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EFFECT OF THE AMBIENT TEMPERATURE ON THE PERFORMANCE OF SMALL SIZE sCO_2 BASED PULVERIZED COAL POWER PLANTS

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ABSTRACT

The present work focuses on the analysis of a novel coal fired sCO_2 power plant concept developed in the frame of sCO2-Flex H2020 EU funded project. Fossil fuel fired power plants are expected to improve their flexibility in the future energy scenario characterized by a large share of non-predictable and non-dispatchable renewable energy sources. This upcoming context requires a new generation of coal fired power plants with a smaller size, a high flexibility and minor requirements for the installation site like no need of water consumption. Carbon dioxide in supercritical cycles is recognized to be a possible solution for this technology shift and could replace in the future common steam Rankine cycles. This paper focuses on the impact of ambient temperature variation on a small size coal fired sCO_2 power plants equipped with a dry cooling heat rejection unit, with the aim of understanding the effect on plant operability and system performance. A dedicate tool is implemented for off-design behavior assessment and different control strategies are investigated. Results show that without a proper design of the heat rejection unit a small increase of ambient temperature may drastically limit the maximum attainable power output of the plant. This penalizing effect is more pronounced in hot locations, but this issue can be limited by adopting a sufficient over-sizing of the cycle heat rejection unit (HRU) or wet-and-dry solutions.

INTRODUCTION AND SCOPE OF WORK

In near future, the growing share of non-dispatchable renewable energy sources in the electricity mix [1] and the lack of economically viable large-scale electricity storage [2] will involve a drastic change of existing and new fossil fuel power plants operation. Coal power plants will gradually shift their role from base-load operation to cover peak demand and to provide energy services to the electrical grid. However, current coal power plants are not designed for part-load operation nor adequate to undergo rapid power output fluctuations, which are necessary characteristics to meet short noticed load variations caused by unpredictable renewable energy sources. The founding idea of the H2020 sCO_2-Flex project [3] is to improve the flexibility of pulverized coal power plants by adopting smaller modular plants based on sCO_2 Brayton cycles instead of large plants based on conventional steam Rankine cycles. The advantage of adopting sCO_2 as working fluid is represented by a more compact cycle equipment and in particular the possibility to design low number of stage, small diameter turbomachinery that can allow faster start up and ramp-up/ramp-down transients. Another advantage is represented by the very low minimum load which is around 25%, a value remarkably higher than ultra-supercritical (USC) power plants (40-50%) and the high performance in part load as confirmed by numerical research on both fossil fuel based [4][5][6][7], concentrating solar power [8][9][10] and waste heat recovery power plants [11][12][13]. A final positive aspect related to the adoption of sCO_2 power plants is represented by the high working fluid temperature variation in the HRU (around 35°C) which allows to adopt air-cooled units with small cold end temperature difference without involving an excessive footprint or cooling air mass flow rate. This peculiarity allows a significant reduction of water consumption and an easy installation of such systems independently of the availability of a river or sea in proximity of the site location. However, differently from water-cooled HRU that can benefit from a relatively stable minimum temperature of the cooling medium, for dry-cooled units the ambient temperature variation on daily and seasonal base can affect the cycle minimum temperature with a consequent impact on sCO_2 power plant performance and operability. This aspect has been scarcely investigated by the scientific literature so far, generally just focusing on the techno-economic consequences related to HRU design assumptions and only considering the nominal operation of the plant [14][15][16][17].

Pidaparti et al. highlighted the techno-economic potential of dry air cooling for the sCO_2 cycles limiting their analysis to the nominal design of the plant [18].

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Chen et al. [19] studied the effect of different CO_2 inlet condition, including also the variation of CO_2 inlet temperature, on the compressor performances. However, the authors limited their study to component level, not considering how the overall system performances would be affected.
The scope of this paper is to discuss the effects related to the ambient temperature variation for the coal fired sCO_2 power plant designed in the frame of sCO_2-Flex H2020 EU project and to quantify the loss of efficiency and the limitation of attainable power output caused by a progressive departure of compressor inlet temperature from the nominal conditions. Finally, two possible solutions to this issue are proposed: the first requires an overdesign of the heat rejection unit gas cooler, while the second implies the adoption of wet-and-dry solutions.

SCO_2-FLEX PLANT NOMINAL DESIGN

Figure 1 depicts the cycle configuration selected for the 25 MW_{el} sCO_2-Flex plant, a recuperative recompressed cycle provided by High Temperature Recuperator (HTR) bypass [20].
The methodology for the thermodynamic optimization of the system and the study of its part load behavior have already been discussed in several works [21][22][23][24] and are reported in [4]. Table 1 reports the main assumptions while Table 2 reports the optimal design parameters (marked with *) plus some relevant optimized quantities. In order to maximize the boiler efficiency, stack temperature is pushed down to its lower bound value (130°C), while the air combustion temperature is increased in a Ljungström heat exchanger with final values equal to the upper bound value (350°C).
This involves that the pinch point temperature differences at the HT-PHE outlet section is higher than the minimum achievable technical value as also shown in Figure 2 that depicts the optimal boiler T-Q diagram. Table 3 reports the system power balance: around 35% of turbine electric power output is consumed by compressor motors, the obtained net cycle efficiency is 42.36%, while the boiler efficiency is 94.37% for an overall plant efficiency of 39.98%.

