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PART LOAD ANALYSIS OF A CONSTANT INVENTORY SUPERCRITICAL CO_2 POWER PLANT FOR WASTE HEAT RECOVERY IN CEMENT INDUSTRY

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ABSTRACT

The present work investigates the part-load performance of a MW-scale sCO_2 power plant designed as heat recovery unit for an existing cement plant located in Czech Republic, in the framework of the H2020 funded project CO2OLHEAT. The study firstly presents the selected power plant configuration and then focuses on the evaluation of its part load operation due to variation of flue gas mass flow rate and temperature. The range of flue gas conditions at the outlet of the upstream process is retrieved from a preliminary statistical analysis of historical trends obtained through the cement plant monitoring. The numerical model developed for this study aims at providing realistic results thanks to the adoption of turbomachinery performance maps provided by the project partners. Moreover, heat exchangers have been modelled through a discretized approach which has been validated against manufacturer data, while piping inventory and pressure losses have been assessed through a preliminary sizing that considers the actual distances to be covered in the cement plant. Performance decay is estimated for the whole range of flue gas conditions, reporting the most significant power cycle parameters, and identifying the main causes of efficiency loss. The part-load analysis is carried out considering a constant CO_2 inventory, in order to reduce the system complexity and capital cost and simplify plant operation. Results show that the operation entails minor variation of the compressors operative points in the whole range of operating conditions of the cement plant, avoiding the risk of anti-surge bypass activation. Moreover, the plant is able to work at nearly constant thermodynamic cycle efficiency (20.5%-23.0%) for most of the year and benefits from part-load operation in terms of overall performance. These predictions will be used, in next steps of the project, to guide the definition of power plant control during transients related to changes of upstream process conditions or specific needs of power output control.

INTRODUCTION AND SCOPE OF WORK

Waste heat-to-power is one of the main assets to improve the energy efficiency and reduce the footprint of the industrial sector [1][2]. Although the presence of commercial technologies (i.e. Organic Rankine Cycles - ORC for small-scale low-temperature [3] and steam power plants for larger and higher temperature applications [4]), a huge market potential is available for new concepts especially when waste heat is available at mid-to-high temperature [5][6], as for example in the cement production sector [7]. Supercritical carbon dioxide (sCO_2) power plants are widely recognized as a very promising technology for several applications based on solar energy [8][9], IVth generation nuclear reactors [10][11], fossil fuels [12][13] and also waste heat recovery [14], thanks to more compact and less expensive turbomachinery, higher flexibility than steam power plants [15] and higher performance with respect to ORCs [14].

This study focuses on the off-design simulation of a sCO_2 power plant designed as heat recovery unit for an existing cement plant in the framework of the H2020 funded project CO2OLHEAT [16]. The selected cement plant is located in Prachovice, Czech Republic, and its operation undergoes variable conditions depending on the load and the activation of raw mills. This implies a variation in the thermodynamic conditions of the hot flue gas available for the waste heat recovery process, eventually affecting the performance and the operation of the bottoming sCO_2 power unit. The knowledge of the expected trend for sCO_2 cycle main thermodynamic quantities and operating parameters is of fundamental importance for both finalizing the design of each component and for the definition of the control system ensuring a safe, reliable, and efficient operation of the power plant. The understanding of the power plant adaptation to a variation of a specified boundary condition requires a properly developed numerical tool and the

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knowledge of detailed information on the design of the main components, namely the turbomachinery and heat exchangers. This paper aims at providing such information, so to demonstrate the potential of sCO_2 power units in the cement production sector and to provide insights on general industrial waste heat recovery characterized by similar waste heat potential. The off-design analysis investigates both the effect of flue gas flow rate and temperature variations that have been retrieved from a preliminary statistical analysis of historical trends obtained from the cement plant monitoring. Mass flow rate of flue gas released from the cement plant are mainly affected by the number of active raw mills, while flue gas temperature simply varies in a narrow range between 400°C and 370°C. The most representative operating conditions of the selected cement plant are reported in Table 1. The nominal flue gas conditions are assumed for the cement plant running with no active raw mills: flue gas mass flow rate of 230’000 Nm^3/h (situation occurring approximately 5% of the year) and flue gas temperature equal to 400°C.

Table 1: Most representative cases of the cement plant operation (in brackets the fraction of time in which each condition occurs in a year).

