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Design of a sCO2-based Pumped Thermal Energy Storage (PTES) test rig integrated with industrial waste heat recovery

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ABSTRACT

In a Renewable Energy Sources (RES) driven energy scenario, where more and more bulky quantities of RES should be introduced on the grid, the role of energy storages is crucial. Further to already available electric storage technologies (mostly based on batteries), it will be mandatory to have grid flexible large scale energy storages able to operate ramp-up/down with large capacity, which behaviour/management should be as much similar as possible to traditional power plants (also to guarantee specific grid services like grid frequency regulation via rotating inertia etc.) which are currently used to instantaneously regulate the grid.

At this purpose, Pumped Thermal Energy Storage (PTES) offers GWh scale storage without geographical constraints, at reasonable costs, and implementing power and heat pump cycles integrated with thermal energy storage (TES) solutions. A PTES system indeed stores heat in two high temperature and low temperature TES units in charging phase using heat pump (HP) that operates on electricity provided by renewable energy sources (solar, wind etc.). The stored heat is used to drive a power cycle at required times. The choice of the working fluid for power cycle as well as heat pump cycles have a significant importance based on the range of storage temperature. Working fluid in PTES has direct effect on the performance, capital cost and efficiency of the whole operation.

sCO2 as a working fluid has certain aspects that makes it the ideal candidate for large scale PTES applications. sCO2 cycles are indeed fully compatible with the temperature range of TES hot storage sources and sCO2 has already been used in commercial HP solutions (even targeting high temperature HPs). In addition, sCO2 allows energy storages to embody a compact design as well, making the whole PTES footprint smaller compared to technologies based on other working fluids. This is something that has been already acknowledged by some R&D experiences in US and EU even at industrial level (like ENERGY DOME, ECHOGEN or MAN), but certainly not taking full advantage of the clear peculiarities of sCO2 in terms of compact and flexible machines as well as not considering to integrate another typical application of sCO2 power cycle: waste-heat recovery.

In the EU-Funded SCO2OP-TES project (which starts from authors' conceptual idea) it is proposed to fully integrate Waste heat valorisation (from thermal power plants or industries, also in temperature ranges even at T<150-200° not fully exploitable for waste-heat-to-power applications) with a sCO2 PTES cycle featuring a single TES solution, able to store electricity with attractive Round Trip Efficiency values thanks to a Waste Heat driven power-to-heat-to-power approach.

The concept will be studied both via modelling and experimental activities: this paper introduces therefore the project objectives and implementation plan, then focuses on the results derived from modelling and design activities in specific relation to the conceptual design of the sCO2

PTES test rig that University of Genova will realize and operate in the project. This test rig will enable to validate a WH-driven PTES solution based on sCO2 cycles (simple and recuperated HP and power cycles, via a double set of "hot and cold" sCO2 turbomachinery - two compressors and two expanders) where a ~ 220-250°C WH stream from a local power plant will be integrated. This paper presents the features of this test rig and the modelling activities that defined them including the proposed integration and operational regimes too, expected thermodynamic performance at nominal point, and up-scaling considerations.

INTRODUCTION

Energy systems are experiencing world-wide an unstoppable increase of renewable power generation (RES) in their electrical grids, with RES plant generation such as wind, solar, or hydro producing a large quantity of electricity that exceeds demand and that needs to be stored for later use. These resources are sometimes curtailed or sold with negative prices on the grid, thus wasting clean energy or cutting revenues for energy providers. In parallel, a continuous RES penetration as well as the electrification of different processes (from heating and cooling to electromobility and industrial processes) is making transmission lines becoming difficult to be balanced, congested . According to all these aspects, the development of large-scale long-duration (>8 h) energy storage (LDES) technologies will play a crucial role in clean energy future [1] to target weekly/seasonal energy storage and the shifting of RES production as well as to provide the flexibility needed on the grid. For all above mentioned purposes, power-to-heat-to-power solutions based on turbomachinery and thermal energy storages (also known as Carnot batteries [2]) can be a promising long-duration energy storage (LDES) technology offering large energy and power storage capacity, large storage cyclability, rapid response time, no dependency on geographical location (as pumped hydro storage), and durations from 6–12 h up to several weeks providing the flexibility needed for generators and transmission operators [3] [4].

The need of high temperature Heat Pump (HP) cycles as well as reduction in CAPEX (e.g., via compact and reversible machines) has made sCO_2 a promising working fluid for Pumped Thermal Energy Storage PTES [5].

