SIZING CRITERIA AND PERFORMANCE EVALUATION OF DIRECT AIR COOLED HEAT REJECTION UNITS FOR SUPERCRITICAL CO2 POWER PLANTS

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ABSTRACT

Supercritical CO_2 cycles (sCO_2) are recognized as a promising solution for the exploitation of different energy sources: from fossil fuel combustion and nuclear energy to solar energy and waste heat recovery. The large range of possible applications and the possible scarcity of water for sCO_2 systems makes the use of direct air-cooled heat rejection units (HRU) of great interest. This paper deals with the numerical modelling, supported by experimental data provided by an HRU manufacturer, of a sCO_2 system, equipped with a direct air-cooled HRU, exploiting a stream of 53 kg/s of flue gases at 550°C. The selected sCO_2 cycle is a simple recuperative cycle, whose maximum/minimum pressure and maximum temperature are optimized in design conditions obtaining an overall recovery efficiency of 21.71%. The analysis of the system in off-design conditions is carried out for different ambient temperatures and the optimal HRU operational strategy is investigated considering the possibility of varying the HRU fan rotational speed with no variation of the working fluid inventory. Overall results are presented as the trend of minimum cycle pressure and minimum cycle temperature required to maximize the system power output at every considered ambient temperature. It is found that the plant performance is strongly penalized by high ambient temperatures for two main reasons: (i) the sharp increase of the compressor/turbine power ratio because of the loss of real gas effects in compression and (ii) the severe penalization of the compressor efficiency, which moves its operating point towards regions characterized by poor performance. For these reasons, more effective HRU solutions should be adopted in locations characterized by relatively high seasonal temperatures.

1. INTRODUCTION

Small scale (1-10 MW_{el}) waste heat recovery (WHR) applications are widely distributed worldwide and can be adopted to enhance the efficiency of many different industrial processes (steel (Zhang et al. 2013), glass (Campana et al. 2013), cement production (Karellas et al. 2013)) and natural gas pipelines compressor stations (Gómez-Aláez et al. 2017). In this range of power output Organic Rankine Cycles (ORC) are recognized as the most reliable solution while for larger sizes steam power plants present a more affordable option. However, both solutions suffer from difficulties: safety and environmental issues related to the adoption of organic fluids affect ORC technology while steam cycles are characterized by poor turbine efficiency and slow transients. In recent years CO_2-based transcritical and supercritical cycles have been proposed as a viable option for the exploitation of several different energy sources including WHR. The main advantages of CO_2 cycles are represented by (i) the high cycle efficiency attainable with a compression process close to the working fluid critical point, (ii) the use of an environmentally friendly, widely available, safe and thermally stable working fluid, (iii) the compactness of the components that leads to a high flexibility of the system. A comparison between ORC and sCO_2 cycles have been carried out by Astolfi et al. (2018) showing the potential of the second solution when a high temperature heat source is available: different cycle configurations have been compared and both cascade cycle and simple recuperative cycles resulted competitive against ORC. However, the use of sCO_2 requires the development of specific components and the definition of control strategies tailored to the specific cycle configuration. More than 50 cycle configurations are proposed in literature (Crespi et al. 2017) and the best one shall be selected for each application according to the heat source properties. In order to propose a cycle ready to penetrate the market and compete with conventional solutions, it is crucial to limit the complexity of the system and possibly using components developed for other applications in order to exploit already established scale economies and to foster technological transfer between different industrial fields. The scientific literature about sCO_2 cycles is mainly focused on the challenging design of the turbomachinery that must face difficulties related by the marked real gases effects of the fluid and the recuperators that show very large pressure differences between cold and hot streams, the need of enhancing the global heat transfer coefficient while limiting the pressure drops. On the contrary, the accurate description of the HRU is often neglected and this component is simply modelled as a lumped component with a sufficiently large heat transfer area and cooling fluid mass flow rate. However, the design of the HRU is not trivial and may strongly affect the operation of a sCO_2 power plant due to the large volume and mass of working fluid contained and its crucial role in the control of the main compressor inlet temperature. In this specific case, an air gas cooler is considered as the best option with the aim at designing a power system versatile to different locations and without the need of cooling water. The positive aspect is related to the numerous similarities between the HRU of sCO_2 power cycles and the HRU of CO_2 refrigeration cycles. In particular, for both cases the inlet temperature and pressure are similar and vary in the range 80-150°C and 60-100 bar respectively and the CO_2 goes through a similar transformation with a final temperature close to the critical point or in saturated liquid conditions. Technology transfer is eased and, with the aim of presenting a realistic operation of the HRU, the present work is supported by LU-VE, one of the market leaders in the production of air-cooled condensers and gas coolers for refrigeration and small ORC power systems. LU-VE experimented a remarkable growth of the production of HRU for CO_2 systems with a number of sold unit almost tripled in the last five years. Moreover, LU-VE produces both conventional dry air and innovative humid air sprayed coolers also called Emeritus® coolers where the heat transfer is enhanced by spraying water on both battery fin and/or on adiabatic cooling panels allowing for a marked reduction of water consumption and scaling issues (Astolfi et al. 2017).