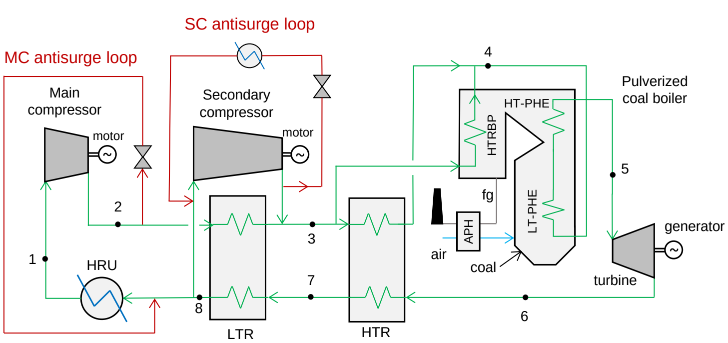

Figure 1. sCO_2-Flex plant layout. In green is represented the main CO_2 flow, while in red the antisurge loops of the compressors.

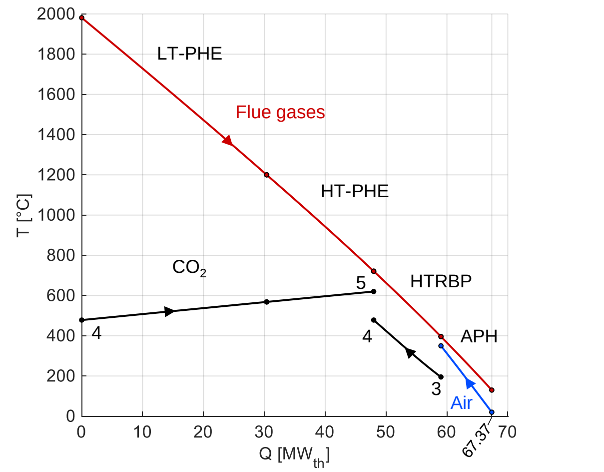

Figure 2. Boiler T-Q diagram (temperature vs. thermal power).

Table 1. Main assumptions for sCO_2 power cycle design.

Main assumption	
Plant design net electric power, MW_{el}	25
Maximum cycle temperature, °C	620
Minimum cycle temperature, °C	33
Turbine isentropic efficiency,	85.4 %
Main compressor polytropic efficiency	82.4 %
Second. compressor polytropic efficiency	81.8 %
Generator mech.-electrical efficiency	96.4 %
Motor mech.-electrical efficiency	98.4 %
LTR and HTR pinch point, °C	10
Boiler CO_2 Δp, bar	3.75
HRU CO_2 $(\Delta p/p_{in})$	0.5%
Recuperators hot side $(\Delta p/p_{in})$	0.5%
Nominal cooling air temperature, °C	20
Cooling air temperature increase, °C	20

Table 2. Main results for sCO_2 power cycle and coal boiler

Boiler and cycle optimized results	
CO_2 mass flow at turbine inlet, kg/s	269.19
CO_2 mass flow at HRU, kg/s	173.49
CO_2 mass flow at HTR bypass, kg/s	30.17
Split ratio SR (*)	0.64
Bypass ratio BR (*)	0.11
Maximum cycle pressure, bar (*)	250
Minimum cycle pressure, bar (*)	79.78
Coal mass flow rate, kg/s	3.70
Combustion air mass flow rate, kg/s	24.56
Flue gases mass flow rate, kg/s	27.92
Adiabatic flame temperature, °C	1980.96
Flue gases stack temperature, °C	130
Optimal boiler pinch point, °C (*)	101.05
Optimal Ljungström pinch point, °C (*)	45.93
HRU cooling air mass flow rate, kg/s	1596.60

Table 3. sCO_2-Flex plant power balance and performance

System Power Balance	
Turbine electric power, MW_{el}	38.82
Main compressor electric power, MW_{el}	5.33
Secondary compressor electric power, MW_{el}	8.22
Heat rejection auxiliaries consumption, MW_{el}	0.273
Electrical and mechanical losses, MW_{el}	1.65
Q LHV coal, MW_{th}	62.53
Q HRU, MW_{th}	32.09
Q stack, MW_{th}	3.29
Cycle efficiency, %	**42.36**
Boiler efficiency, %	**94.37**
Overall efficiency, %	**39.98**

All the carbon dioxide streams thermodynamic properties of the optimized sCO_2 power cycle are reported in Table 4 while the T-s diagram and the cycle T-Q diagrams (recuperators and HRU) are reported in Figure 3 and Figure 4 respectively. Ad hoc numerical routines are implemented for the calculation of heat transfer area of each heat exchanger in the plant and validated against the data provided by sCO_2-Flex consortium partners. Table 5 reports the main results of the heat exchangers design showing how the two recuperators require a large overall heat exchange area and metal mass due to the small $\Delta T_{pp,LTR}$ and $\Delta T_{pp,HTR}$ selected. Finally, Table 6 reports the breakdown of plant capital cost computed adopting for each component a cost correlation from literature specially developed for sCO_2 power plants. More information about the validity range and the uncertainty of the cost correlations can be found in [25]. Results show a total investment cost of 52.23 million dollars for the 25 MW_{el} plant and a final specific cost which is around 2089 $/$kW_{el}$. As coal-fired systems of such a small scale currently are not a commercial solution, it is not possible to provide a direct comparison of the specific plant cost. However, this figure seems to be competitive with other fuel combustion-based technologies as conventional biomass plants integrating a stoker boiler and a steam turbine present a plant specific costs in the order of 2000-4000 $/$kW_{el}$ [26].