The possibility to couple sCO2 HPs and power cycles for bulky energy storage in Carnot batteries, while integrating external heat inputs (e.g., from Concentrated Solar Power–CSP [6] or Waste Heat [7]), was recently investigated and could bring to the promotion of Thermally Integrated Pumped Thermal Energy Storage (TI-PTES), enabling the possibility to increase PTES electrical Round Trip Efficiency (RTE) and reducing CAPEX (e.g., avoiding the need of "cold TES" for example), valorizing freely available heat sources [8] [9].
The concept of TI-PTES has been already investigated [7], [8], [9] , [10], nevertheless particularly looking at sCO2 TI-PTES, no experimental facility has been developed so far and sCO2 PTES experimental facilities are still quite rare.

The goal of this paper is to present the results of some preliminary modelling and design activities towards a first of its kind laboratory of sCO2 TI-PTES system harvesting a 250°C Waste Heat (WH) source from a local power plant.

CONCEPT PRESENTATION

PTES systems are thermomechanical energy storage that rely on two separated thermodynamic cycles, one heat pump, or charging cycle, and one traditional heat engine/power cycle, or discharging cycle. Their alternate utilization allows to store electrical energy in form of heat in one or two thermal energy storages (TES), and then release this energy back into the electrical form, thus relying on traditional power plant technologies. The final net effect is equivalent to an electrical energy storage.

Since many losses can be accumulated during energy conversions, the efficiency of these energy storage systems rely on the minimization of first and second principle losses of each component. One way to improve the apparent RTE, by boosting the power output or reducing the power input, is to integrate it with a freely available heat source (e.g. a waste heat source), as shown in Figure 1. In this case, the charging cycle receives a thermal energy input from a waste heat source and upgrades its temperature storing electrical energy, which is then released through a discharging cycle exchanging with the ambient for the low temperature sink (air cooler). In this way, only one energy storage is required, at high temperature, and at the same time it is possible to decouple the thermodynamic points of the two cycles. The decreased temperature and pressure difference in the heat pump cycle enhance the COP of the charging cycle, and possibly allows an higher temperature of the turbine during the discharging cycle, leading to a higher apparent RTE. In such a case of thermally-integrated PTES, where additional heat is absorbed from external resource, the apparent electrical RTE is a performance indicator that might be misleading, and an exergy-based RTE should be preferred: this aspect has been discussed by the authors in [9]. However, assuming a freely available WH, the apparent RTE can also make sense from an energy utility/electric market application perspective: this perspective is assumed in this paper.

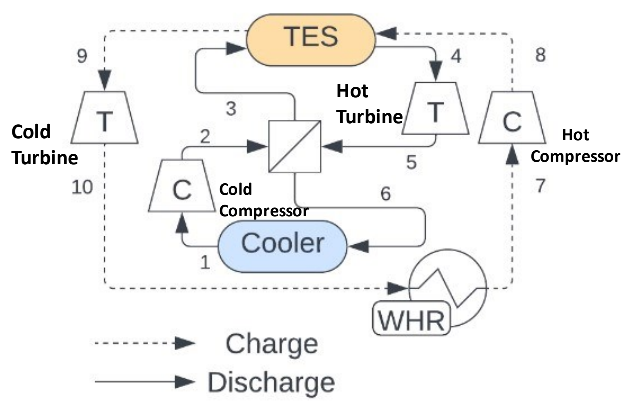

Figure 1 –sCO2 based TI-PTES conceptual layout (in this case integrate with WH recovery)

RESULTS AND DISCUSSION

In the framework of Tirreno Power Vado Ligure Combined Cycle Power Plant, a WH stream at around 250°C (auxiliary steam from the power plant) could be exploited as freely available heat. This context is the validation site of the EU funded SCO2OP-TES project [11] where a consortium composed by different EU research centres and technology manufacturers is promoting the realization and validation of a 100 kWel prototype of this TI-PTES system, able to validate the overall project concept as well the control and modelling tools developed in the project. At this purpose, considering some constraints (at laboratory and budget level) in terms of energy storage size, machine operating pressure (up to 200 bar) and temperature (WH source at 250°C, TES material up to 400°C), the authors investigated the design of the laboratory setup, studying charging and discharging thermodynamic cycles via the proprietary WTEMP-EVO, a component-based in-house thermo-economic simulation tool [9], [10].

