D5.1 - Report on the nominal design of different plant configurations and sensitivity analysis on main design parameters

January 2019

WP 5, T 5.1

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sCO2-Flex

This project has received funding from the European Union's Horizon 2020 research and innovation programme under grant agreement N° 764690.

Technical References

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Project Title	Supercritical CO2 Cycle for Flexible and Sustainable Support to the Electricity System
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[1] PU = Public

PP = Restricted to other programme participants (including the Commission Services)

RE = Restricted to a group specified by the consortium (including the Commission Services)

CO = Confidential, only for members of the consortium (including the Commission Services)

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Summary

POLIMI has finalized the MATLAB code for the design and optimization of sCO_2 cycles that will be at the basis of the off-design tool and the dynamic tool required in WP5. With reference to results reported in D1.3, the following analyses have been performed:

1. ***VALIDATION OF D1.3 RESULTS***
   Results of D1.3 are checked and validated. The MATLAB numerical code developed at POLIMI replicates perfectly the results provided by EDF in D1.3 and obtained with Aspen Plus. Energy balances of different components and of the overall thermodynamic cycle are verified.

2. ***SENSITIVITY ANALYSES ON MAIN CYCLE PARAMETERS***
   A series of sensitivity analyses on the most significant cycle parameters have been performed in order to catch their influence on the final cycle efficiency and on the sizing of the main components.

The nomenclature adopted in this document is the one introduced in D1.1.

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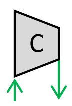
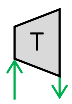
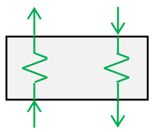
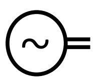
This project has received funding from the European Union's Horizon 2020 research and innovation programme under grant agreement N° 764690.

Nomenclature

Symbol	Description	Unit
sCO_2	Supercritical CO_2	-
WP	Work package(s)	-
T	Temperature	°C
p	Pressure	MPa
η	Efficiency	%
HTR	High temperature recuperator	-
LTR	Low temperature recuperator	-
REC	Both LT and HT recuperators	-
PHE	Primary heat exchanger	-

Acronym	Partner
POLIMI	Politecnico di Milano
BHGE	Baker Hughes General Electric
EDF	Electricité de France

Symbol	Type	Labelling
	Heat sinks	MSCUSXX
	Heat sources (heaters)	MSHSOXX
	Compressor	MSCOMXX
	Turbine	MSTURXX
	Recuperator	MSRCUXX
	Electric motor or generator	GEN

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1. Introduction and brief description POLIMI numerical model

This document summarizes the main outcome of the preliminary calculations carried out so far by POLIMI in the framework of WP5 of the H2020 project sCO_2-Flex. sCO_2-Flex main goal is to define the best sCO_2 cycle configuration for the next generation coal power plants. This objective will be tackled taking into account not only the nominal system performance but also its flexibility, namely the ability to perform fast transients and to reach low part-load operation with outstanding performance. The preliminary study performed in WP1 and reported in D1.3 have identified two promising cycle configurations that will be further investigated during the project: cycle #16 represents the recompression cycle with HTR bypass while cycle #23 identifies the partial cooling cycle with double reheat. The efficiency of both cycles has been optimized varying cycle pressures and adopting specific constraints on maximum temperature and pressure and using realistic components' performance as suggested by WP2, WP3 and WP4 activities. However, in order to understand the real potential of sCO_2 power plants against USC power plants a series of sensitivity analyses is required in order to highlight the impact of the most relevant assumptions and suggest possible modifications to the hypotheses at the base of WP1 calculations. In order to face this task POLIMI has implemented a dedicated numerical tool in MATLAB [1] adopting Refprop 9.1 [2] for the calculations of the thermodynamic properties of CO_2. It has been preferred to develop an in-house numerical code instead of using a commercial software in order to have maximum flexibility and adaptability of the code and to be able to simulate any kind of cycle configuration in both design and off-design conditions. Furthermore, in this manner, the link between the on-design code and the off-design steady state or the dynamic numerical code will be more straightforward. The code can compute the cycle thermodynamic performances by solving mass and energy balances and a preliminary component sizing for all the different cycle configurations investigated in D1.1, whereas in this deliverable only the cycles selected in D1.3 are investigated. The sensitivity analyses have been carried out varying both design parameters (component performances, pressure drops and heat exchanger temperature differences) and maximum and minimum cycle temperatures evaluating also the effect of condensation. Finally, a dedicated sensitivity analysis has been carried out showing the preliminary results of the trade-off between heat exchangers dimensions (namely thermal inertia of the system) and cycle efficiency giving an insight for the design of future sCO_2 power plants characterized by high flexibility.

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2. Validation of D1.3 results

Numerical model validation has been carried out on the two cycle configurations selected in D1.3. Cycle #16 represents the recompression cycle with HTR bypass while cycle #23 identifies the partial cooling cycle with double reheat. Cycle layout and Ts diagram are reported for both cycle configurations in Figure 2.1 and Figure 2.2, respectively.

Validation is performed by imposing the same thermodynamic conditions, namely the same maximum and minimum pressures and temperatures of the cycle and adopting the same assumptions of D1.3. The MATLAB numerical code developed at POLIMI is able to replicate the results provided by EDF in D1.3 with an accurate evaluation of both the net power output and the cycle efficiency.

Results of the validation are reported in Table 2.1.

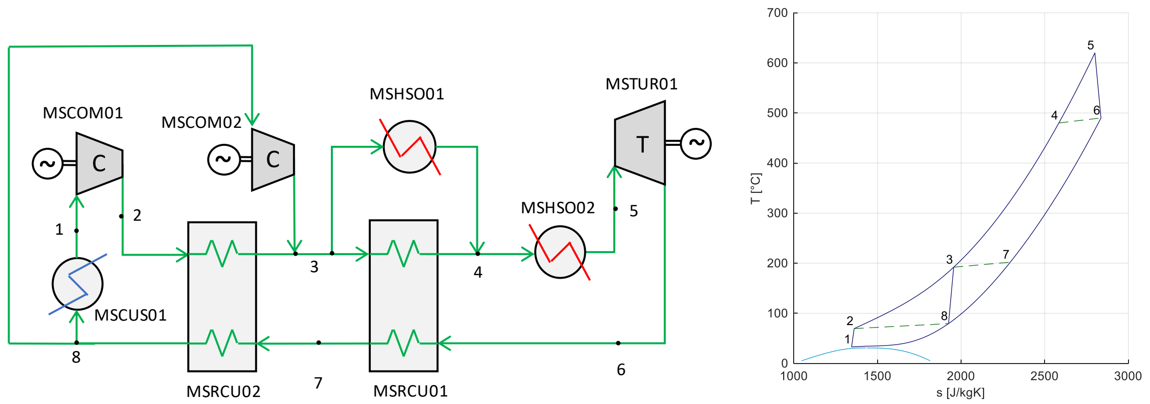

Figure 2.1 – Cycle layout and Ts diagram of the recompressed configuration with HTR bypass (16).