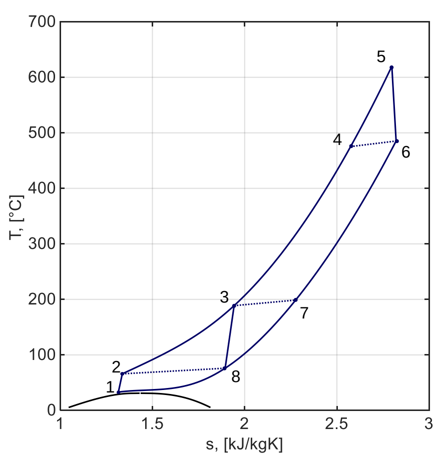

Figure 3. T-s diagram of the optimal cycle.

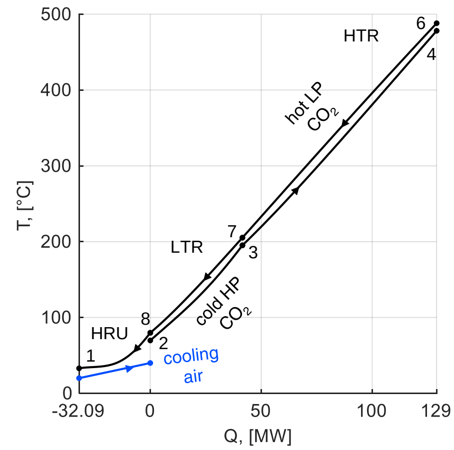

Figure 4. T-Q diagram of the optimal cycle.

Table 4. Thermodynamic streams of the sCO_2 cycle.

Point	**T** (°C)	**p** (bar)	**ρ** (kg/m^3)	**h** (kJ/kg)	**s** (kJ/kgK)
1	33.00	79.78	609.40	306.29	1.34
2	69.73	250.00	738.30	336.50	1.36
3	195.19	249.95	332.83	575.77	1.96
4	478.13	249.85	170.18	941.94	2.58
5	620.00	246.10	138.70	1120.13	2.80
6	488.13	80.99	55.93	970.61	2.83
7	205.30	80.58	95.45	645.47	2.30
8	79.83	80.18	161.03	491.27	1.93

METHODOLOGY AND RESULTS

The impact of ambient temperature variation on the performance and the maximum attainable power output of the present pulverized coal sCO_2 power plant is investigated with a set of control strategies which mainly aim to guarantee the operability of the system while limiting the efficiency penalization. If no corrective actions are implemented, the increase of ambient temperature causes an increase of compressor inlet temperature with a progressive departure from the critical point region, resulting in a marked drop of density of the working fluid. The loss of real gas effects at compressor inlet leads to a higher main compressor specific consumption (for a given pressure ratio) and a rapid increase of the CO_2 volumetric flow rate (for a fixed coal mass flow rate at boiler burners). The first effect penalizes the cycle thermodynamic efficiency while the second one may significantly limit the operability of the system in part-load and off-design conditions.

Compressors off-design performances are evaluated through performance maps derived by the Baker Hughes General Electric manufacturer data. For each point characterized by a volumetric flow rate and an enthalpy rise, the shaft speed and the variable inlet guide vanes (VIGV) position have been optimized in order to maximize the component polytropic efficiency.

The normalized performance maps for the main and secondary compressor are reported in Figure 5.top and Figure 5.bottom respectively, showing the optimal polytropic efficiency of the component as function of its inlet volumetric flow rate ratio and enthalpy rise.

To ensure compressors operability and lifetime it is important to avoid the surge phenomenon. For this reason, if the required volumetric flow rate is lower than 1.1 times the compressor surge limit value, the anti-surge bypass valve is opened to guarantee stable compressor operation at the expenses of an increase in power consumption and thus penalizing system efficiency.

Considering the map of the main compressor (Figure 5.top), it is possible to notice that it is sufficient to increase the volumetric flow rate by 20% from the nominal point to reach the upper limit of the operative map and a similar limit stands also for the secondary compressor (Figure 5.bottom). Figure 6 depicts the ratio between the density of CO_2 and the density at compressor inlet point in nominal conditions varying compressor inlet temperature and pressure. It is possible to highlight that the operability of the main compressor (-20% in density) can be strongly limited by just an increase of inlet temperature equal to 1°C when cycle minimum pressure is kept unchanged (Figure 6.left) while an increase of around 30°C is required to the secondary compressor to reach the map limit at nominal pressure (Figure 6.right).

Table 5. sCO_2-Flex heat exchangers size and thermal duty.

Heat exchanger size				
	Duty, MW_{th}	ΔT_{mln}, °C	HX area, m^2	Metal mass, kg
HRU	32.09	15.05	2182.22	17217
LTR	41.51	12.56	4644.03	29120
HTR	87.52	14.69	11511.97	72185
HTRBP	11.05	227.39	504.12	30686
PHE	47.97	369.17	912.98	45330

Table 6. Breakdown of sCO_2-Flex plant capital cost.