	T_{FG}=400°C	**T_{FG}=370°C**
No raw mills in operation (5%) 230’000 Nm^3/h of FG available	Case A <u>(nominal)</u>	Case D
One raw mill in operation (85%) 165’000 Nm^3/h of FG available	Case B	Case E
Two raw mills in operation (10%) 100’000 Nm^3/h of FG available	Case C	Case F

NOMINAL POWER PLANT DESIGN

The nominal design of the power plant is based on the assumptions defined in the framework of the CO2OLHEAT project and agreed with all the consortium partners, including constraints and specifications of component manufacturers as Baker Hughes (BH), Siemens Energy (SIE), Bosal and Heatric. The plant is based on a simple recuperated cycle without neither recompression nor recuperator bypass, as reported in Figure 1: the choice is motivated by the relatively low maximum temperature of the heat source (i.e., the flue gas) and the high minimum stack temperature in order to avoid acid condenses (150°C). Minimum cycle thermodynamic conditions are set to 32°C and 85 bar to properly exploit the high density of CO_2 in the proximity of the critical point and improve thermodynamic cycle efficiency. The maximum cycle pressure at compressor outlet is set to 216.9 bar, which correspond to a turbine inlet pressure of around 210 bar, a value considered a good tradeoff between cycle performance, component manufacturability and techno-economic feasibility. Resulting cycle pressure ratio is equal to 2.55 and it is achieved with two centrifugal compressors in series that have been designed by BH. The compressors are installed on the same shaft and are mechanically driven by two centripetal turbines in series, thus creating a compact turbo-expander unity. To ensure a safe start-up of the system and to properly balance the power required by the compressors and the power delivered by BH turbine in all operating conditions, an electric motor (“helper”) is connected to the shaft, consuming 246 kW of electric power in nominal conditions. In this study a direct flue gas-pressurized CO_2 primary heat exchanger has been preferred to the use of an heat transfer fluid (HTF) loop based on diathermic oil. Goal is to maximize turbine inlet temperature, reduce the complexity of the system and avoid a large inventory of flammable liquid on site eventually leading to a possible reduction of capital and operational cost related to additional equipment, piping and fire protection. On the other hand, the use of a HTF loop would be preferrable in case of lack of space close to the upstream process or in case of discontinuous process where the use of thermal storage allows to decouple waste heat recovery and power production. According to the use of direct heat introduction, maximum temperature of the cycle is set at 360°C, namely 40°C lower than the flue gas nominal temperature in order to design with a reasonable heat transfer area. The expansion is then completed in a power turbine, designed by SIE, that exploits the residual pressure ratio and produces the electrical power output by means of an electrical generator. Power turbine operation can be controlled with an admission valve that involves a pressure drop of 0.3 bar in wide open position. A CO_2 mass flow rate of 1 kg/s is extracted downstream of the compressors and reintroduced in the power cycle to compensate for Dry Gas Seals (DGS) leakages in the turbines (0.5 kg/s for mechanical drive turbine and 0.4 kg/s for power turbine) and injected in minor amount upstream of the cooler (0.1 kg/s). Turbomachinery efficiencies are assumed equal to the preliminary nominal values provided by manufacturers: 73% for the compressors, 84% for the mechanical-drive turbines, and 82% for the power turbine. An additional efficiency loss equal to 5% is accounted for mechanical and electrical losses in both shafts.