Methodology

WTEMP-EVO is developed in MATLAB®, integrating Coolprop [12] libraries for fluid properties, and it can simulate energy systems through the assembly of the desired layout, as explained in [13]. The tool (which workflow is presented in Figure 2) evolves the solution of each component using simple characteristic equations for mass and energy balances, and pressure computation; some of them are reported in the followings.

$$p_{Out} = p_{In} \cdot \beta_{Compr} \tag{1}$$

$$p_{Out} = p_{In} * (1 - \Delta p_{\%Loss}) \tag{2}$$

$$h_{Out} = h_{In} + \eta_{Turb} \cdot (h_{Out-isoentr} - h_{In}) \tag{3}$$

$$h_{Out} = h_{In} + {(h_{Out-isoentr} - h_{In})}/{\eta_{Comp}} \tag{4}$$

$$\varepsilon_{HEX} = {Q_{HEX}}/{Q_{HEX-max}} \tag{5}$$

$$Q_{HEX} = \dot{m}_{cold} * (h_{Out-cold} - h_{In-cold}) \tag{6}$$

Once the desired cycle layout is defined, it is assembled by calling the functions corresponding to the necessary components in the layout (turbomachinery, heat exchangers, etc), and a system of nonlinear equations is formed accordingly. Then, some of the variables are set, accordingly to the assumptions, to define the degrees of freedom of the layout, and the system of equations is solved numerically until convergence is achieved.

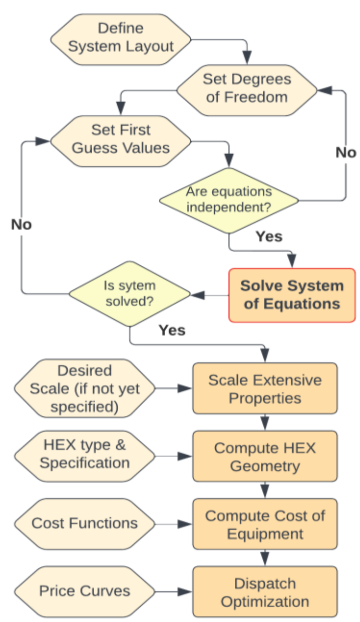

Figure 2 - Algorithm flowchart of the TEMP-EVO tool

Different solvers of the thermodynamic problem are available in the tool, mainly relying on the MATLAB predefined algorithms. The base algorithm consists of an implementation of the Newton-Raphson root-finding method, to take advantage of the speed of the algorithm. However, the initial iteration guess of the variables plays an important role and can lead to non-convergence of the system of equations, especially for large and highly-nonlinear systems. Thus, the implementation of a wiser definition of the initial guesses, the limitation of the iterative step in the method, and the

possibility of splitting the system into subsequently independent subsystems, can improve the convergence. If none of these modifications is sufficient to guarantee the convergence to the solution in the specific case, then a switch to other algorithms is implemented, with the possibility to choose between root-finding ones, like those in the MATLAB function "fsolve", or a more general minimization of the errors.
Following the thermodynamic resolution of the cycle, it is possible to compute the geometry and the cost of the main equipment necessary to realize the layout, as described in [13].

Preliminary lab design

A preliminary design of the pilot plant laboratory (Figure 3 – Table 1) has been performed showing that for a mass flowrate of around 0.9-1 kg/s the target of a "hot compressor" able to elaborate sCO2 in the required operating conditions on an electric power of 100 kWel was satisfied.
Starting from this first design, a sensitivity analysis has been performed also in order to investigate which could be the component of the prototype where manufacturers should focus their efforts towards RTE maximization.
Scope of the analysis was, therefore, first to study some possible design conditions of the pilot plant and then to make a sensitivity analysis on how the system performance is affected by machinery efficiencies and heat exchanger effectiveness (and related size), in both charging and discharging cycles.

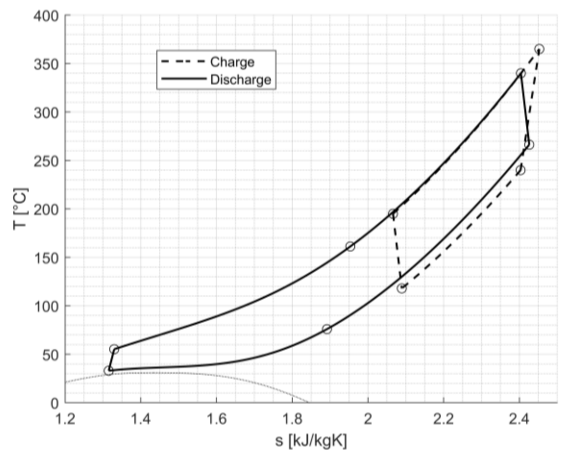

Figure 3 – T-S diagram of the reference cycle of the TI-PTES test rig

Table 1 – Operating points of the reference cycle of the TI-PTES test rig

	p [bar]	T [°C]	m [kg/s]
1	83	33	0.9
2	183.65	54.8	0.9
3	181.82	161.07	0.9
4	180	340	0.9
5	84.7	266.3	0.9
6	83.83	75.3	0.9
7	77,5	240	0.9
8	200	365	0.9
9	198	183.3	0.9
10	78.3	103	0.9