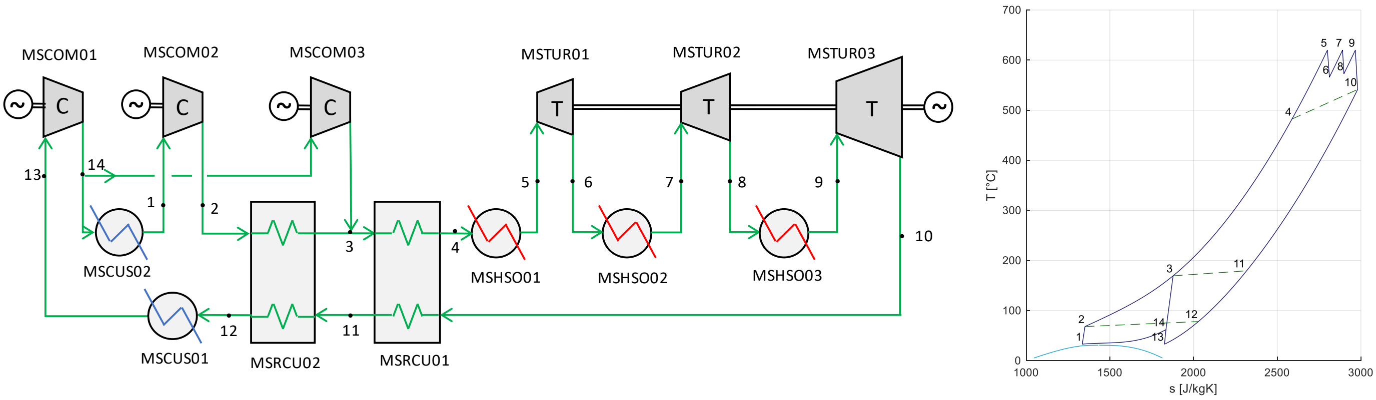

Figure 2.2 – Cycle layout and Ts diagram of the partial cooling configuration with double reheat (23).

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Table 2.1 – Main results comparison between D1.3 and MATLAB code implemented by POLIMI in D5.1 for cycle #16 and cycle #23.

Cycle		*#16*		*#23*	
Author		*D1.3*	*POLIMI*	*D1.3*	*POLIMI*
Split ratio*	-	0.642	0.642	0.567	0.567
Total CO_2 mass flow	*kg/s*	271.8	271.3	189.3	189.4
Total expansion power	MW_e	38.61	38.53	38.03	38.05
Total compression power	MW_e	-13.56	-13.53	-13.04	-13.05
Total heating duty	MW_{th}	59.15	59.03	56.26	56.29
Total cooling duty	MW_{th}	32.18	32.13	29.41	29.42
Boiler inlet temp.	*°C*	192.1	192.2	482.4	482.4
Cycle efficiency	***%***	**42.35**	**42.35**	**44.41**	**44.41**

*= ratio between the amount of mass flow rate elaborated by the main compressor (thermodynamic stream 1 in the cycle layouts) and the total amount circulating in the plant.

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3. Sensitivity analyses

3.1. Methodology

In order to understand the effect of the different assumptions on the cycle thermodynamic efficiency, a sensitivity analysis is carried out for the configurations selected in D1.3 report: the recompression cycle with HTR bypass (#16) and the partial cooling cycle with double reheat (#23) configuration.

The main cycle parameters varied during the analysis are:

1. turbomachinery isentropic efficiency;
2. heat exchangers pressure drops;
3. recuperators pinch point temperature difference;
4. cycle maximum temperature (namely the turbine inlet temperature);
5. cycle minimum temperature (namely the main compressor inlet temperature).

All the simulations have been performed with D1.3 assumption regarding the mechanical and electrical losses.

The sensitivity analysis on the design criteria or the maximum performance of the cycle components (items 1, 2 and 3 in the list above) is reported in Paragraph 3.2 for both cycle configurations. These parameters have been selected considering that they are the main input parameter for all the optimization calculations carried out in WP1. In particular, the combined effect of different compressor and turbine efficiency can dramatically affect the cycle performance driving the optimization algorithm towards different optimal pressure ratios and different optimal cycle configurations. Recuperators minimum temperature difference plays a relevant role as well in the techno-economic optimization of the system: increasing its value allows to reduce the system cost but leads to lower performance because of the larger irreversibility of the heat recovery process. Finally, pressure drops may strongly affect cycle performance considering that sCO_2 power plants are closed gas cycles where compressor consumption is a relevant share of turbine power output.

The sensitivity analyses on the cycle maximum temperature and the cycle minimum temperature, even enabling working fluid condensation, are reported in Paragraph 3.3 and Paragraph 3.4, respectively. The large effect of these two parameters on cycle efficiency is motivated considering the sCO_2 cycle configuration that is basically a closed gas cycle with marked real gas effects across the main compression. Increasing the maximum temperature always leads to an increase of turbine specific work for a fixed pressure ratio thus leading to an efficiency increase. On the contrary, reducing the cycle minimum temperature involves an increase of fluid density and a lower consumption of the main compressor. Finally, if the minimum temperature drops below the working fluid critical temperature the CO_2 condenses, allowing to further increase system efficiency but involving the use of a pump instead of a gas compressor.

A more detailed analysis on the effect of recuperator cold end temperature difference on cycle performance and heat exchanger sizing (in terms of metal mass) is reported in Paragraph 3.5.