2. NUMERICAL CODE DESCRIPTION

A dedicated numerical tool has been implemented in MATLAB allowing for the optimization of the system design and the evaluation of part load operation. CO_2 thermodynamic properties are provided by Refprop 9.1 database leading to an accurate evaluation of real gas effects close to the critical point of the working fluid. With the developed numerical code, any cycle configuration can be implemented, optimized and simulated over a large range of off-design conditions. According to the small scale WHR application discussed in this study, the cycle configuration selection is pushed towards the simplest plant scheme among those proposed in literature, namely the simple recuperative cycle, whose schematic is reported in Figure 1, left. This cycle adopts a single compressor working close to the CO_2 critical point in order to exploit fluid real gas effects and reduce compressor consumption. Recuperative process allows limiting the heat discharged at the HRU but on the other side it increases the temperature at the PHE inlet thus limiting the exploitation of the variable temperature heat source. However, the low compressibility factor at compressor outlet involves a specific heat on the recuperator cold side larger than the specific heat of the low-pressure fluid on the hot side leading to a final temperature difference far larger than recuperator the pinch point. A solution to enhance the heat recovery efficiency would be to implement a recuperator bypass in order to further cool down the heat source and reduce the temperature differences in the recuperator (see Figure 1, right). A comparison among these two cycle configurations will be presented in future steps of this work.

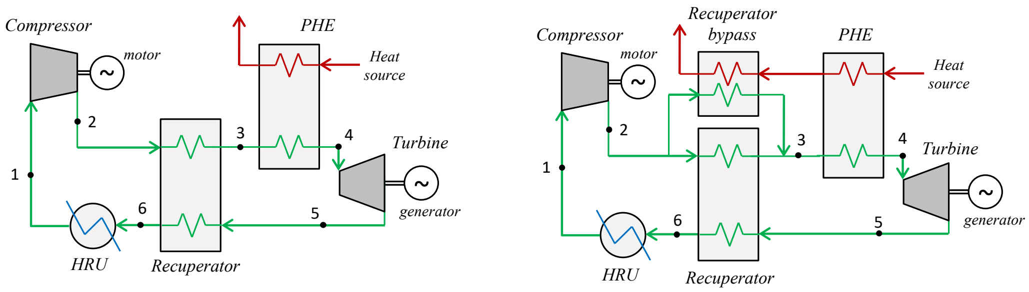


Figure 1 : Schematic of the selected simple recuperative sCO_2 cycle (left) and of the simple recuperative sCO_2 cycle with recuperator bypass (right).

2.1 Design optimization

The optimization of the plant design is carried out with the aim of maximizing the net power output including the consumption of the HRU auxiliaries. Table 1 reports the assumption adopted for the cycle design: among them the most significant one is the minimum cycle temperature that is selected in order to perform the main compression close to the critical point of the fluid, exploiting the real gas effects and increase the system efficiency. Turbomachinery efficiency and heat exchanger minimum temperature differences have been assumed considering different references from literature although a real industrial benchmark is not available for most of the components. The optimization variables are the working fluid minimum and maximum pressures and the turbine inlet temperature and for each quantity a proper upper and lower bound value is considered in order to obtain a feasible final solution. Heat source is modelled as an ideal gas with a mass flow rate $\dot{m}_{hs}$ of 53.1 kg/s, a maximum temperature $T_{hs,max}$ of 550°C, a minimum allowable temperature of 150°C and a specific heat $c_{p,hs}$ equal to 1.15 kJ/kgK, representing a flue gas stream from a combustion process available from either a gas turbine or and industrial process (steel, glass, cement industry).