Economic analysis		
HRU, M$	2.09	3.99%
Main Compressor, M$	2.38	4.56%
Secondary Compressor, M$	2.83	5.42%
Turbine, M$	2.20	4.21%
LTR, M$	4.15	7.95%
HTR, M$	6.50	12.45%
Boiler, M$	21.49	41.14%
Compressors motors, M$	1.38	2.65%
Gearbox, M$	0.44	0.83%
Generator, M$	0.80	1.54%
Contingency, M$	3.10	5.93%
Engineering, M$	4.87	9.32%
Total capital cost, M$	52.23	
Plant specific cost, $/kWel	2089	

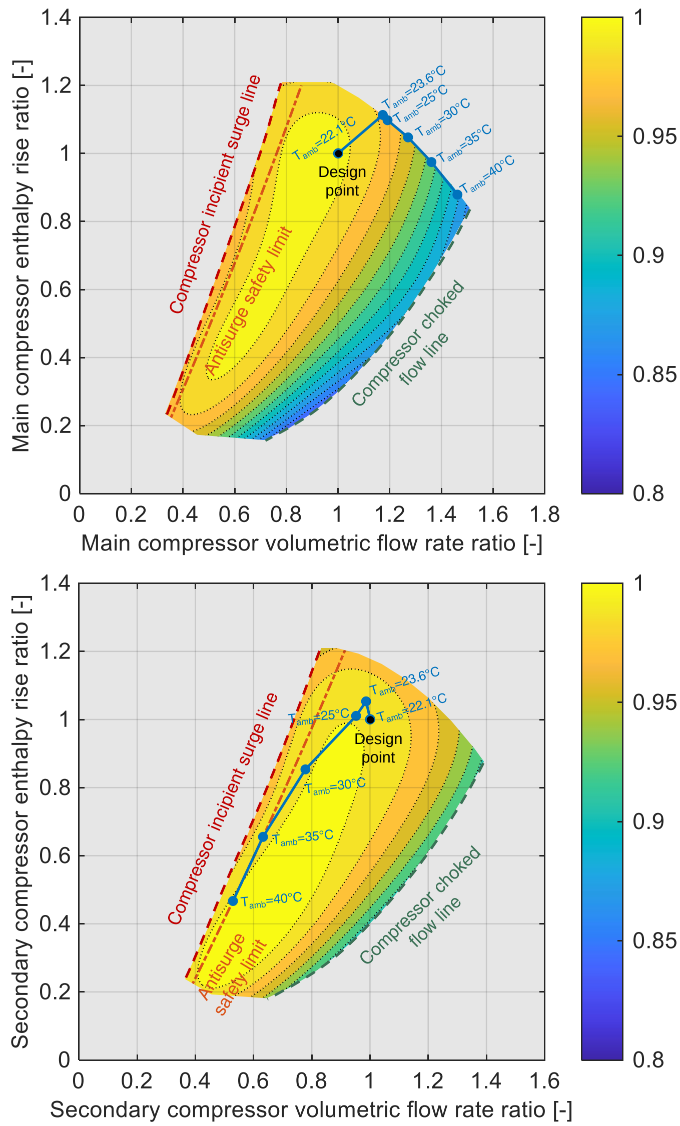

Figure 5. Normalized operational map of the main compressor (top) and secondary compressor (bottom).

An increase of compressor inlet pressure can mitigate this effect allowing to keep the volumetric inlet flow rate equal to the nominal one. However, for an inlet temperature increase of only 2°C the corresponding compressor inlet pressure should increase by 5.2 bar (which is 6% of nominal minimum pressure) causing a reduction of plant pressure ratio and a consequent loss of efficiency. Considering a system cooled by a dry air heat rejection unit, it is clear that a temperature departure from nominal point higher than the overmentioned limit can be easily obtained most of the year involving the need of facing this issue for closed Brayton supercritical sCO_2 cycles designed with compressor inlet condition close to the critical point.

Therefore, in these plants it is important to implement some strategies in order to preserve the operability of the plant also for higher ambient temperature variations.

The same problem is faced also for conventional open-cycle gas turbines where the mitigation of performance decay at high ambient temperature is commonly pursued by adopting a water-cooled battery fin heat exchanger at compressor inlet or by ambient air humidification with demineralized water spraying. Differently, steam Rankine cycles performances are less affected by ambient temperature variations since these cycles are usually condensed by means of a water-cooled unit.
Figure 7 to Figure 10 depict the trend of different quantities as function of ambient temperature. Figure 7 depicts the plant gross power output, the HRU fan consumption and the plant net power output, Figure 8 depicts the coal mass flow rate, the cycle pressure ratio and the compressor inlet temperature, Figure 9 depicts the trend of cycle thermodynamic efficiency, boiler efficiency and overall plant efficiency, Figure 10 shows the variation of enthalpy variation in turbine, main compressor, secondary compressor plus the specific power output.

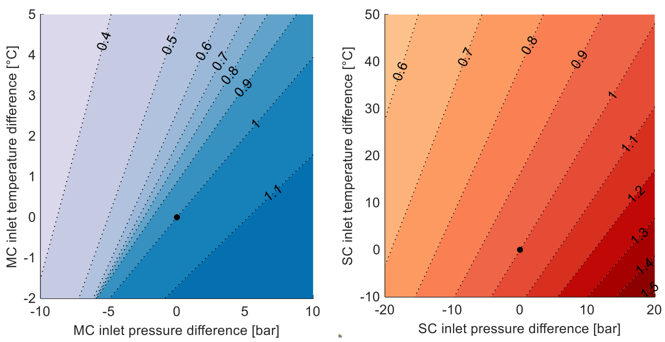

Figure 6. Ratio between the local density of CO_2 and the density at main compressor (left) and secondary compressor (right) inlet point in nominal conditions.