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3.2. Sensitivity analysis on components' design parameter

A first sensitivity analysis has been carried out acting on the first three parameters representing the design criteria or the performance of the cycle components while maintaining the same cycle maximum and minimum pressures and temperatures. This approach can be justified considering that small variations will not affect appreciably the optimal value of cycle pressure ratio. With respect to the assumptions adopted in D1.3, more conservative values for each parameter have been assumed considering a moderate penalization (***cases A***) or a severe penalization (***cases B***). In these analyses only one parameter is varied while the other values are kept equal to D1.3 assumption. Moreover, two last cases are computed with the combined effect of all the ***Case A*** parameters and all the ***Case B*** parameters in order to evaluate a penalized and a very pessimistic case.

Results

Table 3.1 reports the results of the sensitivity analysis for the recompression cycle with HTR bypass (#16) configuration, while Table 3.2 reports the results of the sensitivity analysis for the partial cooling cycle with double reheat (#23) configuration.

Results are also graphically displayed in Figure 3.1 and Figure 3.2 for cycle #16 and cycle #23, respectively.

From the results obtained, the following general observations can be highlighted:

- Pressure drops moderately affect the plant efficiency: increasing the relative pressure drops in the recuperators from 0.5% up to 2% (with respect to the inlet pressure, a common value for gas/vapor heat exchangers) results in a reduction of 1.63% points of efficiency for cycle #16 and equal to 1.34% for cycle #23. Moreover, it is not necessarily true that relative pressure drops should be equal on both cold and hot sides. A similar result is obtained for the primary heat exchanger (namely, the boiler comprehensive of all the reheat stages) pressure drops.
  A more detailed heat exchangers design should be carried out to properly compute the pressure drops in the different plant heat exchangers.

- Pinch point temperature difference in recuperators plays a relevant role in cycle optimization. The value assumed in D1.3 equal to 10°C is relatively low for a gas/gas heat exchanger considering the attainable values of global heat transfer coefficient. With a 20°C assumption the efficiency is reduced by 2.21% points of efficiency for cycle #16 and equal to 1.68% for cycle #23.

- The effect of the compressor isentropic efficiency is the same for both the main (MSCOM01 for cycle #16 and MSCOM02 for cycle #23) and the secondary compressor (MSCOM02 for cycle #16 and MSCOM03 for cycle #23). The most pessimistic assumption on each of the two turbomachinery leads to a reduction of efficiency of around 0.46% points of efficiency for cycle #16.
  For cycle #23 the compressor efficiencies have been varied simultaneously for the main, the secondary and the pre-compressor, leading to a decrease of around 1% points of cycle efficiency for a reduction of 5% of the nominal values of efficiency of the three compressors.

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- Turbine isentropic efficiency is the parameter that mostly affects the plant performance. A reduction of turbine efficiency equal to 5% of its nominal value leads to a reduction of almost 2% percentage point of cycle efficiency.

- The combined effect of case A and case B assumptions shows remarkable drops in cycle efficiency: for the recompression cycle with HTR bypass (#16) the efficiency drops down to **39.10%** (combined ***case A***) and **35.11%** (combined ***case B***) while for the partial cooling cycle with double reheat (#23) down to **41.56%** (combined ***case A***) and **38.15%** (combined ***case B***).

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Table 3.1 – Results of the sensitivity analysis for the recompression cycle with HTR bypass (#16).

Pressure drops along the PHE				
		base case	***A (x1.5)***	***B (x2)***
Δp_{PHE}	*[kPa]*	250	375	500
η_{cycle}	*[%]*	**42.35**	**42.20**	**42.04**
$\Delta\eta_{cycle}$	*[%]*	**0**	**-0.16**	**-0.31**

Pressure drops along the recuperators (LTR and HTR)				
		base case	***A (x2)***	***B (x4)***
$(\Delta p/p)_{REC}$	*[%]*	0.5	1	2
η_{cycle}	*[%]*	**42.35**	**41.83**	**40.72**
$\Delta\eta_{cycle}$	*[%]*	**0**	**-0.52**	**-1.63**

Pinch point temperature difference of the recuperators (LTR and HTR)				
		base case	***A (x1.5)***	***B (x2)***
$\Delta T_{pp,REC}$	*[°C]*	10	15	20
η_{cycle}	*[%]*	**42.35**	**41.22**	**40.14**
$\Delta\eta_{cycle}$	*[%]*	**0**	**-1.13**	**-2.21**

Isentropic efficiency of the main compressor (MSCOM01)				
		base case	***A (-2.5%)***	***B (-5%)***
$\eta_{MSCOM01}$	*[%]*	82.17	80.12	78.06
η_{cycle}	*[%]*	**42.35**	**42.12**	**41.88**
$\Delta\eta_{cycle}$	*[%]*	0	-0.23	-0.47

Isentropic efficiency of the secondary compressor (MSCOM02)				
		base case	***A (-2.5%)***	***B (-5%)***
$\eta_{MSCOM02}$	*[%]*	82.44	80.38	78.32
η_{cycle}	*[%]*	**42.35**	**42.12**	**41.88**
$\Delta\eta_{cycle}$	*[%]*	**0**	**-0.23**	**-0.48**

Isentropic efficiency of the turbine (MSTUR01)				
		base case	***A (-2.5%)***	***B (-5%)***
$\eta_{MSTUR01}$	*[%]*	84.3	82.19	80.09
η_{cycle}	*[%]*	**42.35**	**41.42**	**40.46**
$\Delta\eta_{cycle}$	*[%]*	**0**	**-0.93**	**-1.89**

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Table 3.2 – Results of the sensitivity analysis for the partial cooling cycle with double reheat (#23).