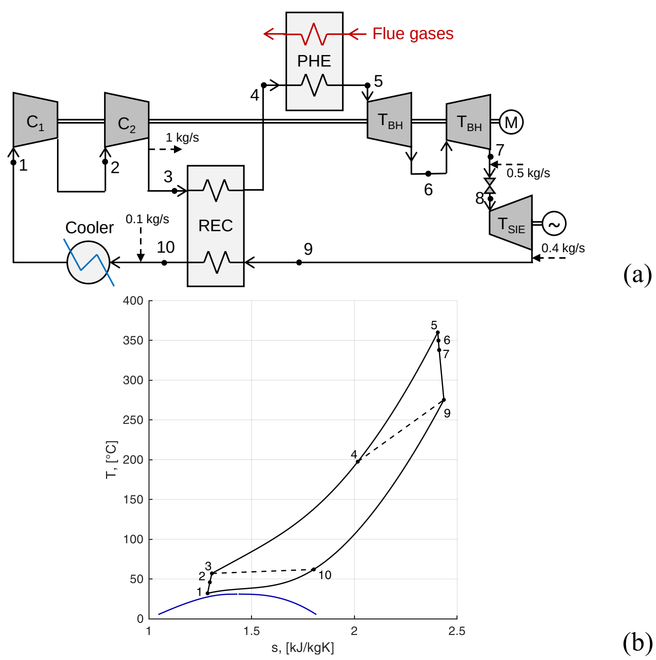

Figure 1: Layout (a) and T-s (temperature-specific entropy) diagram (b) of the sCO_2 power cycle.

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The heat transfer surfaces and internal volumes of the different heat exchangers are calculated by matching the assumed pressure drops by means of numerical routines proprietary of Politecnico di Milano, mostly based on previous experience from the H2020 sCO2-Flex project [17].

The Primary Heat Exchanger (PHE) consists of a finned tube HX with direct heat transfer between flue gas and the CO_2, modelled through the same methodology presented in [14]. Supercritical carbon dioxide flows inside the tubes while the flue gas stream flows across the finned tubes bundles. The recuperator is designed as a printed circuit heat exchanger (PCHE) with a pinch point temperature difference of 5°C and pressure losses on the low pressure (LP) side and high pressure (HP) side of 1.25 bar and 0.75 bar respectively, as suggested by the consortium partner Heatric, responsible of the recuperator design. The cooler is designed as several dry air-cooled heat exchangers bays arranged in parallel: each single unit is made up of batteries consisting of different rows of small diameter aluminum finned copper tubes through which air is blown or sucked by fans. The CO_2 side pressure drop in the component is estimated to be equal to 4 bar.

Piping length and diameter have been determined by consortium partner Simerom through a preliminary analysis of the distances to be covered in the cement plant. This data is particularly useful for an accurate evaluation of the pressure losses of the sCO_2 power cycle as well as for the estimation of the CO_2 inventory held within the system, equal to 1551.7 kg. The resulting net power output is 2.17 MW, with a cycle efficiency of 23.17% referred to the inlet thermal power and 11% if referred to the maximum power available from flue gas cooling down to 150°C, a limit generally imposed to avoid the condensation of acid compounds.

OFF DESIGN SIMULATION NUMERICAL APPROACH

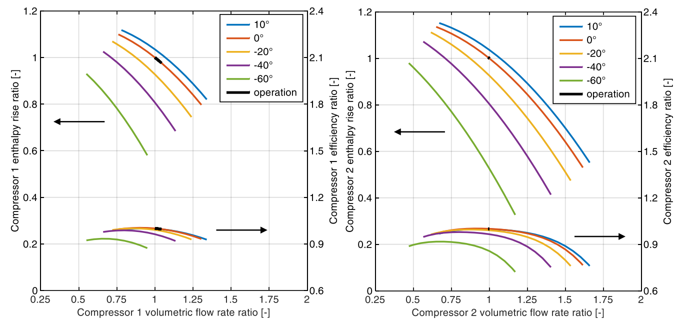
In this work the use of a CO_2 tank/vessel for active inventory change is not implemented in order to maintain the control strategy of the plant as simple as possible and to reduce the system capital cost. The compressors are operated at fully open (0°) Inlet Guide Vanes (IGV) and their efficiency is calculated based on the operating maps provided in Figure 2. During off-design operation the first compressor inlet temperature is maintained equal to the nominal value (32°C) by regulating the cooling air mass flow rate in the HRU, while cycle minimum pressure is not controlled and varies according to the constant inventory operation. Mechanical drive turbines work in sliding pressure operation: their isentropic efficiency and reduced mass flow rate (see Equation (1)) are characterized with the correlations reported in Figure 3.a as function of the ratio u/c between the peripheral speed u and the spouting velocity c, defined according to Equation (2). The power turbine operation is computed through the same methodology but using the turbine pressure ratio as the input parameter (see Figure 3.b). The admission valve at power turbine inlet is not employed in steady-state operation in order to maximize cycle performance and CO_2 leakages mass flow rates due to DGS are considered constant and equal to design values for all off-design operation.