Component	Power [kW]
Compressor CC	107.6 kW_{el}
Turbine CC	43.87 kW_{el}
Compressor DC	15.8 kW_{el}
Turbine DC	60.2 kW_{el}
Hot TES Primary HEX	209.5 kW_{th}
Cold TES HEX	164.5 kW_{th}
WHR	144.5 kW_{th}
Recuperator	209.6 W_{th}

Assumptions

A sensitivity analysis is then performed calling the solver of the system with a grid-type variation of the chosen sensitivity parameters. A detailed list of the fixed parameters as well as sensitivity ranges is reported in Table 2. While superior pressure is fixed up to 200 bar (mostly for safety reasons), lower pressure has been kept as a variable in the sensitivity analysis and it will be defined following previous authors' experience [9][10] in order to operate in supercritical conditions and maximise cycles performances

Table 2 – Assumptions

Parameter	Unit	Value
TES max T	350	°C
Cycles superior pressure	200	bar
TES min T	250	°C
WH Inlet T	250	°C
min T difference in HEX	10	°C
Ambient temperature	25	°C
Pressure losses	1	%
TES mass flow	1	kg/s
Hot Compressor Efficiency	60-85	%
Cold Compressor Efficiency	70-85	%
Hot Turbine Efficiency	70-90	%
Cold Turbine Efficiency	30-60	%

The most relevant KPI that has been investigated (as key parameter for any type of storage, but particularly for LDES) is the round-trip efficiency, electrical only as previously discussed, defined as it follows:

$$RTE_{el} = \frac{E_{DC}}{E_{CC}} = \frac{P_{DC} \cdot t_{DC}}{P_{CC} \cdot t_{CC}} = COP \cdot \eta \tag{7}$$

As it can be seen in Table 2, different turbomachinery isentropic efficiency ranges has been considered between charging and discharging cycles.

For what it concerns the charging cycle, indeed, the laboratory will foresee the installation of a radial "hot compressor" (η in the range 60-85%) and of a bladeless "cold turbine" (η in the range 30-60%), While for the discharging cycle a volumetric "cold compressor" (η in the range 70-85%) and a centrifugal "hot turbine" (η in the range 70-90%) will be considered.

Sensitivity analysis on sCO2 machines isentropic efficiency

This section deals with the sensitivity analysis of the machinery involved in the charging and the discharging cycle. The sensitivity plots are made varying two turbomachine efficiencies at a time.

<u>Charging Cycle</u>

The charging cycle is first studied for the sensitivity of the thermodynamic parameters on hot compressor and cold turbine efficiency. As the high pressure side is fixed at 200 bar, for the change in isentropic efficiencies of the turbomachinery in the charging cycle there is a variation

of lower pressure side of the cycle to compensate for the energy loss or gain because of the variation of isentropic efficiencies. The whole sensitivity analysis indeed has been performed considering a fixed heat input coming from the hot TES thus charging and discharging cycles compensate by changing their lower pressures due to the variations of machines' efficiencies and related variations of operating temperatures.
In this sense Figure 4 shows that variation in low pressure side with respect to the hot compressor isentropic efficiency. The low side pressure values are equally sensitive to the cold turbine efficiency variation as the change in the isentropic efficiency of either of the turbomachines leads to the change in the low side pressure . However, the hot compressor efficiency is plotted here with the waste heat max temperature, which is the inlet temperature of the waste heat source. Although the waste heat temperature is fixed at 250°C, but when running the sensitivity analysis, there is a slight change in the waste heat temperatures for the simultaneous changes in the turbomachinery of the discharging cycle, however as the figure shows that the system pressure ratio is still independent of the small variation of waste heat maximum temperature. The pressure ratio is designed independently from the WH temperature, but is sensitive to the variations to the turbomachinery performance. Keeping the superior pressure constant (200 bar) the higher efficiency of the compressor leads to smaller low side pressures values and vice-versa, showing 84 bar low side pressure for 85% efficiency and around 106 bar for 60% efficiency. Therefore, more efficient compressors require larger pressure ratios.