Pressure drops along the PHE				
		base case	***A (x1.5)***	***B (x2)***
Δp_{PHE}	*[kPa]*	250	375	500
η_{cycle}	*[%]*	**44.41**	**44.28**	**44.15**
$\Delta\eta_{cycle}$	*[%]*	**0**	**-0.13**	**-0.26**

Pressure drops along the recuperators (LTR and HTR)				
		base case	***A (x2)***	***B (x4)***
$(\Delta p/p)_{REC}$	*[%]*	0.5	1	2
η_{cycle}	*[%]*	**44.41**	**43.97**	**43.07**
$\Delta\eta_{cycle}$	*[%]*	**0**	**-0.44**	**-1.34**

Pinch point temperature difference of the recuperators (LTR and HTR)				
		base case	***A (x1.5)***	***B (x2)***
$\Delta T_{pp,REC}$	*[°C]*	10	15	20
η_{cycle}	**[%]**	**44.41**	**43.55**	**42.73**
$\Delta\eta_{cycle}$	**[%]**	**0**	**-0.85946**	**-1.68**

Isentropic efficiency of the compressors (MSCOM01, MSCOM02, MSCOM03)				
		base case	***A (-2.5%)***	***B (-5%)***
$\eta_{MSCOM02}$	*[%]*	81.57	79.53	77.49
$\eta_{MSCOM03}$	*[%]*	75.41	73.53	71.64
$\eta_{MSCOM01}$	*[%]*	82.55	80.48	78.42
η_{cycle}	*[%]*	**44.41**	**43.93**	**43.43**
$\Delta\eta_{cycle}$	*[%]*	**0**	**-0.48**	**-0.98**

Isentropic efficiency of the turbines (MSTUR01, MSTUR02, MSTUR03)				
		base case	***A (-2.5%)***	***B (-5%)***
η_{MSTUR01}	*[%]*	86.10	83.95	81.80
η_{MSTUR02}	*[%]*	88.60	86.39	84.17
η_{MSTUR03}	*[%]*	89.20	86.97	84.74
η_{cycle}	*[%]*	**44.41**	**43.48**	**42.52**
$\Delta\eta_{cycle}$	*[%]*	**0**	**-0.93**	**-1.89**

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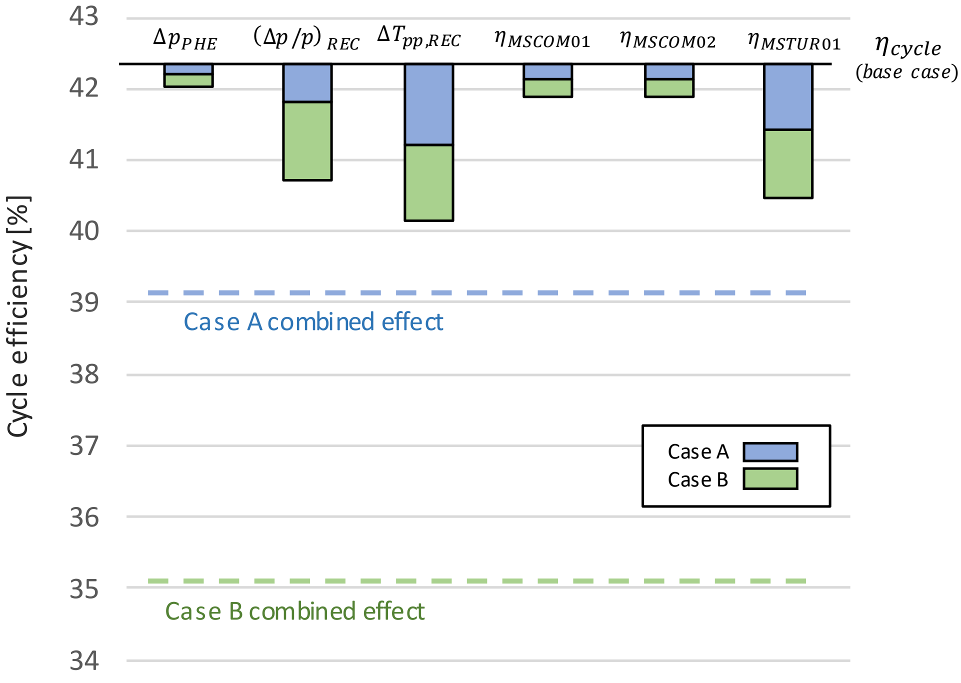

Figure 3.1 – Cycle efficiency variation for the recompression cycle with HTR bypass (#16) assuming moderate (case A) and severe (case B) penalization on main components' assumption.

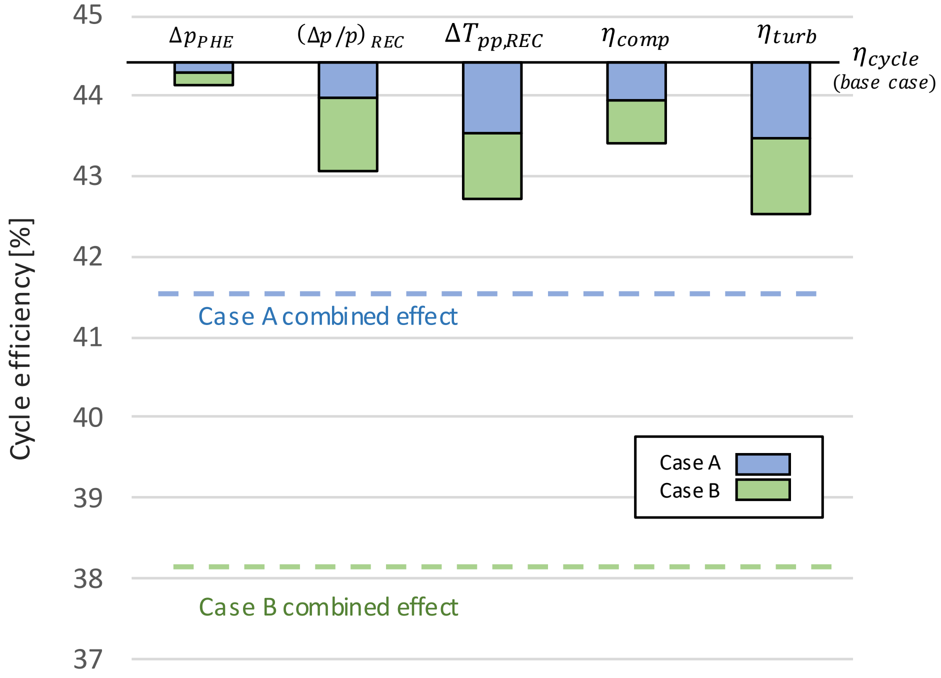

Figure 3.2 – Cycle efficiency variation for the partial cooling cycle with double reheat (#23) assuming moderate (case A) and severe (case B) penalization on main components' assumption.