Finally, the heat exchangers in off-design conditions are simulated computing the heat transfer coefficients for both the CO_2 and the flue gas while the HXs pressure drops are updated with the simplified correlation reported in Equation (3).

$$m_{red,turb} = \left.\frac{\dot{m}\sqrt{T}}{p}\right|_{in,turb} \tag{1}$$

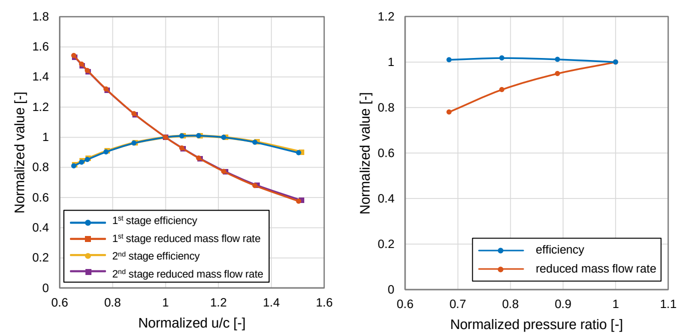
$$c = \sqrt{2 \cdot \Delta h_{is,turb}} \tag{2}$$

$$\Delta p = \Delta p_{nom}\left(\frac{\rho_{nom}}{\rho}\right)\left(\frac{\dot{m}}{\dot{m}_{nom}}\right)^2 \tag{3}$$



(a) (b)

Figure 2: First (a) and second (b) compressor operating maps (normalized enthalpy rise and efficiency) as function of the normalized volumetric flow rate.



(a) (b)

Figure 3: Baker Hughes (a) and Siemens (b) turbine operating curves (normalized reduced mass flow rate and efficiency) as function of the normalized ratio (u/c) and turbine pressure ratio, respectively.

OFF DESIGN RESULTS

The off-design analysis investigates both the effect of flue gas flow rate and temperature deviation from nominal conditions. In particular, in the numerical simulations these two parameters are varied in the following ranges as suggested by the statistical analysis of historical data:

- Flue gas flow rate is varied from the nominal value (230’000 Nm^3/h, no raw mills in operation) to 40% of the nominal value, corresponding to 92’000 Nm^3/h.
- Flue gas temperature is varied from the nominal value, equal to 400°C, to 370°C.

As the combined effect of these two flue gas condition variations is investigated, the results of the off-design analysis

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are presented as contour maps displaying how the main parameters and figures of merit of the sCO_2 power cycle vary in off-design operation. Cases A to F (see Table 1) are reported with markers and letters on the displayed maps.

Heat recovery from flue gas

The flue gas temperature at PHE outlet is not controlled and tend to decrease during part-load operation (see Figure 4.a) due to the fact that the PHE surface results oversized in part-load conditions. For this reason, the thermal power input to the cycle decreases less than the flue gas mass flow rate for the same heat source temperature, as reported in Figure 4.b. For example, Case C, characterized by nominal flue gas temperature equal to 400°C and a reduction of 56% of the mass flow rate, implies a decrease of heat input to the cycle of only ~14%. During part load operation the plant tends to exploit a larger fraction of the thermal power available from the exhausts, whose maximum amount can be calculated considering a minimum stack temperature of the exhausts $T_{FG,min}$ equal to 150°C. As a result, reducing the flue gas mass flow rate allows increasing the heat recovery factor χ_{rec} (see Equation (4) and Figure 4.c) from the nominal value of 0.47 to a value close to 1 for case F.

$$\chi_{rec} = \frac{\dot{Q}_{in,cycle}}{\dot{Q}_{FG,max}} = \frac{\dot{m}_{FG} c_{p,FG} (T_{FG,max} - T_{stack})}{\dot{m}_{FG} c_{p,FG} (T_{FG,max} - T_{FG,min})} \tag{4}$$