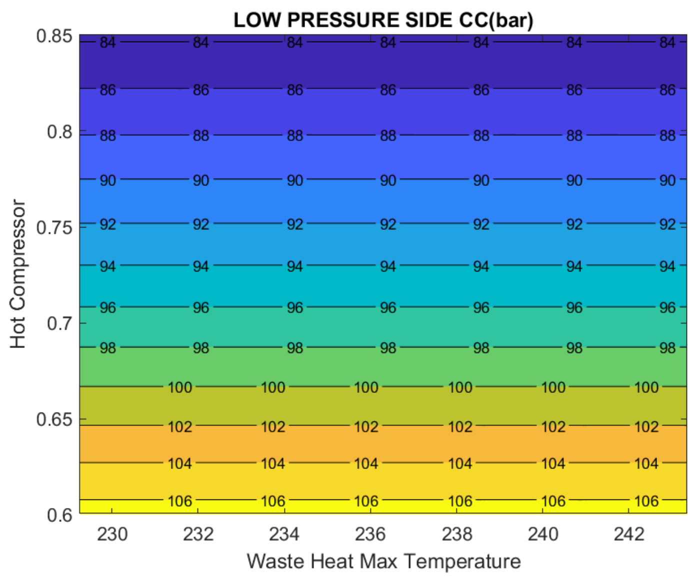

Figure 4 – Low pressure side of the charging cycle

The waste heat recovery is one rerlevant aspect of this configuration as to evaluate the proper use of the freely available heat source. The waste heat recovery (WHR) can be calculated as:

$$WHR = \frac{T_{WH,Max} - T_{WH,Min}}{T_{WH,Max} - T_{amb}} \times 100 \quad (8)$$

where WHR is waste heat recovery, $T_{WH,Max}$ is waste heat maximum temperature when entering into system heat exchanger, $T_{WH,Min}$ is the temperature at which the waste heat leaves the system.

Given the parameters of the test rig, the waste heat recovery variation with respect to the turbomachinery efficiency in the charging cycle can be seen in Figure 5, where hot compressor(HC) is plotted on the vertical axis and cold turbine (CT) on the horizontal axis. The higher turbomachinery efficiency leads to higher WHR as evident by the 17% of the waste heat recovered for the 85% compressor and 60% of turbine efficiency. The trend seems fairly linear with the waste heat recovery contours getting thinner with higher turbomachinery efficiencies, showing a greater chance of recovering more heat with even a smaller change in efficiency of the turbomachinery as the efficiencies become higher. From the sensitivity point of view, waste heat recovery seems to be more sensitive to the variation of efficiency of cold turbine than the hot compressor. This is because the cold turbine efficiency determines the turbine outlet temperature, which, in turns, determines the waste heat outlet temperature and eventually the amount of heat recovered.