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3.3. Maximum cycle temperature effect on cycle efficiency

Increasing the cycle maximum temperature potentially leads to a relevant increase of cycle efficiency allowing to increase the difference between turbine and compressors specific work. However, varying this parameter while maintaining the same pressure ratio of the reference case leads to suboptimal solutions and it is thus necessary to carry out a new cycle optimization for each value of cycle maximum temperature. Adopting the turbomachinery efficiency of the reference case in this procedure may lead to inconsistent results with solutions that exclude the less efficient component. For these reasons, the results are obtained adopting a mean value of compressor and turbine efficiency calculated as the average of the expanders and compressors efficiency adopted in D1.3 for cycle #16 and cycle #23, respectively. Adopted values are reported in Table 3.3. All the other assumptions are equal to D1.3 assumptions.

It has to be noted that for this reason even the base case efficiency is different from the values of D1.3.

Table 3.3 – Mean turbomachinery efficiencies adopted in the sensitivity analysis on maximum temperature, minimum temperature and condensation for cycle #16 and #23.

Turbomachinery mean efficiencies			
		Cycle #16	Cycle #23
$\eta_{compressors}$	[%]	82.31	79.84
$\eta_{turbines}$	[%]	84.30	87.87

Table 3.4 and Table 3.5 report some specific numerical result for cycle #16 and #23 respectively while Figure 3.3 depicts the trend of cycle efficiency as a function of the turbine inlet temperature for both configurations. Increasing the cycle maximum temperature leads to an efficiency increase of about 0.4 percent points for each ten degree of temperature increase. The target cycle efficiency equal to 48% can be reached for a maximum temperature of 760°C for the recompression cycle with HTR bypass (#16), while for the partial cooling with double reheat configuration (#23) a temperature of 710°C would be necessary.

Table 3.4 – Results of the sensitivity analysis on the cycle maximum temperature for the recompression cycle with HTR bypass (#16).

Turbine inlet temperature							
		-40 °C	**-20 °C**	**base case**	**+30 °C**	**+80 °C**	**+130 °C**
T_{max}	[°C]	580	600	620	650	700	750
η_{cycle}	[%]	40.32	41.27	42.19	43.50	45.54	47.42
$\Delta\eta_{cycle}$	[%]	-1.86	-0.91	0	+1.31	+3.35	+5.24

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Table 3.5 – Results of the sensitivity analysis on the cycle maximum temperature for the partial cooling cycle with double reheat (#23).

Turbine inlet temperature							
		-40 °C	**-20 °C**	**base case**	**+30 °C**	**+80 °C**	**+130 °C**
T_{max}	*[°C]*	580	600	620	650	700	750
η_{cycle}	*[%]*	42.75	43.66	44.54	45.80	47.79	49.63
$\Delta\eta_{cycle}$	*[%]*	-1.79	-0.88	0	+1.27	+3.25	+5.09

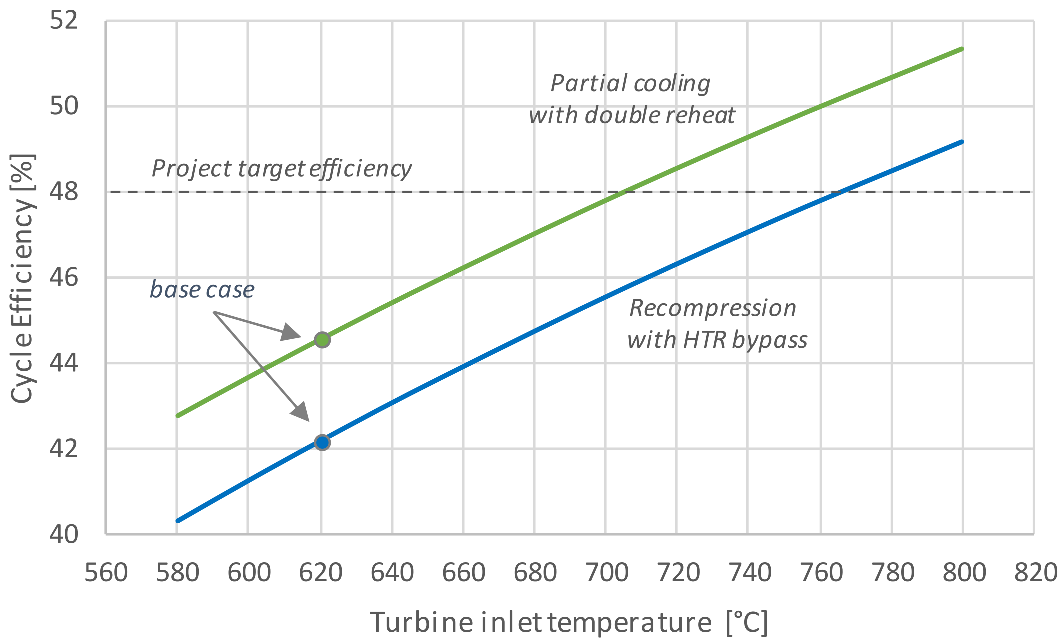

Figure 3.3 - Trend of cycle efficiency as a function of the cycle maximum temperatures (turbine inlet temperature) for the recompression cycle with HTR bypass (#16, blue line) and for the partial cooling cycle with double reheat (#23, green line).

It must be noted that in this analysis the boiler efficiency, which greatly influences the effective plant efficiency, it is not considered. Maintaining the same pressure levels and turbomachinery efficiency, an increase of the turbine inlet temperature leads to an increase in the turbine outlet temperature and consequently an increase on the boiler inlet temperature due to the higher heat recovery in the HTR regeneration process. This fact would lead to a decrease of performances of the boiler together with additional technological issues related to the material selection. A further analysis is required in order to asses properly this aspect by developing a proper boiler model to be included into the analysis.