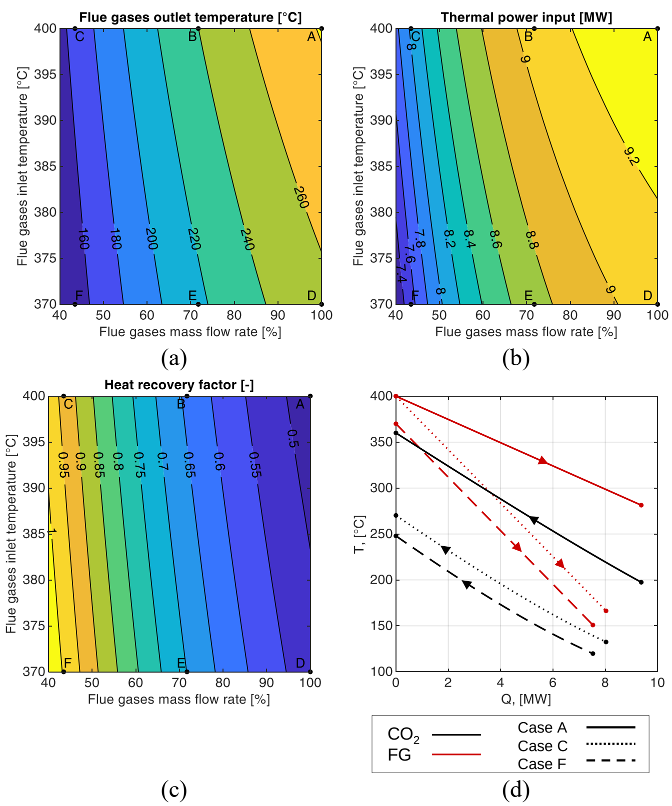

Figure 4: Flue gas temperature at PHE outlet (a), thermal power transferred in the PHE (b) and heat recovery factor as function of off-design flue gas conditions. PHE T-Q diagram for cases A, C, F (d).

This aspect is also clearly visible from the T-Q diagrams (temperature – thermal power) of the PHE which is depicted for case A (nominal), case C (minimum FG flow rate) and F (minimum FG flow rate and minimum temperature) in Figure 4.d. Cases with minimum FG flow rate are characterized by smaller duty but a lower FG minimum temperature and thus a larger heat recovery factor. Moreover, it is also noticeable how the CO_2 temperature at the outlet of PHE (i.e., the maximum cycle temperature) tends to decrease for low flue gas mass flow rates (cases C and F in Figure 4.d).

Power plant operating conditions

The CO_2 temperature at the outlet of PHE (first turbine inlet temperature) and the CO_2 mass flow rate processed in the power cycle are reported in Figure 5.a and Figure 5.b, respectively. While maximum cycle temperature decreases rapidly when FG mass flow rate reduces, the CO_2 mass flow rate in the power cycle tends to remain fairly constant, as it is proportional to the slope of the CO_2 profile in the T-Q diagram, which remains similar. Its value passes from 45.5 kg/s of the nominal conditions to a value equal to 40.4 kg/s in case F (-11.2%).

Figure 5.c and Figure 5.d depict the cycle maximum and minimum pressure as function of the flue gas conditions. It is possible to notice that, as the flue gas mass flow rate and temperature decrease, both pressure levels decrease as a result of the sliding pressure operation of the turbines and the strong reduction of average CO_2 temperature in the PHE.

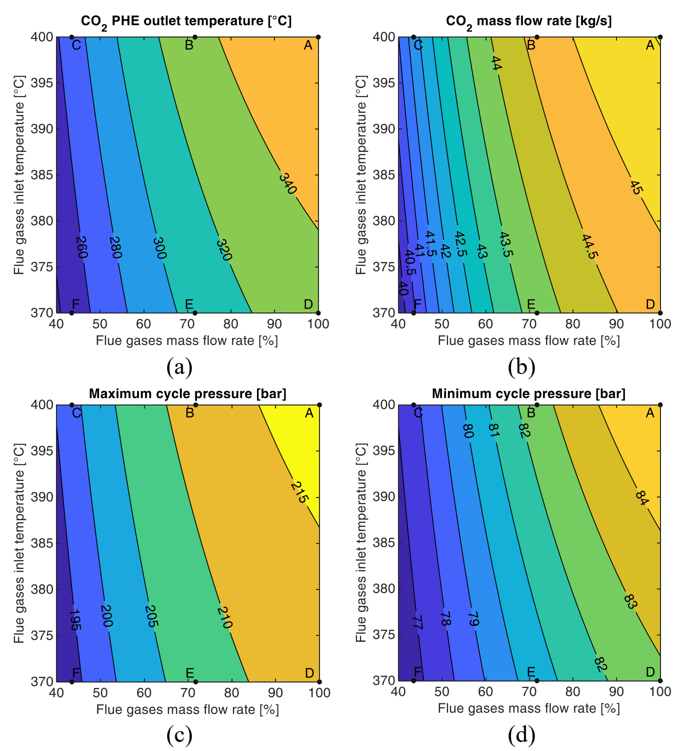

Figure 5: CO_2 temperature at PHE outlet (a), CO_2 mass flow rate processed by the cycle (b), cycle maximum (c) and minimum pressure (d) as function of the off-design flue gas conditions.