In each diagram three regions are present, corresponding to the following actions proposed to deal with ambient temperature increase considering the dry-cooled sCO_2-Flex power plant working at full load. All the simulations are carried out varying the cycle maximum pressure according to the sliding pressure turbine operative curve while keeping minimum pressure and maximum temperatures equal to the nominal values by inventory change.

1) A first action consists in keeping the main compressor inlet temperature equal to the nominal value by increasing the cooling air mass flow rate by adjusting the HRU fan rotational speed. This action would lead to a decrease of the HRU pinch point temperature difference as well as a reduction of the air temperature increase across the component. However, the maximum variation of the HRU fan rotational speed guaranteed by the electrical motors for V-shaped battery finned gas coolers is generally set at +25% of the nominal value. This strategy allows to feed the boiler with a nominal amount of coal (Figure 8) maintaining the gross power of the plant constant while the net power output slightly decreases because of the increased consumption of HRU caused by the increase of both the cooling air mass flow rate and pressure drops (Figure 7). Since main compressor inlet conditions remains unchanged all the other cycle thermodynamic points are unvaried from those in Table 1. As result, the components specific power remains almost constant (Figure 10) and cycle thermodynamic efficiency and plant overall efficiency (Figure 9) just slightly decrease because of the larger HRU consumption.
2) For an ambient temperature of 22.1°C, the maximum HRU fan rotational speed is reached, and the compressor inlet temperature starts to rise but it is still possible to fuel the system with a nominal coal mass flow rate (Figure 8). The main compressor volumetric flow rate increases as the compressor enthalpy head (Figure 10) due to the larger temperature increase in the component and a nearly constant cycle pressure ratio (Figure 8). The main compressor operative point is rapidly pushed towards the performance map upper limit (Figure 5.top) while the secondary compressor operative point slightly changes since the higher temperature inlet in off design (+6.8°C) does not appreciably affect the fluid compressibility factor. The system efficiency decreases according to the larger main compressor specific consumption (Figure 9).
3) For an ambient temperature of 23.6°C corresponding to a compressor inlet of 33.9°C, the upper bound of the main compressor operative map is reached and the only possibility to operate the system for higher ambient temperatures is to reduce the amount of pulverized coal fed to the boiler burners. Reducing the amount of heat input in the cycle leads to a decrease of the CO_2 mass flow rate, allowing to maintain the compressor operative point at the limit of the operative map. On the contrary, the operative point of the secondary compressor is progressively pushed towards the surge safety limit involving the activation of the antisurge loop. Although, the plant efficiency is slightly penalized, the main issue consists in the significant limitation of the plant maximum power output that can be offered on the energy market, thus strongly limiting the plant revenues during hot ambient temperature hours. As reported in Figure 7 for a temperature of 35°C the maximum power output is below 40% of the nominal one.

Another possible solution is represented by the variation of the split ratio through a motorized valve: as the ambient temperature increases, it could be possible to send the additional CO_2 flow rate from the main to the secondary compressor preserving system operability at higher temperatures without limiting fuel mass flow rate. This solution has not been investigated in this work and it will be evaluated in future publications.

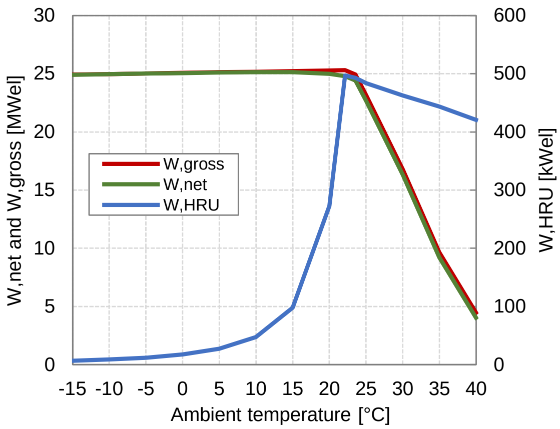

Figure 7. HRU fan consumption and plant net and gross power output as a function of the ambient temperature for the maximum attainable load.

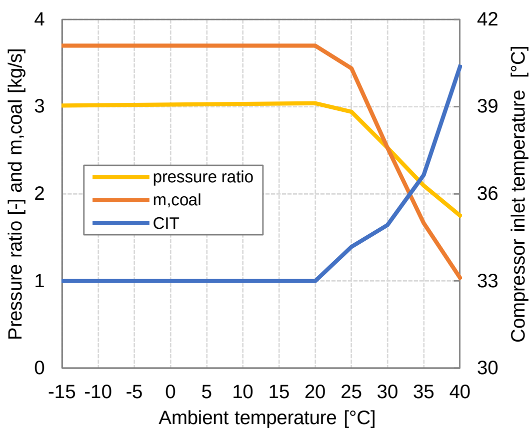

Figure 8. Compressor inlet temperature (CIT), coal mass flow rate and cycle pressure ratio as a function of the ambient temperature for the maximum attainable load.