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3.4. Minimum cycle temperature effect on the cycle efficiency

Decreasing the cycle minimum temperature potentially leads to a relevant increase of cycle efficiency allowing to exploit more marked real gas effects or possibly exploit the working fluid condensation with the aim of reducing main compressor specific work. However, as for the previous case, varying this parameter while maintaining the same pressure levels of the reference case leads to suboptimal solutions and it is thus necessary to carry out a new cycle optimization for each value of cycle minimum temperature. For example, in the recompression cycle with HTR bypass (#16), maintaining the same pressure levels of D1.3 and increasing the cycle minimum temperature from 33°C to 35°C leads to an increase of the main compressor work of around 2.69 MW_{el} (5.20 MW_{el} against 7.89 MW_{el}, considering a 25 MW_{el} cycle). This is mainly due to the behaviour of CO_2 in the proximity of the critical point and the sharp variation of thermodynamic properties because of real gas effects. In this example, the variation of 2°C leads to a strong variation of CO_2 density at the main compressor inlet (from 609.6 kg/m^3 to 405.41 kg/m^3) and the consequent specific work increase. Even in this case it is convenient to adopt an isentropic efficiency value equal for all the turbines and an isentropic efficiency value equal for all the compressors. Adopted values are reported in Table 3.3. All the other assumptions are equal to D1.3.

Table 3.6 and Table 3.7 report some specific numerical results for cycle #16 and #23 respectively while Figure 3.4 depicts the trend of cycle efficiency as a function of the main compressor inlet temperature for both configurations. Decreasing the cycle minimum temperature leads to an increase of about 0.3 percent points of efficiency for each degree of temperature reduction. As a result, the target cycle efficiency (equal to 48%) can be reached only adopting a condensing cycle for the partial cooling with double reheat configuration (#23) with an extremely low minimum cycle temperature (21°C) while this value of cycle efficiency cannot be achieved for the recompression cycle with HTR bypass configuration (#16).

Table 3.6 – Resume of the results of the sensitivity analysis on the cycle maximum temperature for the recompression cycle with HTR bypass (#16).

Cycle minimum temperature							
		-10 °C	**-5 °C**	**-1 °C**	**base case**	**+1 °C**	**+2 °C**
T_{min}	*[°C]*	23	28	32	33	34	35
η_{cycle}	*[%]*	44.61	43.46	42.44	42.19	41.93	41.67
$\Delta\eta_{cycle}$	*[%]*	+2.42	+1.28	+0.26	0	-0.26	-0.52

Table 3.7 – Resume of the results of the sensitivity analysis on the cycle minimum temperature for the partial cooling cycle with double reheat (#23).

Cycle minimum temperature							
		-10 °C	**-5 °C**	**-1 °C**	**base case**	**+1 °C**	**+2 °C**
T_{min}	*[°C]*	23	28	32	33	34	35
η_{cycle}	*[%]*	47.49	46.08	44.83	44.54	44.25	43.97
$\Delta\eta_{cycle}$	*[%]*	+2.96	+1.55	+0.29	0	-0.29	-0.57

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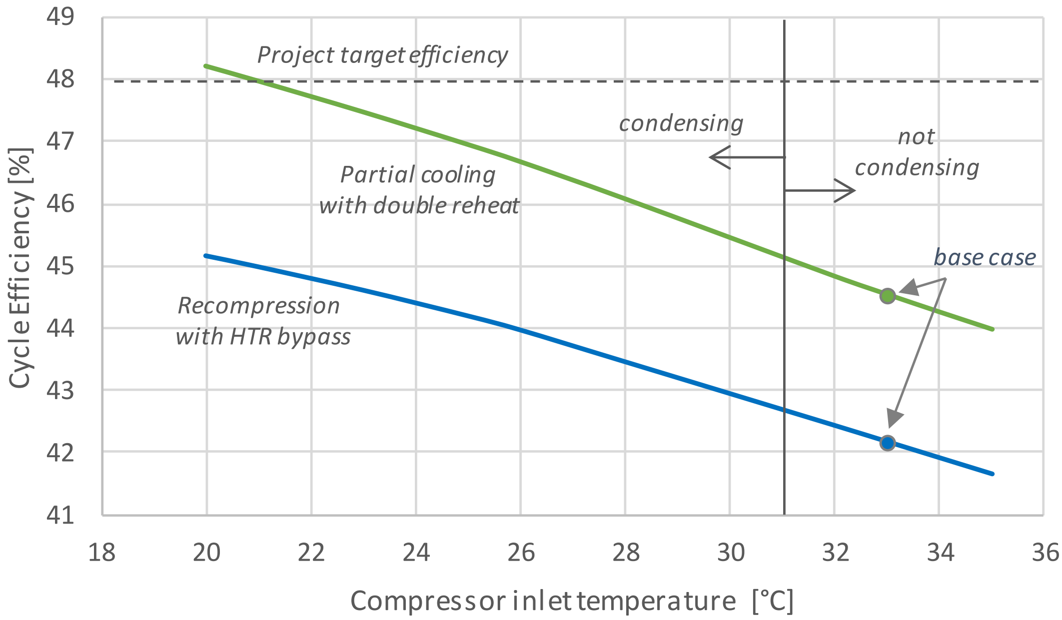

Figure 3.4 - Trend of cycle efficiency as a function of the cycle minimum temperature (main compressor inlet temperature) for the recompression cycle with HTR bypass (#16, blue line) and for the partial cooling cycle with double reheat (#23, green line).

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3.5. Effect of LTR and HTR cold end temperature differences

A further analysis has been performed in order to evaluate the individual influence of each cold end temperature difference on the cycle efficiency and on the total metal mass of the two heat exchangers (HTR and LTR).

The recuperators have been modelled as printed circuit heat exchangers (PCHE) and their total metal mass has been evaluated considering them with the methodology developed by Dostal in [3] and adopted by the authors in the conference paper “Multi Objective Optimization of Flexible Supercritical CO_2 Coal-Fired Power Plants” submitted to ASME Turbo Expo 2019.

Figure 3.5 represents the TQ diagrams for the recompression cycle with HTR bypass (#16, left) and for the partial cooling cycle with double reheat (#23, right). It is possible to notice that cycle #16 features a higher degree of recuperation as the total thermal power recuperated is 30% higher (around 130.77 MW_{th} for cycle #16 and 99.68 MW_{th} for cycle #23).