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The cycle maximum pressure passes from a nominal value of 216.9 bar to a value of 193.2 bar for case F while, for the same case, the minimum pressure decreases from 85 bar to 76.7 bar. On the other hand, the cycle pressure ratio variation in the whole off-design operation is limited, with a maximum variation range restricted to -2.3/+0.9%.

Turbines, generator and electrical helper operation

Figure 6.a depicts the power required by the electric helper balancing the turbo-expander shaft. The electric consumption increases from 246 kW (Case A nominal condition) to 330 kW (Case F), mainly due to the decrease of the maximum cycle temperature (i.e., the first turbine inlet temperature) at nearly constant cycle pressure ratio.

Maximum cycle temperature reduction leads to a consequent decrease of CO_2 temperature at power turbine inlet (Figure 6.c) and also a reduction of its specific work as it is possible to notice from Figure 6.d, where power turbine isentropic enthalpy drop varies from a design point condition of 69.0 kJ/kg to a value of 47.8 kJ/kg (-31%) in case F. As a consequence, although the CO_2 mass flow rate is little affected, the SIE turbine power output appreciably decreases from more than 2.4 MW in the design condition down to slightly less than 1.6 MW in condition F (see Figure 6.b).

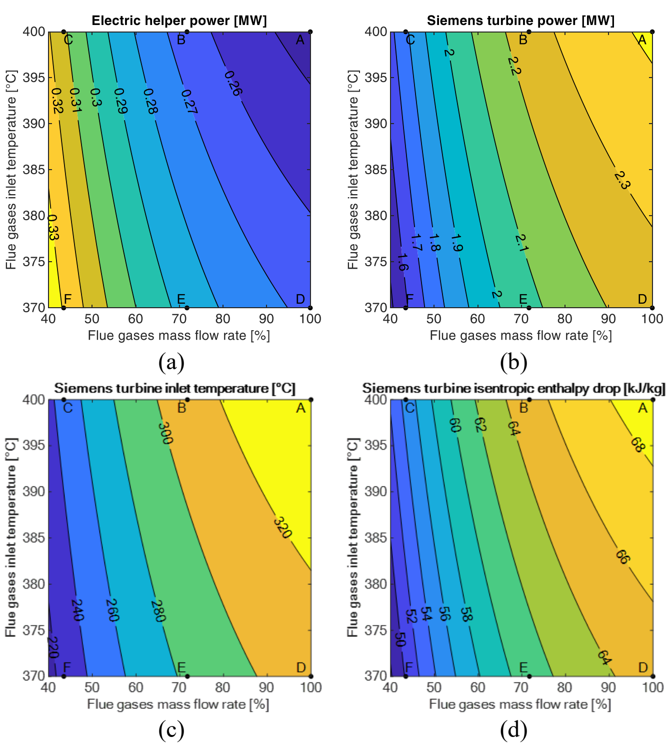

Figure 6: Electric helper power consumption (a), Siemens turbine power (b), Siemens turbine inlet temperature (c) and isentropic enthalpy drop (d) as function of the off-design flue gas conditions.

Compressors operation

Considering the whole range of off-design operation of the plant, while the first compressor operating point deviates only slightly from the nominal conditions, for the second compressor the variation is almost negligible, as noticeable from Figure 2.a and Figure 2.b, respectively.

This aspect is due to the almost constant volumetric flow rate at both compressors inlet, due to the combined effect of the slight decrease of both the CO_2 mass flow rate and the cycle minimum pressure, which cause a consequent reduction of the inlet density to the compressors (Figure 7.a and Figure 7.b). As a consequence, the efficiency variation during off-design operation of these two components is very limited.