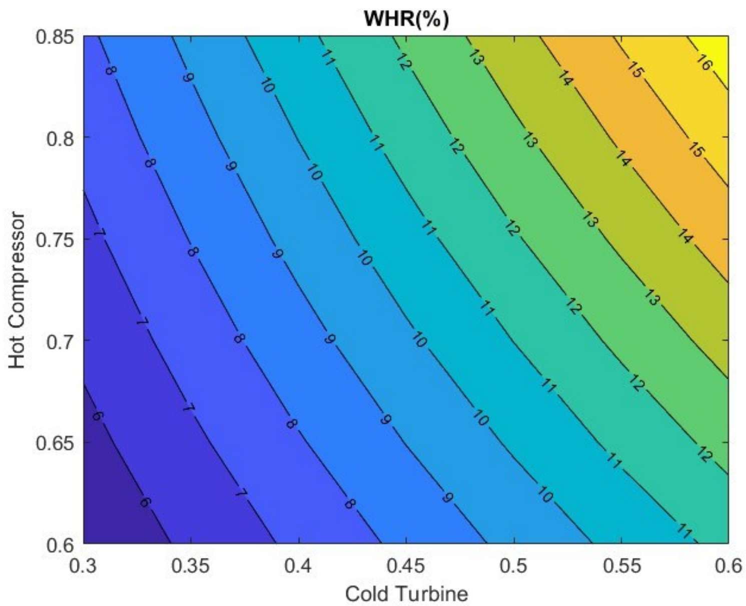

Figure 5 – Waste heat recovery percentage during charging cycle

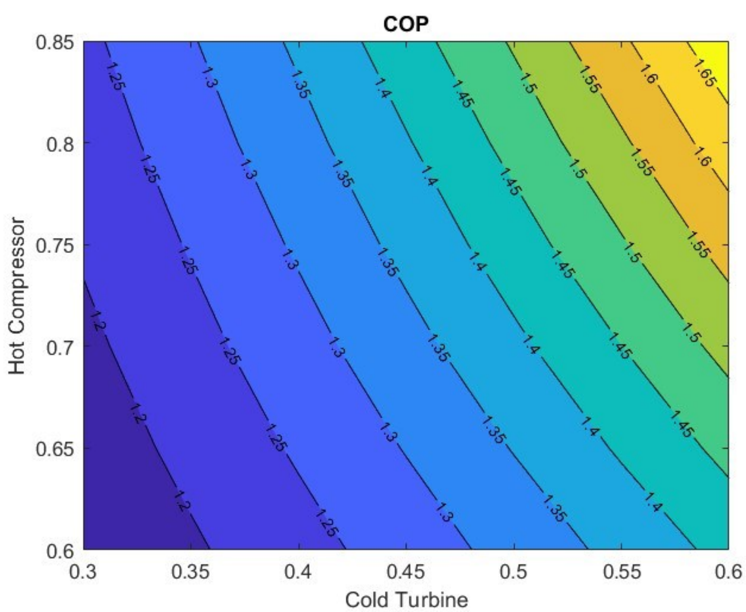

Figure 6 – Coefficient of Performance during charging cycle

Similar to the WHR, COP shows more or less the same trend in terms of sensitivity, i.e. COP is more sensitive to the cold turbine efficiency variation than the hot compressor, as shown in Figure 6. Depicting that for a decent efficiency to start with, there is a higher chance of achieving higher COP with a smaller increase in the efficiency. In other words, if we look at the section of the figure where the hot compressor and cold turbine efficiencies are varied from 60% to 75% and 30% to 45% respectively, the increase in the COP can be seen to be much lower compared to the section for the variation from 75%-85% and 50%-60% respectively showing that at higher efficiencies of turbomachines the COP become more sensitive. For the test rig, the maximum achievable COP is around 1.65, which, although low, can be attributed to the cold turbine low efficiency range.

Discharging Cycle

In addition to the cold compressor (0.7-0.85 with 0.05 step) and the hot turbine (0.7-0.9 with 0.05 step) efficiency sensitivity, the discharging cycle has been analyzed also for sensitivity on recuperator effectiveness as well (0.8 - 0.85 - 0.9). The thermal energy storage temperatures, mass flow rate as well as the high pressure side of the CO2 loop are fixed.

Similar to the charging cycle, the lower pressure side of the discharging cycle is not constrained by the system and is subject to change to accommodate for the variations in cold compressor and hot turbine efficiencies. Figure 7 shows that for the change in the effectiveness of the recuperator, the dependency of the lower side pressure changes can be seen vividly. For 0.8 effectiveness of the recuperator, the lower side pressure is highly sensitive to the hot turbine efficiency and nearly constant for the cold compressor efficiency. However, as the effectiveness of the recuperator increases, the cold compressor efficiency becomes more relevant for the low side pressure. High turbine efficiency brings to lower values of inferior pressure. Nevertheless as the recuperator effectiveness increases, the overall thermal energy recuperated in this heat exchanger increases thus bringing to more relevant impact of cold compressor on low side pressure of the cycle. However such impact is less significant if compared to the hot turbine one. (Fig.7 variations in the three pictures).

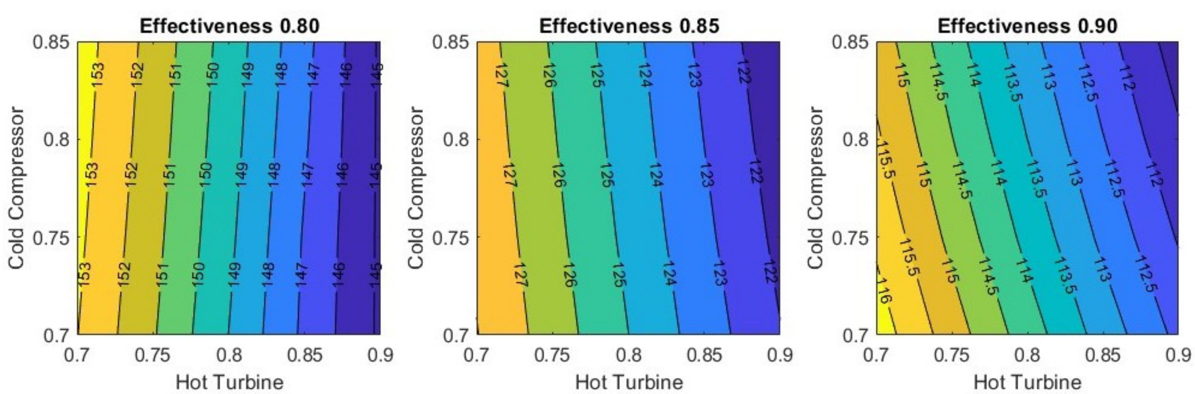

Figure 7 – Low pressure side of discharging cycle

Regarding the efficiency of the discharging cycle (Figure 8), the cold compressor and hot turbine show a relatively linear trend, with cycle thermal efficiency being more sensitive to the hot turbine performance than cold compressor. Looking at the effectiveness of the recuperator, low 0.8 to high 0.9 we can see significant thinning of the contour sections, suggesting that changes in efficiencies of the turbomachinery for higher effectiveness can cause significant changes in the efficiency of the cycle as compared to the low effectiveness recuperators. With highest performance assumptions of discharge cycle machinery, the thermal efficiency of around 30% can be achieved for the test rig.