Table 1: Main assumptions for the cycle design

Heat source data and cycle design assumptions			
Heat source mass flow rate $\dot{m}_{hs}$, kg/s	53.1	HRU CO_2 $(\Delta p/p_{in})$	0.5%
Heat source temperature $T_{hs,max}$, °C	550	Recuperator hot side $(\Delta p/p_{in})$	0.5%
Minimum heat source temperature $T_{hs,min}$, °C	150	Recuperator cold side $(\Delta p/p_{in})$	0.5%
Heat source specific heat $c_{p,hs}$, kJ/kgK	1.15	Turbine isentropic efficiency, η_{turb}	85%
Maximum admissible cycle temperature, °C	525	Compressor efficiency, η_{comp}	80%
Maximum admissible cycle pressure, bar	250	Generator/motor efficiency $\eta_{me,t}/\ \eta_{me,c}$	96.4%
Minimum cycle temperature, °C	33	Ambient air temperature, °C	20
Recuperator pinch point ΔT_{REC},°C	10	HRU electric consumption per MW of heat rejected ξ (data provided by manufacturer)	0.0085
PHE pinch point ΔT_{PHE},°C	25		
PHE CO_2 Δp, bar	2.5	Heat source heat transfer coefficient, W/m^{2}K	125

The numerical model includes also a set of routines for the heat exchanger design, the calculation of the heat transfer coefficients, the internal volume of the plant components and the mass of the heat exchangers. The recuperator is modelled as a printed circuit heat exchanger (PCHE) and main information is derived by the work of Dostal et al. (2004), integrated with manufacturer data and already presented by Alfani et al. (2019). The primary heat exchanger is modelled as a finned tube heat exchanger with flue gases flowing outside the tubes. Performance of the LU-VE heat exchangers are predicted with proprietary correlations computing the heat transfer coefficients and the pressure drops as function of the operating conditions. From the design of each heat exchanger and assuming reasonable piping length between the different components, the inventory of CO_2 can be calculated. The main assumptions related to the heat exchangers design adopted in this work are reported in Table 1.

Table 2: Heat exchangers main design assumptions

Other heat exchangers design assumptions			
PHE		**PCHE**	
Tube internal diameter, mm	20	Thickness of plate, mm	1.5
Ratio of tube pitch to external diameter	1.25	Diameter of semi-circular channel, mm	2
Ratio of finned ext. area to plain ext. area	12	Thickness of wall between channels, mm	0.4
Tube material	Carbon steel	Heat exchanger material	INCOLOY 800

The main indexes used to evaluate the system performance are the cycle net efficiency (including HRU auxiliaries consumption) η_{cycle}, the heat recovery factor χ and the overall recovery efficiency η_{rec} defined as:

$$\eta_{cycle} = \frac{\dot{W}_{net}}{\dot{Q}_{in,cycle}} = \frac{\dot{W}_{t} - \dot{W}_{c1} - \dot{W}_{c2} - \dot{W}_{HRU,aux}}{\dot{Q}_{in,cycle}} \qquad \text{Eq.1}$$

$$\chi = \frac{\dot{Q}_{in,cycle}}{\dot{Q}_{hs,max}} = 1 - \frac{\int_{T_{hs,min}}^{T_{stack}} \dot{m}_{hs} cp_{hs}(T) dT}{\int_{T_{hs,min}}^{T_{hs,max}} \dot{m}_{hs} cp_{hs}(T) dT} \qquad \text{Eq.2}$$

$$\eta_{rec} = \chi\, \eta_{cycle} = \frac{\dot{W}_{net}}{\dot{Q}_{hs,max}} \qquad \text{Eq.3}$$

2.2 Off-design analysis

The analysis of the system in off-design conditions is carried out considering different ambient temperatures while keeping constant the mass flow rate and the maximum temperature of the heat source in order to catch the effect of the ambient conditions on the cycle performance and the HRU operational strategy. Maximum cycle temperature is kept equal to the nominal one in order to avoid metal overheating, while maximum cycle pressure is determined by assuming choked flow in the turbine nozzles and sliding pressure operation. Each off-design solution is by verifying energy and mass balances on each component while pressure drops (Δp), heat transfer coefficients (h) and fan consumptions ($\dot{W}_{HRU,aux}$) are calculated with reference to the nominal value and adopting the exponential functions reported in Eq. 4-6 except for HRU unit that adopts ad hoc correlations (Crespi et al. 2017). Turbine efficiency is kept constant considering the limited variation of mass flow rate and pressure ratio in the investigated range of conditions while compressor efficiency penalization is neglected in part load since this work aims to understand the actual operating range of this machine and suggest a criteria for the selection of the nominal point within the operational map.