On the contrary, if ambient temperature decreases it would be possible to reduce the compressor inlet temperature below the nominal point. However, in agreement with BHGE [27][28], the sCO_2-Flex partner in charge of the turbomachinery design, this strategy is not recommended since reduction of the compressor inlet temperature may lead to cavitation issues at the impeller inlet as vapor bubbles may form during the working fluid acceleration in the distributor and the stator. On the other hand, if cycle minimum temperature is below the critical one it would be possible to enable condensation by a proper tuning of the cycle minimum pressure with positive effects on thermodynamic efficiency but issues on the main compressor/pump operation related to the change of fluid volumetric behaviour. For these reasons, when ambient temperature decreases the compressor inlet temperature is kept equal to the nominal value by reducing the HRU fan rotational speed. Plant efficiency and power output slightly decreases because of the reduction of boiler efficiency determined by the lower ambient temperature and the consequent lower combustion air temperature.

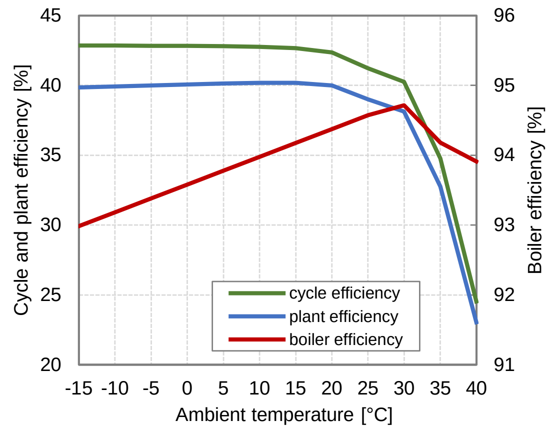

Figure 9. Cycle, plant, and boiler efficiencies as a function of the ambient temperature for the maximum attainable load.

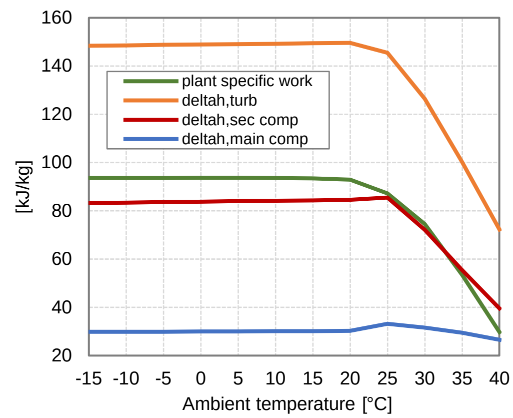

Figure 10. Plant specific work, turbine enthalpy drop and compressors enthalpy rise as a function of the ambient temperature for the maximum attainable load.

Finally, if the system is running at minimum load there is no limitation due to ambient temperature increase. Once HRU fan speed limit is reached strategy 2 can be adopted up to very high ambient temperatures because of the larger volumetric flow rate increase available starting from minimum load condition and nominal ambient temperature. The plant efficiency is progressively penalized by the increase of main compressor specific consumption, but the system operability is not limited (Figure 11).

EFFECT ON ANNUAL PLANT OPERATION

The effect due to power plant efficiency penalization and power output limitation can be evaluated on annual base for different site locations comparing the yearly energy production with a system working with a constant ambient temperature like for water cooled systems. Starting point is the knowledge of hourly data for dry bulb temperature and relative humidity plus information on the power plant load trend during the year. Weather data for different EU location can be obtained by EnergyPlus [29] while USA TMY3 data can be obtained from NREL System Advisor Model [30]. A reference coal fired power plant weekly load trend is obtained by [31] where the role of fossil fuel power plants is discussed for future scenario with high share of RES. Coal fired power plants are expected to run at maximum load for 9 hours a day (from 6 am to 1 pm and from 7 pm to 9 pm) in correspondence of the peak of grid energy consumption and at minimum reduced load in the central weekday hours (because of the abundance of solar energy) and during the night and the weekend (because of the reduction of energy demand). The same power plan is adopted in this study considering a minimum load of 25% [4]. Results are provided for two location: Prague and Sevilla. Meteorological data are reported in Figure 12 while Table 7 reports monthly and yearly energy produced considering a constant ambient temperature or by considering the actual power plant penalization due to ambient temperature variation for both selected locations.

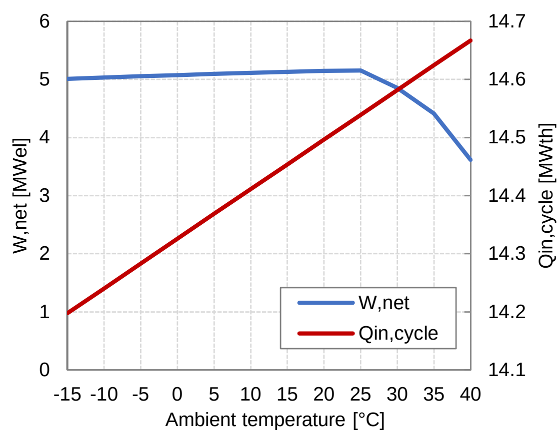

Figure 11. Plant net power output and cycle heat input at 25% of coal input as function of the ambient temperature.