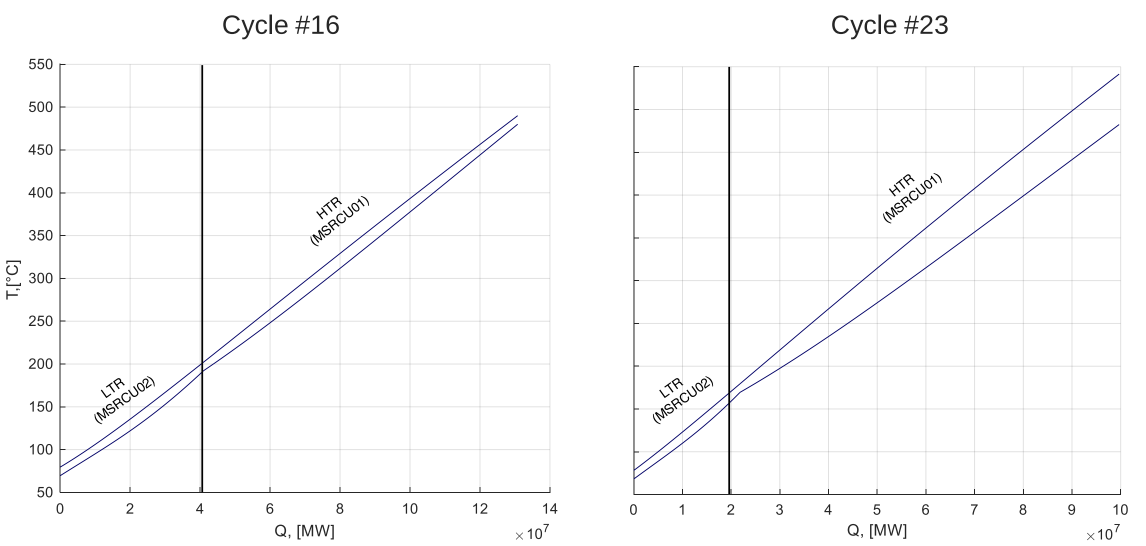

Figure 3.5 - TQ diagram of both recuperators (LTR and HTR) for the recompression cycle with HTR bypass (#16, left) and for the partial cooling cycle with double reheat (#23, right).

The analysis has been repeated varying each cold end temperature difference between 5°C and 25°C on a 5x5 grid.

Figure 3.6 depicts the results on a cycle efficiency vs. total recuperator metal mass diagram. In the recompression cycle with HTR bypass (#16), a variation of the LTR pinch point impact remarkably on the cycle efficiency (solid lines in Figure 3.6), while it has a very little influence on the total recuperators weight. On the contrary, the HTR minimum temperature difference (dashed lines in in Figure 3.6) has a strong effect on both figures of merit. With the aim at increasing cycle efficiency without increasing excessively the thermal inertia and investment cost of the system it looks promising to decrease the LTR minimum temperature difference while adopting larger values for the HTR component. A different trend is found for

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the partial cooling cycle with double reheat (#23) where the effect of the recuperator pinch point temperature difference on the cycle efficiency and the total recuperator mass is similar for both LTR and HTR components. The optimal solution in terms of trade-off between cycle efficiency and flexibility is hence less trivial and best solution may adopt intermediate values for recuperators minimum temperature difference.

Contours maps of Figure 3.7 represents the total recuperators weight (dotted lines and colormap) and the cycle efficiency (continuous lines) against the temperature difference in the LTR and HTR heat exchangers for the recompressed cycle with HTR bypass (cycle #16, on the left) and for the partial cooling cycle with double reheat (cycle #23, on the right).

The red line represents the Pareto front of optimal combination of cold end temperature differences of the LTR and HTR. Three significant points are represented on the graph: the white dot represents the base case cycle obtained with D1.3 assumptions; the orange dot represents a cycle with the same cycle efficiency but featuring the minimum total mass of the recuperators while the green dot represents a cycle with the same total mass of the recuperators but featuring the maximum attainable cycle efficiency.