$$\Delta p = \Delta p_{design} \left(\frac{\rho_{design}}{\rho}\right)\left(\frac{\dot{m}}{\dot{m}_{design}}\right)^2 \qquad \text{Eq.4}$$

$$h_X = h_{X,design}\left(\frac{\dot{m}_X}{\dot{m}_{X,design}}\right)^{\alpha} \quad \text{with} \begin{cases} X = CO_2 & \alpha = 0.8 \\ X = gas & \alpha = 0.6 \end{cases} \qquad \text{Eq.5}$$

$$\dot{W}_{HRU,aux} = \dot{W}_{HRU,aux,design}\left(\frac{\dot{m}_{air}}{\dot{m}_{air,design}}\right)^{2.78} \qquad \text{Eq.6}$$

The power output can be maximized at any operating point by acting on the cycle minimum temperature and pressure and by considering additional constraints related to turbomachinery operative limits or inventory control limit. In particular, the part load control of closed ideal gas cycles can be obtained through the removal of a fraction of the fluid mass from the circuit to be temporarily stored in a dedicated vessel, with the aim of reducing the cycle pressures and allowing the turbomachinery to work close to nominal conditions also in part load. This strategy, for sCO_2 cycles has not necessarily the same positive effects, since strong real gas effects close to the critical point determine large variations of fluid compressibility factor even for small changes of operating conditions. However, with the aim of studying a small WHR system that shall be easily installed and operated in a large variety of applications, a totally sealed system without inventory change may represent a large advantage leading to a reduction of system cost and an easier control system.

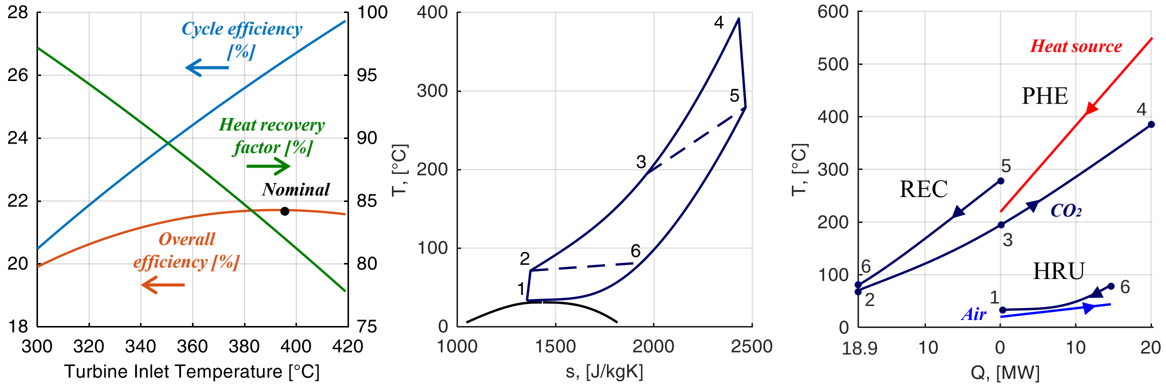
3. RESULTS

In this section the main results attainable with the methodology presented are reported. First, an analysis at nominal ambient temperature is presented in order to discuss then trend of main quantities as function of compressor inlet temperature and inlet pressure. The second analysis is focused on the description of the system operation for different ambient temperatures. Finally, the overall results are presented and the potential of adopting a wet-and-dry solution for hot climate locations is discussed.

3.1 Plant design performance

In Figure 2.a it is possible to see how the cycle that maximizes the overall recovery efficiency, with a maximum temperature of 392°C, represents the best compromise between the cycle efficiency that increases with the maximum cycle temperature and the heat recovery factor that increases by reducing the hot flue gases stack temperature.

Furthermore, in Figure 2.b-c are presented the optimal cycle T-s and T-Q diagrams, while the main results are reported in Table 3. Net power output of the optimized cycle is 5.3 MW_{el} with a cycle thermodynamic efficiency of 26.29% and a thermal recovery efficiency equal to 82.58%, combined to obtain an overall recovery efficiency equal to 21.71%.



(a) (b) (c)

Figure 2: (a) Trends of the main system efficiencies and heat recovery factor as function of the turbine inlet temperature, (b) T-s diagram of the best cycle design and corresponding T-Q diagrams of the heat source/CO_2 heat exchangers (c).