Results clearly show that the larger difference between these two values is for Sevilla location (-6.3%) due to the higher average temperature while in Prague the penalization of energy output on annual base is extremely limited (-0.6%). This is mainly due to the power plan selected with sCO2 cycle running at minimum load in central hour of the day where the penalization due to higher temperature is stronger. However, the same conclusions are also valid for different power plans and the penalization for a plant installed in Prague is always dramatically lower than the energy output penalization for Sevilla thanks to the lower average ambient temperature and the limited number of hours with dry bulb temperature above 20°C. From these results it seems not necessary to adopt corrective actions for locations with low annual average temperature since the penalization is very limited while for hot locations two possible solutions are proposed that can be followed separately or in combination:

a) first one consists in designing the system with an oversized HRU in order to have the possibility exploit the larger heat transfer area and larger air mass flow rate with the aim to operate the thermodynamic cycle in nominal condition up to higher ambient temperatures;
b) second one consists in adopting wet-and-dry solutions with the aim to reduce air temperature by the cooling effect provided by heat and mass transfer process with a mass flow rate of softened water that is first sprayed on the fin battery and then distributed at the top of the adiabatic panel. Dry bulb ambient temperature, depending on the amount of water used, can be cooled down to almost the wet bulb temperature.

a) HRU oversizing

The quantification of the potential of adopting a larger HRU is evaluated by finding the maximum ambient temperature that allows to operate the thermodynamic cycle in nominal condition (Table 4) with maximum HRU fan rotational speed (+125%). The analysis is repeated for different multipliers of HRU heat transfer area and nominal air mass flow rate. System at nominal ambient condition can obviously work with all the HRU unit active and with a low fan speed with a reduction of HRU auxiliaries consumption or by bypassing some of them. Figure 13 reports the maximum ambient temperature attainable without varying the compressor inlet temperature against the HRU multiplier.

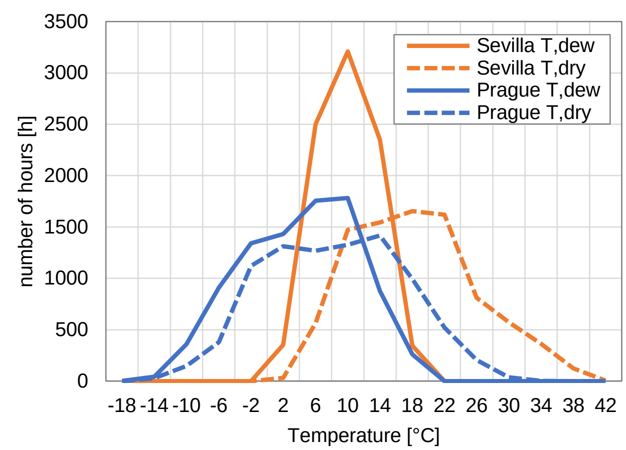

Figure 12. distribution of dry bulb and wet bulb temperature for Prague and Sevilla.

Table 7. Monthly and yearly energy yield for a constant ambient temperature or considering the actual penalization due to ambient temperature variation.

	20°C	**Prague, T_{dry}**		**Sevilla, T_{dry}**	
	GWh	**GWh**	**ΔEel%**	**GWh**	**ΔEel%**
Jan.	8.40	8.36	0.5%	8.40	0.0%
Feb.	7.43	7.39	0.5%	7.43	0.0%
Mar.	8.20	8.17	0.3%	8.19	0.1%
Apr.	7.88	7.86	0.2%	7.81	0.9%
May	8.40	8.32	0.9%	7.95	5.3%
Jun.	7.88	7.82	0.7%	6.96	11.7%
Jul.	8.20	8.13	0.9%	6.56	20.0%
Aug.	8.40	8.22	2.1%	6.74	19.7%
Sep.	7.68	7.68	0.0%	6.75	12.1%
Oct.	8.40	8.39	0.1%	7.97	5.1%
Nov.	8.07	8.05	0.3%	8.05	0.3%
Dec.	8.00	7.97	0.4%	8.00	0.0%
Year	**96.92**	**96.36**	**-0.6%**	**90.80**	**-6.3%**

Results show that with an overdesign equal to 300% it is possible to run the system in nearly nominal condition up to 32°C ambient temperature thus partially solving the critical issues related to system operability and reduction of attainable power output. Above this ambient temperature strategy 2 and strategy 3 can be adopted. The associated capital cost increase is 4.17 M$ involving a change of plant specific cost of about +8%. For Prague location where maximum ambient temperature is 32°C, an oversizing of HRU would allow to run the power plant close to nominal conditions with no limitation in power output. On the contrary, for Sevilla location where maximum ambient temperature is 41°C the HRU oversizing would allow to limit energy output penalization.

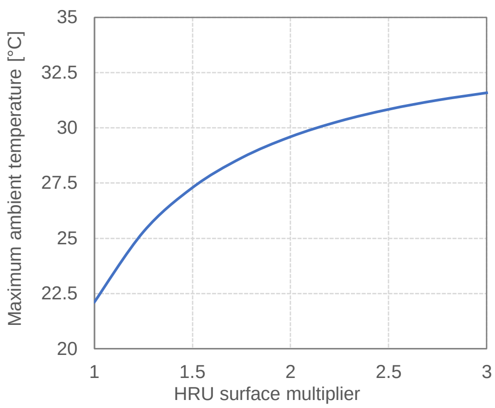

Figure 13. Maximum ambient temperature at which it is possible to maintain the nominal compressor inlet temperature as a function of the HRU size multiplier.

b) Wet-and-dry solution

Wet-and-dry heat rejection units adopt a system of water spray on the gas cooler heat transfer surface and/or use adiabatic panels which consists in a set of corrugated paper sheets mounted ahead the fin battery. Both solutions are already implemented in carbon dioxide gas coolers and condensers of refrigeration industry with an easy technological transfer to the power production sector. Figure 14 depicts the picture of a LU-VE wet-and-dry EMERITUS unit with a schematic of the adiabatic panel-water spray system.