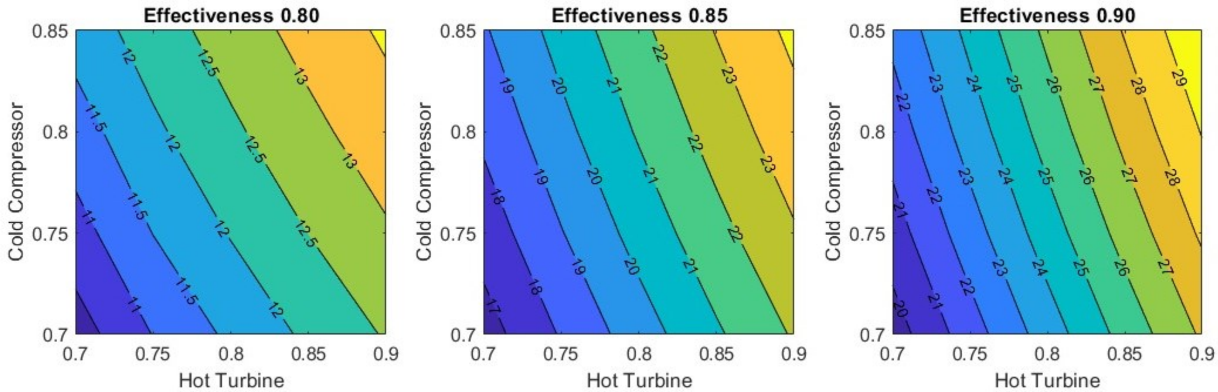

Figure 8 – Thermal efficiency of discharging cycle for various recuperator effectiveness values

Round Trip Efficiency Sensitivity

After the separate evaluation of charging and discharging cycle, this section explores the sensitivity of round trip efficiency (RTE) on the four machine isentropic efficiencies along with the different recuperator effectiveness. While two components are being varied, other two are kept constant with the highest possible efficiencies that can be assigned to them. In the figures hot turbine is referred to as HT, cold turbine as CT, cold compressor as CC and hot compressor as HC, whereas the effectiveness of the recuperator in the discharging cycle has been referred to as EFF.

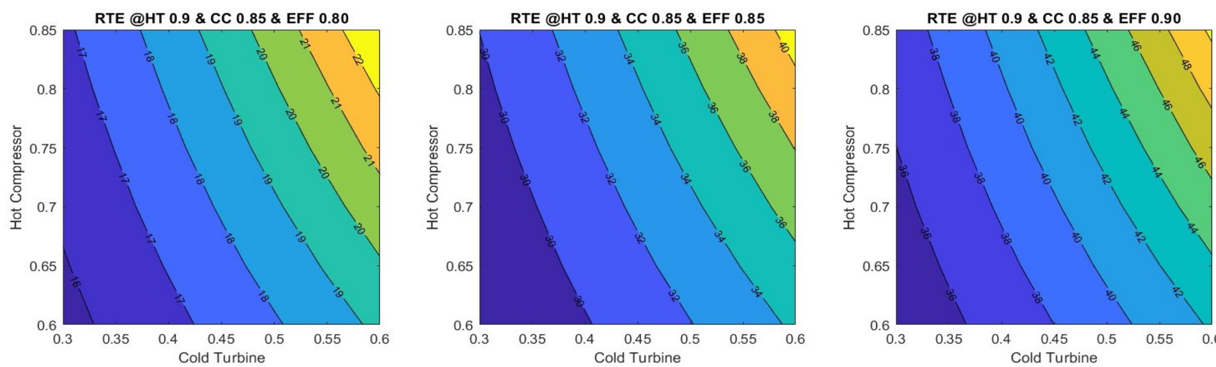

Figure 9 – RTE sensitivity on turbomachinery performance of charging cycle

Figure 9 shows the RTE variations and sensitivity for the hot compressor and cold turbine i.e. components of the charging cycle, where the discharging cycle components i.e. HT and CC are at constant maximum allowable efficiencies of 0.9 and 0.85. However the effectiveness of the recuperator in the discharging cycle is changing from 0.8, 0.85 to 0.9 and is shown by three different contour plots. The effectiveness of the recuperator does not have any direct connection with the charging cycle turbomachines, however its impact through RTE can be identified: in fact, the maximum RTE reached is 22%, 41% and 50% for 0.8, 0.95 and 0.9 EFF, respectively. In terms of sensitivity, it can be seen that, similarly for the COP, the RTE is more sensitive upon the cold turbine performance rather than the hot compressor.