Table 3: Optimum cycle layout results

Cycle design results			
CO_2 mass flow at turbine inlet, kg/s	77.5	Thermal power recovered, MW_{th}	20.15
Maximum cycle pressure p_2, bar	250	Available thermal power, MW_{th}	24.40
Minimum cycle pressure p_1, bar	79.18	Turbine electric power, MW_{el}	7.93
Turbine inlet temperature T_5, °C	392.33	Compressor electric power, MW_{el}	2.51
Heat source outlet temperature, °C	219.67	HRU auxiliaries consumption, kW_{el}	121.97
Total CO_2 mass within the plant, kg	2222.6	Net plant power output, MW_{el}	5.30
Cooling air mass flow rate, kg/s	610.21	**Cycle thermodynamic efficiency**	**26.29%**
Cooling air mass temperature rise, °C	23.9	**Heat recovery factor**	**82.58%**
Ambient temperature, °C	20	**Overall recovery efficiency**	**21.71%**

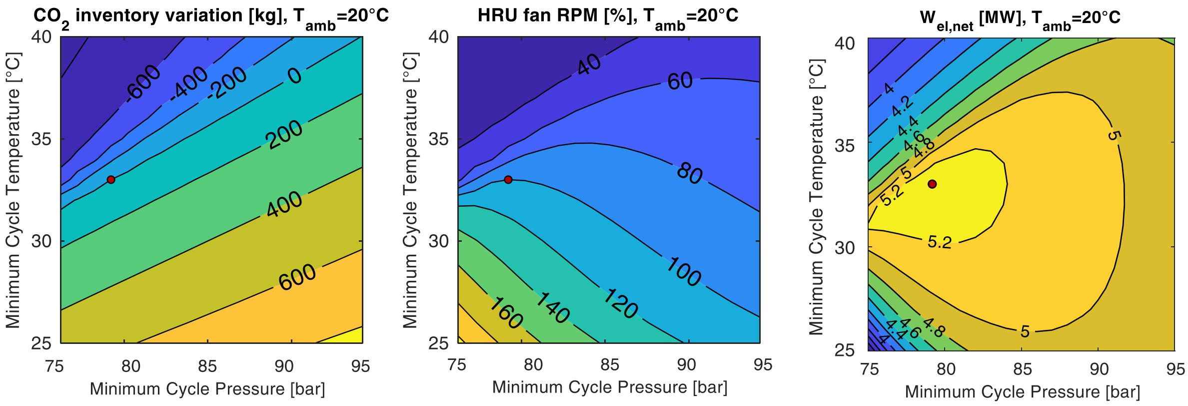
Table 4: Heat exchangers sizing results

Parameter	**HRU**	**REC**	**PHE**
Heat duty, MW	14.35	18.22	20.15
Hot side heat transfer coefficient, W/m^2K	4954.5	3959.0	125
Cold side heat transfer coefficient, W/m^2K	75.8	5801.2	2940.1
Global heat transfer coefficient, W/m^2K*	1069.8	2145.6	896.2
Internal heat transfer surface, m^2	1199.4	238.3	300.1
HX metal mass, kg	9463	1494.3	8804.4
CO_2 inventory in the component, kg	1062.9	50.5	405.3

*=referred to internal area

3.2 Off-design maps for the nominal ambient temperature case

Figure 3 depicts the maps of fluid inventory (Figure 3.a), fan rotational speed/fan nominal rotational speed ratio (Figure 3.b) and cycle power output (Figure 3.c) as function of the compressor inlet temperature (namely the minimum cycle temperature and the HRU outlet temperature) and cycle minimum pressure for an ambient temperature of 20°C. Fluid inventory is strongly dependent on both cycle minimum pressure and temperature: in particular, as the nominal point is close to the critical point of the working fluid, a small increase of cycle minimum temperature leads to a dramatic reduction of fluid density at HRU outlet and consequently a reduction of the fluid inventory in this component. All the solutions lying on the constant inventory line can be obtained by controlling the fan rotational speed. Reducing the fan speed leads to a lower cooling air mass flow rate (and lower external heat transfer coefficient) at the HRU and thus to larger approach point temperature difference in the component and a consequent higher CO_2 minimum pressure.



(a) (b) (c)

Figure 3: Maps of fluid inventory (a), fan rotational speed/fan nominal rotational speed ratio (b) and cycle power output (c) as function of the minimum cycle temperature and cycle minimum pressure. Red marker represents the design condition.

As result, the compressor specific work increases due to the partial loss of real gas effects. This fact, combined to the decrease of the cycle pressure ratio, leads to a remarkable reduction of net power output as it can be seen in the map in Figure 3.c, which presents a region of maximum close to the nominal point represented as a red marker. It can be noted that it would be possible to slightly increase the power output by increasing the fan speed while keeping constant the fluid inventory, however this benefit is almost negligible confirming the validity of the design solution.