Wet-and-dry solution allows for a reduction of the aforementioned issues by enhancing the heat exchanger duty during hot hours for a fixed air mass flow rate thus limiting the compressor inlet temperature increase and reducing the HRU footprint. Main drawback of this concept is related to water consumption that shall be softened before the use in order to limit corrosion and salt deposition on heat transfer surface. The minimum annual consumption of water (thus neglecting the efficiency of spray and adiabatic panels) can be calculated from weather data and plant operation. Calculations are carried out considering the exact amount of water required to bring the actual ambient temperature equal to the nominal value (20°C) neglecting the effect of enhanced heat transfer coefficient given by liquid water droplets and fins and increased heat capacity due to air humidification, thus leading to conservative results.

Results reported in Table 8 show that for Sevilla location it is possible to always reduce the ambient temperature close to the target value with the exception of few hours during the year where maximum temperature attainable with complete saturation is 22.2°C. The total annual energy can be increase to a value slightly higher than the value of a system working with constant ambient temperature. The minimum amount of water is around 20000 ton/year with a higher consumption during summer months. This consumption seems very large but considering a cost of softened water of 1.17 $/m^3$ [33] it represent a small variation of annual variable costs mainly related to coal consumption and in the order of 4.2 M$ considering a specific cost of coal equal to 76$/ton. Moreover, water consumption is around 0.13% of the water consumption associated to the adoption of a water cooled HRU demonstrating that wet-and-dry solution can be a powerful way to improve the energy production of a sCO_2 coal fired power plant in hot climate locations.

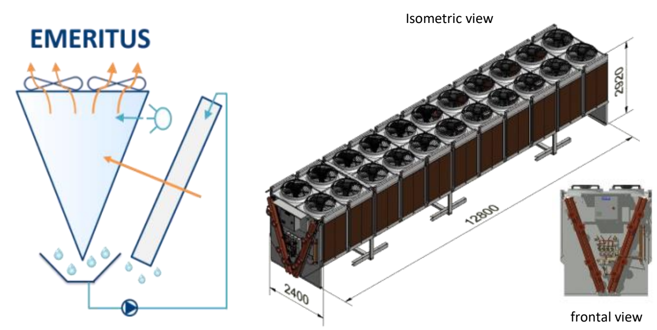

Figure 14. (left) Conceptual configuration of LU-VE EMERITUS technology, combining adiabatic panels and water spray system. Air is represented by orange streams while water by light blue streams. (right) Isometric and frontal views of a 22 fans Emeritus heat exchanger [32][33].

Table 8. Annual results for Sevilla location and wet-and-dry HRU.

	E_{el}, GWh	Spray hours	Water, ton
Jan.	8.40	0	0
Feb.	7.43	14	15
Mar.	8.20	88	194
Apr.	7.88	150	524
May	8.41	311	1637
Jun.	7.88	469	2944
Jul.	8.21	642	4987
Aug.	8.40	650	4996
Sep.	7.69	550	3167
Oct.	8.41	302	1512
Nov.	8.08	76	233
Dec.	8.00	7	4
Year	**96.97**	**3259**	**20214**

CONCLUSIONS

The present study allows to highlight the following aspects:

- Dry air cooled sCO_2 based coal fired power plants differently from steam Rankine power plants are strongly affected in their operation for ambient temperatures higher than the nominal one.
- Most critical component is the main compressor which volumetric flow rate increase is limited by the operative map leading to a decrease of maximum attainable power output for ambient temperatures just few degrees Celsius above the nominal value.
- The actual effect of this limitation on the plant operability during a representative year strongly depends on the location. Analysis is repeated for Prague and Sevilla demonstrating that in the first the annual penalization is nearly negligeable while it is higher in the second.
- In case of strong penalization of annual energy output for hot climate locations two solutions are proposed: first one requires to oversize the HRU while the second to adopt wet-and-dry gas coolers. Both solutions are feasible form techno-economic point of view and can be implemented separately or in combination.

NOMENCLATURE

Symbols

A	Area (m^2)
h	Specific enthalpy (kJ/kgK)
Eel	Electric energy (GWh)
$\dot{m}$	Mass Flow Rate (kg/s)
p	Pressure (bar)
Q	Thermal Power (W)
s	Specific entropy (kJ/kgK)
T	Temperature (°C)
W	Power (W)
η	Efficiency (%)
ρ	Density (kg/m^3)

Acronyms

APH	Air Preheater
BHGE	Baker Hughes General Electric
HRU	Heat Rejection Unit
LHV	Lower Heating Value
LTR	Low Temperature Recuperator
LT-PHE	Low Temperature Primary Heat Exchanger
HRU	Heat Rejection Unit
HTR	High Temperature Recuperator
HTRB	High Temperature Recuperator Bypass
HT-PHE	High Temperature Primary Heat Exchanger
HX	Heat Exchanger
sCO_2	Supercritical CO_2
USC	Ultra Super Critical

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