Net power output and cycle performance

In part load operation cycle efficiency (Figure 8.b) decreases from a nominal value of 23.2% (case A) to a minimum value of 16.7% (case F), corresponding to -28% in relative terms. However, the performance decay is not constant across the operation range and there is a wide span of conditions where the performance remains close to the nominal one. In particular, considering the actual cement plant operation, the sCO_2 power system can be operated with a conversion efficiency above 20% for most of the year, as the cases B and E, which represent 85% of the yearly operation, feature a cycle efficiency of 21.9% and 20.5%, respectively. On the other hand, in these two conditions the waste heat recovery plant can achieve a lower net power output, ranging from 1.79 MW to 1.99 MW.

Considering the whole range of the cement plant operation, the net power output (Figure 8.a) decreases from a nominal value of 2.17 MW to 1.26 MW (-41.8%) of electricity generated in the most penalizing condition (case F) as a consequence of the consumption increase of electric helper installed on the turbo-expander shaft as well as of the decrease of SIE power turbine electric output. Nevertheless, this result can be considered encouraging as it is obtained with both a reduction of the available flue gas mass flow rate, equal to -56%, and a decrease of 30°C of their maximum temperature.

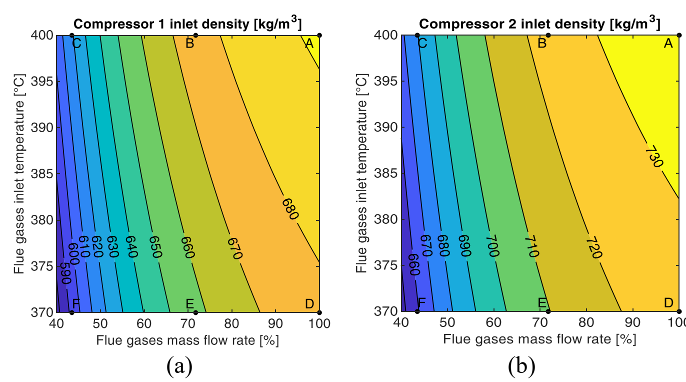

Figure 7: First (a) and second (b) compressor CO_2 inlet density as function of the off-design flue gas conditions.

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Actually, it is possible to notice that the overall plant efficiency (Figure 8.c), defined as the product of the cycle efficiency and the heat recovery factor, increases in part load operation, and presents a maximum close to case C operating conditions (around 17.1% vs. 11% in nominal conditions, equal to +55.5% in relative terms), thanks to the increase of heat source exploitation. In such condition the flue gases have a stack temperature very close to the temperature limit to avoid acid condenses (150°C).Table 2 reports a summary of the main results for the most representative cases.

CONCLUSIONS

This paper presents the numerical assessment of the part-load performance of a sCO_2 power plant for a waste heat recovery application in a cement plant, in the frame of the H2020 funded project CO2OLHEAT. This study demonstrates the possibility to operate waste heat recovery unit based on sCO_2 power cycles at constant CO_2 inventory. This solution not only allows to decrease the installation cost, but also largely simplifies the system operation at part-load and the power plant control system. Numerical results show how the pressure ratio and CO_2 mass flow rate remain relatively constant in the whole range of the cement plant operation, allowing to limit turbomachinery off-design performance decay. As a result, even operating the compressors with fixed IGV aperture, their operative points remain very close to nominal conditions, limiting the issues related to loss of performance and anti-surge bypass activation. This brings to the possibility to operate the plant for most of the year (90%) with an efficiency close to the nominal one.

Furthermore, in spite of a reduction of cycle conversion efficiency at part load operation, the overall plant efficiency actually increases as the sCO_2 power cycle tends to exploit a larger fraction of the thermal power available from the exhausts.

For example, by reducing by about 50% the flue gas mass flow rate at the nominal temperature, even if the cycle efficiency decreases by approximately 5 points (from the nominal value of 23.2% to 18.3%), the heat recovery factor almost doubles (from 47.5% to 93.5%), thus resulting in an overall plant efficiency increase of more than 50 % in relative terms (passing from 11.0% to 17.1%). These results will provide useful insights in the next steps of the CO2OLHEAT project, in particular to guide the definition of power plant control system.

A techno-economic analysis of the plant will represent the following step of this work.