3.3 Optimal results for different ambient temperatures

The same analysis is carried out for different ambient temperatures and results are presented in a single chart for each investigated temperature in the range 5-35°C. Each map presents the contour plot of the net power output with the constant inventory line depicted in green, the nominal fan rotational speed in blue and the maximum rotational speed of the HRU fans in red. This value represents the maximum overspeed achievable by the fans electrical motor and it is equal to the 125% of the nominal rotational speed. Figure 5.f depicts the results for ambient temperature equal to 35°C showing that it is possible to operate the cycle in a region of maximum power output with a totally sealed plant. The higher ambient temperature involves an increase of both cycle minimum temperature and pressure with a consequent reduction of system efficiency and net power output that decreases from 5.30 MW down to 4.26 MW (-20%). On the contrary if the ambient temperature is lower than the nominal one and the fans rotational speed is kept constant the minimum cycle temperature can be reduced potentially increasing the cycle performance. However, also the minimum pressure of the cycle reduces in order to keep the inventory unchanged and this leads to subcritical pressures in the HRU and fluid condensation, as depicted in Figure 5.a-b for ambient temperature equal to 5°C and 10°C respectively. This operating condition may negatively affect the compressor operation that would pump liquid carbon dioxide instead of a compressed gas. On the contrary, it is possible to keep the minimum cycle pressure over the critical point by reducing the fan speed and increasing the cycle minimum temperature: this strategy allows avoiding condensation but on the other side penalizes the cycle efficiency.

For ambient temperatures lower than the nominal it is also possible to notice a marked discontinuity in the net power output while crossing the condensation line: for a fixed subcritical minimum cycle pressure, working with a compressor inlet temperature above the respective saturation value means having superheated CO_2 at compressor inlet with an increase of compressor specific work. On the contrary, the compressor inlet condition is in subcooled liquid state, requiring a remarkably smaller amount of compressor specific work. In Figure 5 this situation is depicted through the T-s diagrams of the sCO_2 cycle in off-design conditions with a compressor inlet temperature equal to 25°C: in Figure 5.a the compressor inlet pressure is equal to 60 bar (superheated vapor conditions) while in Figure 5.b the compressor inlet pressure is equal to 70 bar (subcooled liquid conditions).