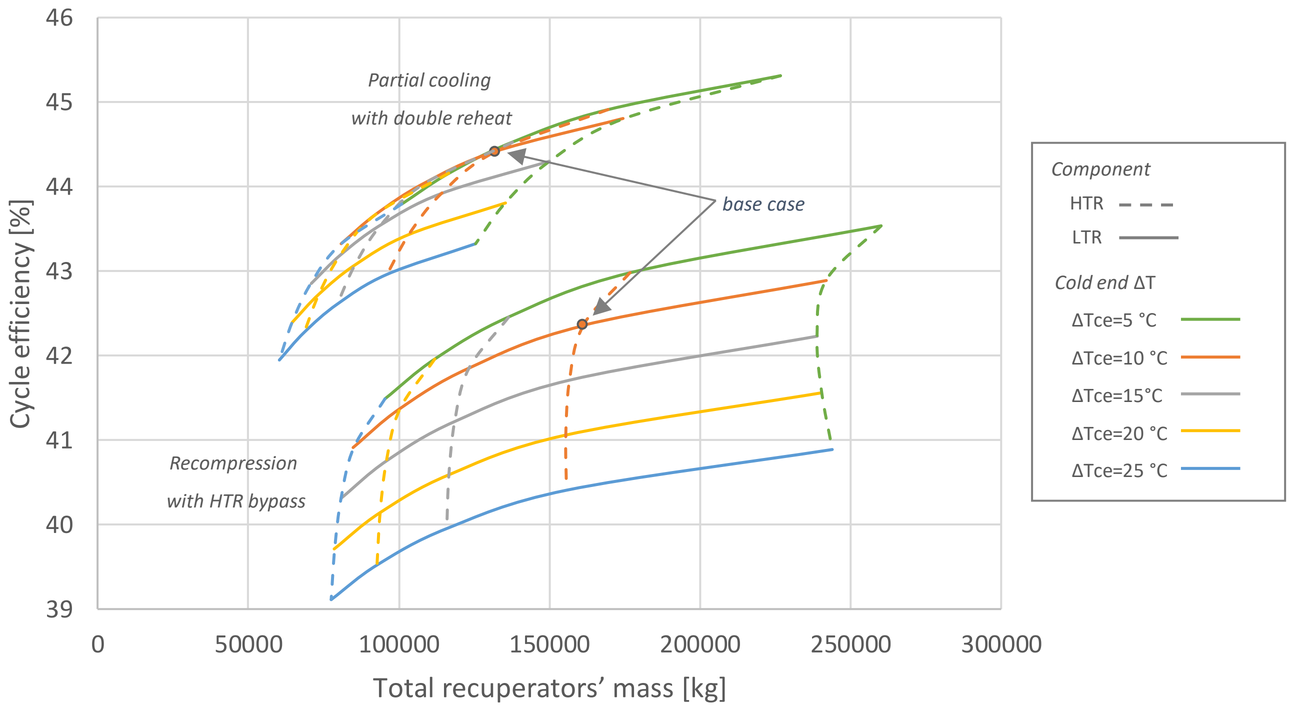

Figure 3.6 - Tradeoff analysis between cycle efficiency and total recuperator weight varying the cold end temperature differences of the LTR and HTR for the recompression cycle with HTR bypass (#16, bottom) and for the partial cooling cycle with double reheat (#23, top).

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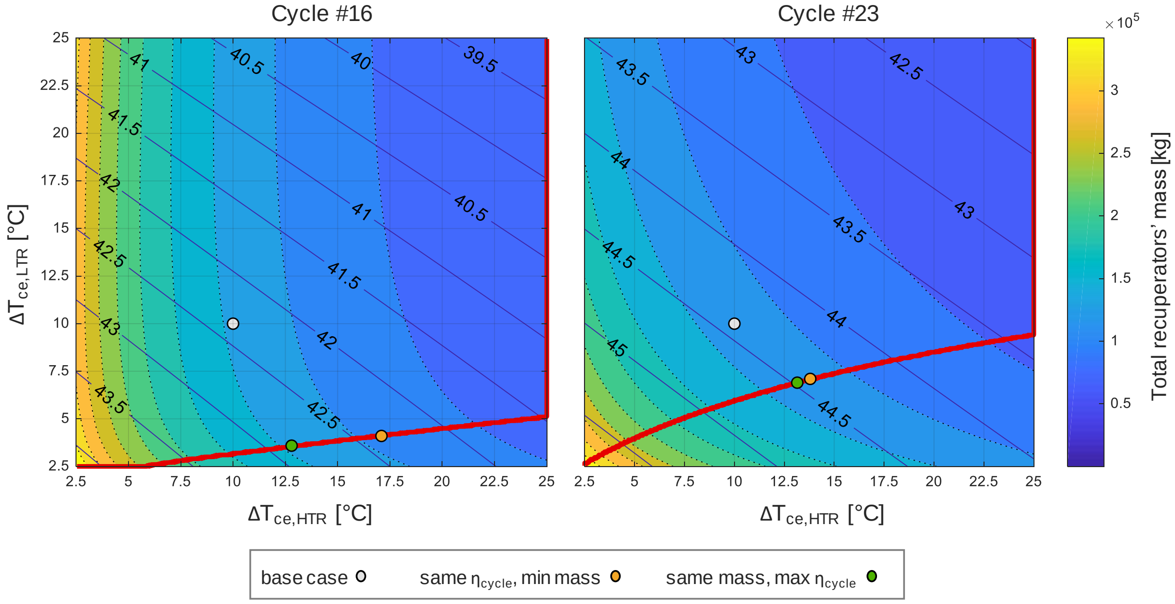

Figure 3.7 – Total recuperator weight (dotted line and color) and cycle efficiency (dashed line) varying the cold end temperature differences of the LTR and HTR for the recompressed cycle with HTR bypass (cycle #16, left) and for the partial cooling cycle with double reheat (cycle #23, right).

4. Conclusions

The main results of the sensitivity analyses carried out in this deliverable lead to a number of recommendations for the other WPs activities. The following points can be highlighted:

- For both cycle configurations, the minimum temperature difference in the recuperators can moderately affect the cycle performance but strongly affects the estimated metal mass of these components and the overall system thermal inertia. However, the results here reported are strongly affected by the assumed global heat transfer coefficient of the recuperators that is influenced by both the component's geometry and fluid velocity. These parameters also affect the recuperator pressure drops. In conclusion, further input is required by WP4 in order to establish a link among pressure drops, global heat transfer coefficient and heat exchanger geometry and dimensions.
- Turbine and compressor efficiencies strongly affect cycle performance. All the sensitivity analyses are carried out with fixed turbomachinery efficiency, but the performance of each single component is expect to vary depending on the mass flow rate, the pressure ratio and the fluid compressibility factor. Further analyses will be carried out including future inputs from WP3 in order to link the nominal turbomachinery performance as function of exogenous parameters in the optimization of the plant performance.

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- As a general observation the turbomachinery may achieve a higher performance for large scale power plants (100 MW) that should increase the plant efficiency by 2-3 points percent of efficiency. On the contrary, pressure drops adopted in these analyses for the heat exchangers look optimistic with respect to the values assumed in other studies. Further investigation is required on the link between pressure drops and global heat transfer coefficients in the recuperators.
- Increasing maximum temperature would allow to meet the project target of 48% efficiency but values higher than 700°C are required. This option should be investigated in WP2 with a dedicated study of a high temperature boiler: in fact, the increase of cycle efficiency could be counterbalanced by a reduction of boiler efficiency resulting in a higher system specific cost and inertia. Furthermore, in order to fit the numerical model on a realistic design of the boiler, more details from WP2 calculations are required on the heat exchanger arrangement and air preheating unit.
- Reducing cycle minimum temperature seems the most promising option for increasing cycle efficiency but it would require a very cold cooling medium (around 10-15°C), a main pump/compressor working from saturated liquid condition to supercritical fluid thermodynamic state and a heat rejection unit able to handle fluid condensation. The technical feasibility of this option requires further evaluations in WP3 and WP4 and will be finalized in WP6.
- In conclusion, with the thermodynamic constraints and the component performances adopted in WP1 calculations it seems difficult to hit the efficiency target of 48%. However, the sensibility analyses show that the performance target would be reached adopting a combination of different strategies: use of more efficient turbomachinery, increase of the dimension of the recuperators and pushing the cycle towards high maximum temperature and allowing condensation.

5. Bibliography

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