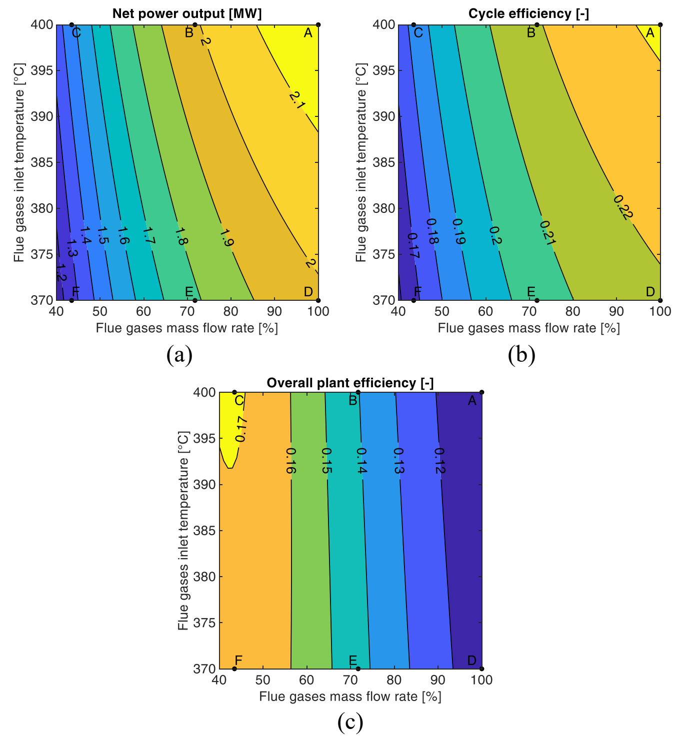

Figure 8: Net power output of the plant (a), cycle efficiency (b) and overall conversion efficiency (c) as function of the off-design flue gas conditions.

Table 2: Summary of the main results for the most representative cases

	A	B	C	D	E	F
sCO_2 thermodynamic cycle						
CO_2 mass flow rate [kg/s]	45.52	44.66	41.74	44.76	43.71	40.42
Maximum pressure [bar]	85.00	82.57	77.72	82.80	80.50	76.71
Minimum pressure [bar]	216.90	211.99	198.41	212.48	207.18	193.24
CO2 temperature at PHE outlet [°C]	360.0	333.1	270.2	331.4	305.7	247.9
Heat and power balance						
Flue gas thermal power [MW]	19.76	14.17	8.59	17.39	12.47	7.56
Cycle thermal power input [MW]	9.38	9.07	8.03	9.08	8.72	7.53
Compressor 1 power [MW]	0.574	0.562	0.519	0.563	0.548	0.500
Compressor 2 power [MW]	0.547	0.537	0.503	0.538	0.526	0.488
BH turbine power [MW]	0.875	0.834	0.711	0.834	0.788	0.658
Siemens turbine power [MW]	2.42	2.25	1.78	2.25	2.07	1.59
Electric helper power [MW]	0.246	0.265	0.312	0.268	0.286	0.329
Net power output [MW]	2.17	1.99	1.47	1.98	1.79	1.26
Heat and power balance						
Cycle efficiency [%]	23.17	21.91	18.30	21.81	20.47	16.69
Heat recovery factor [%]	47.46	64.03	93.48	52.23	69.93	99.59
Overall plant efficiency [%]	11.00	14.03	17.11	11.39	14.31	16.63

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NOMENCLATURE

List of abbreviations

DGS – Dry Gas Seals
EU – European Union
FG – Flue gas
HP – High Pressure
HTF – Heat Transfer Fluid
HX or HE – Heat Exchanger
IGV – Inlet Guide Vanes
LP – Low Pressure
PCHE – Printed Circuit Heat Exchanger
PHE – Primary Heat Exchanger
sCO2 – supercritical CO2
WH2P – Waste Heat to Power
WP – Work Package

List of symbols

c_p – Specific heat capacity [kJ/kgK]
h – Specific enthalpy [kJ/kg]
$\dot{m}$ – Mass flow rate [kg/s]
$m_{red,turb}$ – Turbine reduced mass flow rate
p – Pressure [bar]
Q or $\dot{Q}$ – Thermal Power [MW]
s – Entropy [kJ/kgK]
T – Temperature [°C]
v – Velocity [m/s]
$\dot{V}$ - Volumetric flow rate [m^3/h]
ρ – Density [kg/m^3]
χ_{rec} – Heat recovery factor [-]

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