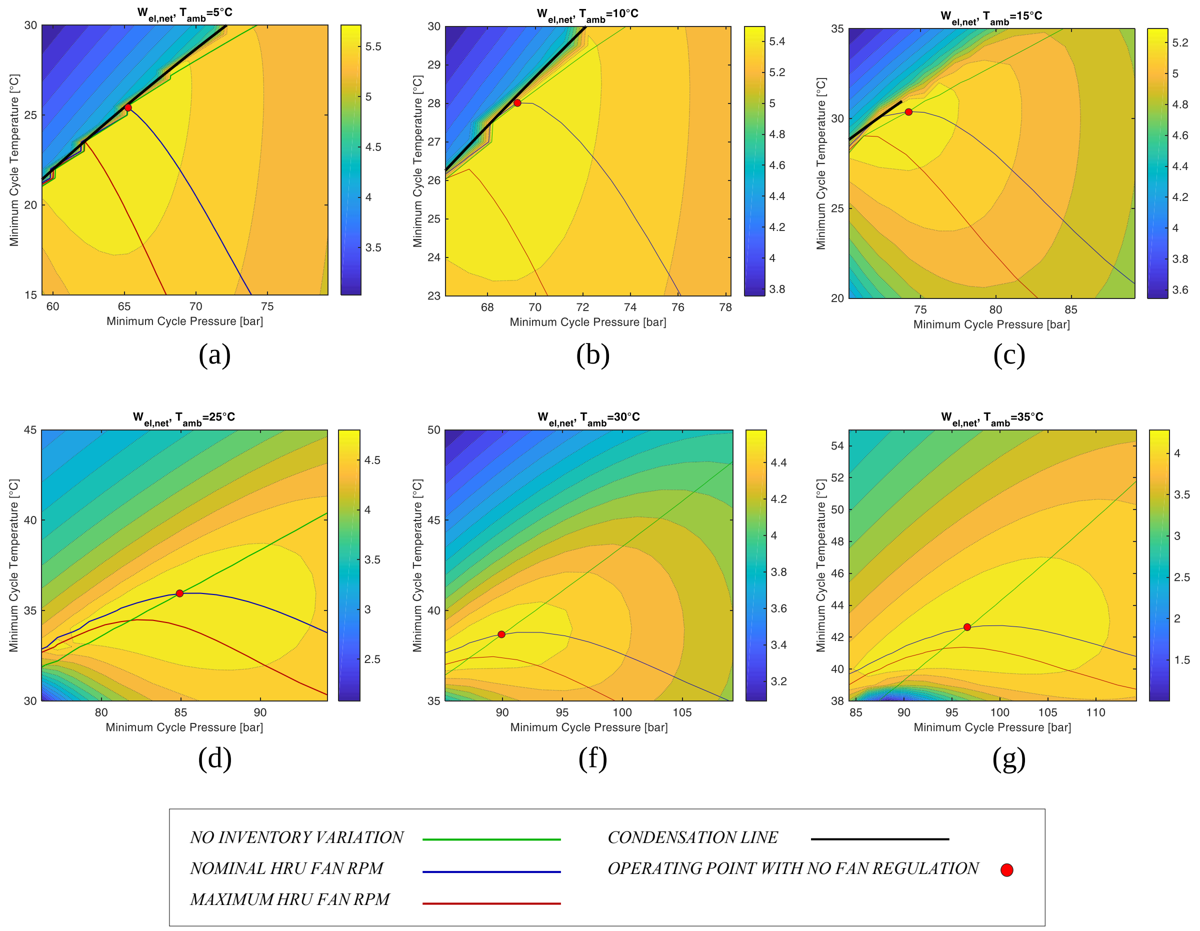


Figure 4: Contour plots of the net power output for different ambient air temperatures. The constant inventory line is depicted in green, the nominal fan rotational speed in blue and the maximum rotational speed of the HRU fans in red.

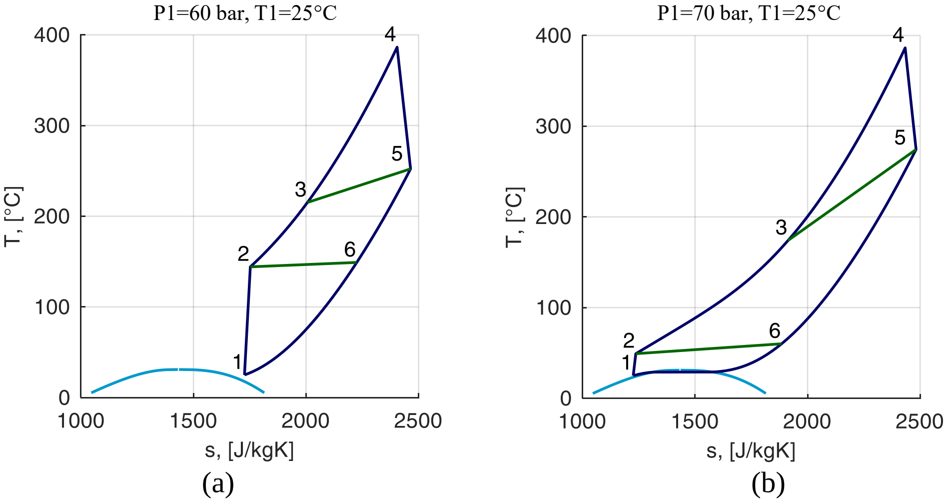
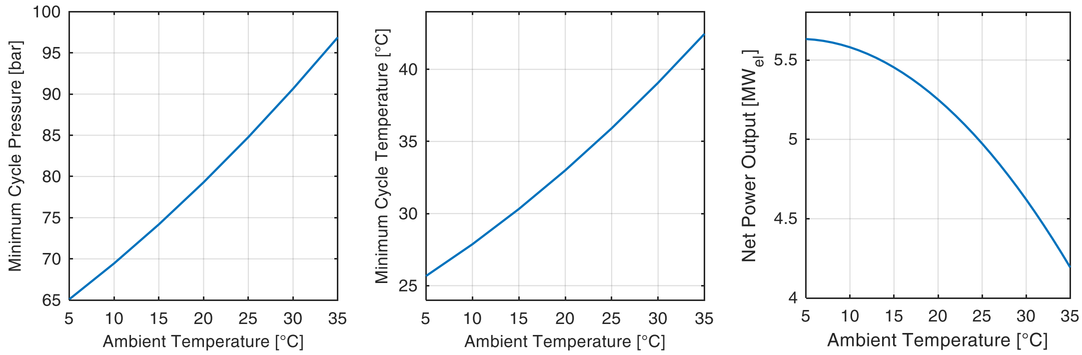


Figure 5: T-s diagrams of the sCO_2 cycle in off-design conditions with a compressor inlet temperature equal to 25°C: (a) compressor inlet pressure is equal to 60 bar (superheated vapor conditions), (b) compressor inlet pressure is equal to 70 bar (subcooled liquid conditions).

