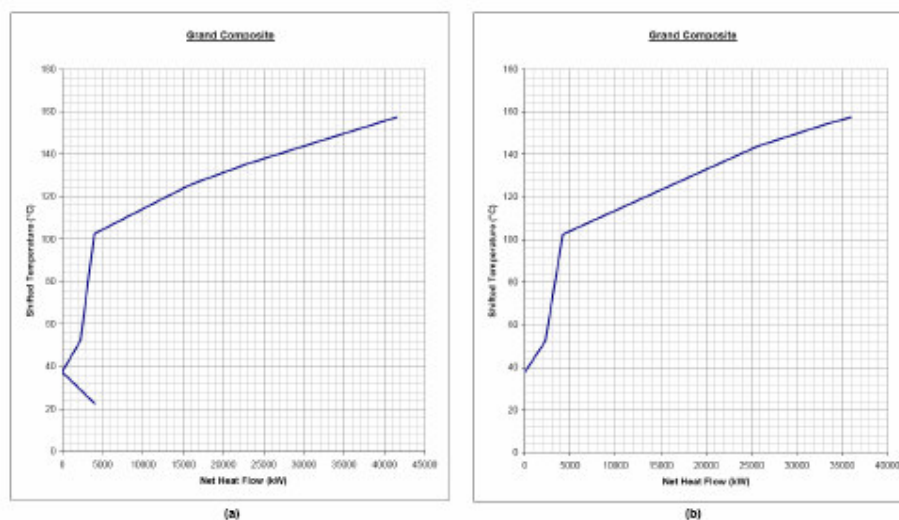


Fig.6