3.4 Overall results

Overall results are presented as the trend of minimum cycle pressure (Figure 6.a) and minimum cycle temperature (Figure 6.b) required to maximize the cycle power output (Figure 6.c) at constant plant CO_2 inventory. Net cycle power output is more penalized for CO_2 power plants than for other WHR power plants based on Rankine cycle because the sharp increase of the ratio between compressor and turbine work (Figure 6.c). It is important to underline that in these overall results the off-design performance of the compressor is not considered but it is possible to highlight some possible issues related to its operation. In particular when ambient temperature increases, the constraint of constant fluid inventory leads to an opposite variation of volume flow rate and enthalpy head on the compressor: the first index increases because the higher minimum temperature affects the fluid density more than the increase of pressure while the second decreases because of the lower pressure ratio. As result for ambient temperature lower than the nominal the compressor operating point is pushed towards the surge line while for higher temperatures the compressor works close to the choke line where it generally shows lower performances. The range of variation for volumetric flow rate and enthalpy head seems moderately large (+20%/-20% for volumetric flow rate and +10%/-11% for the enthalpy head) and, if the compressor nominal operating point is not properly selected, its off-design operation may be unfeasible or penalized in efficiency. For these reasons, wet-and-dry solutions are particularly promising for sCO_2 power plant since they allow to reduce the air temperature down to the wet-bulb value limiting the power output penalization in hot hours and contextually limiting the variation of volumetric flow rate and enthalpy head for the compressor.



(a) (b) (c)

Figure 6: Trends of minimum cycle pressure (a), minimum cycle temperature (b) and net cycle power output (c) as function of the ambient temperature for the sCO_2 cycle with no inventory variation.

4. CONCLUSIONS

In the present work, the study of a simple recuperative sCO_2 cycle as power cycle for waste heat recovery from a hot gas stream at 550°C was performed. The cycle maximum temperature and maximum/minimum pressures have been optimized in order maximize the overall recovery efficiency. A maximum overall recovery efficiency of 21.71% was obtained for a maximum cycle temperature and pressure of 392.33°C and 250 bar respectively and for a minimum pressure of 79.18 bar. The selected maximum temperature guarantees the best compromise between cycle efficiency (26.29%) and heat recovery factor (82.58%). Once a preliminary design of all the main heat exchangers was performed, several off-design simulations have been carried out considering different ambient temperatures while keeping constant the mass flow rate and the maximum temperature of the heat source in order to catch the effect of the ambient conditions on the cycle performance and the HRU operational strategy.

A wide range of cycle minimum temperature and pressure have been investigated in order to identify the optimal off-design operating point for each ambient temperature, considering the additional constraint related to constant inventory control. Eventually, overall results are presented as the trend of minimum cycle pressure and minimum cycle temperature required to maximize the cycle power output at constant plant CO_2 inventory. It is found that the plant performance is strongly penalized by high ambient temperatures due to the sharp increase of the compressor/turbine power ratio caused by the increase of the CO_2 compressibility factor at compressor inlet. Moreover, the operation of the compressor is challenging because of the need of bypass recirculation to avoid surge for low ambient temperatures and reduced compressor efficiency near to choke conditions for high ambient temperatures suggesting that the cycle design conditions must be carefully selected considering the actual range of ambient temperatures. For these reasons, more effective HRU solutions should be adopted in locations characterized by relatively high seasonal temperatures: in particular wet and dry solution as the LU-VE

Emeritus® coolers would be able to maintain the desired low minimum temperature of CO_2 even at ambient temperatures above 35 °C, as demonstrated by refrigeration plants experience. It is important to underline that the operation and the off-design performance of the compressor is not discussed in this paper and it may strongly affect the results. This further aspect will be discussed in future works on the topic.

NOMENCLATURE

Symbols

A	Area (m^2)
cp	Specific Heat Capacity (J/kgK)
h	Heat Transfer Coefficient (W/m^2K)
$\dot{m}$	Mass Flow Rate (kg/s)
$\dot{Q}$	Thermal Power (W)
T	Temperature (°C)
U	Overall heat transfer coeff. (W/m^2K)
$\dot{W}$	Power (W)
η	Efficiency (%)
ξ	Specific HRU Elec. Aux (W_{el}/ W_{th})

Acronyms

APH	Air Preheater
HRU	Heat Rejection Unit
LHV	Lower Heating Value
LTR	Low Temperature Recuperator
HRU	Heat Rejection Unit
HTR	High Temperature Recuperator
HTRB	High Temperature Rec. Bypass
HX	Heat Exchanger
PCHE	Printed Circuit Heat Exchanger
PHE	Primary Heat Exchanger
RPM	Revolutions Per Minute
SCO_2	Supercritical CO_2
SR	Split Ratio
VIGV	Variable Inlet Guide Vanes

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