At the same time Figure 10 shows the sensitivity of RTE for the discharging cycle turbomachinery components, where the charging cycle components i.e. HC is fixed at 0.85 and CT is fixed at 0.6. The sensitivity for both the efficiencies of hot turbine and cold compressor is highest for the EFF of 0.8, whereas it is lowest for EFF 0.85. In general, RTE is more sensitive to hot turbine than the cold compressor.

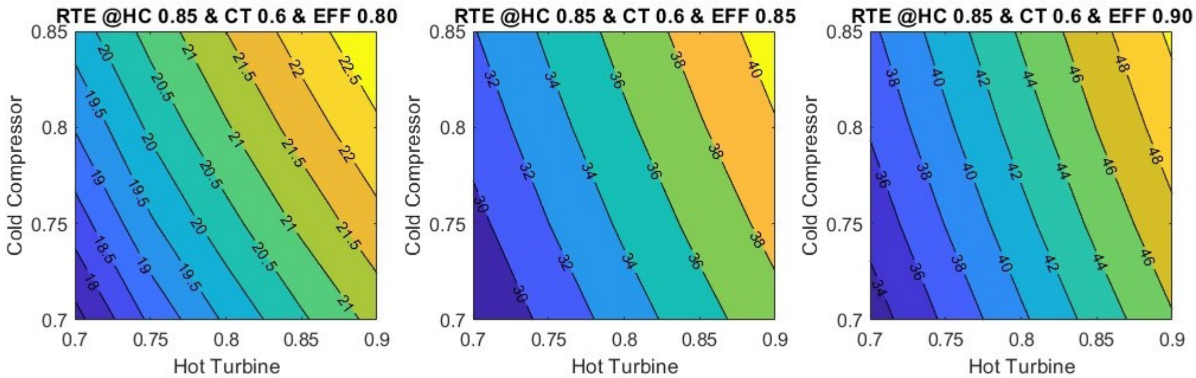

Figure 10 – RTE sensitivity on turbomachinery performance of discharging cycle

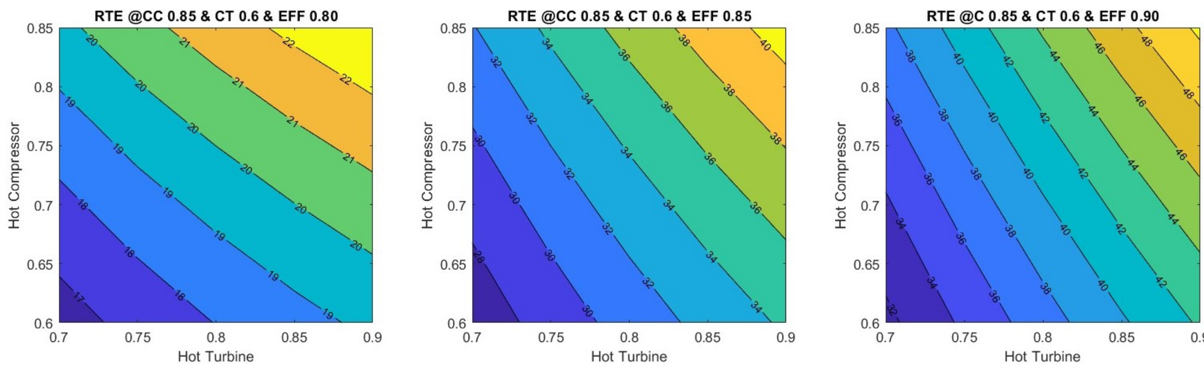

Figure 11 – RTE sensitivity on hot turbine and hot compressor performance

HC and HT, the major components of the charging and discharging cycle, are plotted to understand their effect on the RTE and compare the sensitivity of RTE for different effectiveness, as shown in Figure 11. The CC and CT are kept constant at the efficiencies of 0.85 and 0.6, respectively. At EFF 0.80, RTE is more sensitive to HC efficiencies than the HT efficiency, however as the EFF increases the sensitivity of RTE to HT becomes higher, at 0.90 effectiveness RTE shows more rapid changes for the change in HT efficiency as compared to the HC efficiency. As both of these are cost intensive components, as trade-off consideration it can be concluded that for lower recuperator effectiveness, it is better to have a high efficiency compressor, whereas if we have a highly effective recuperator, it is preferable to have highly efficient turbine.

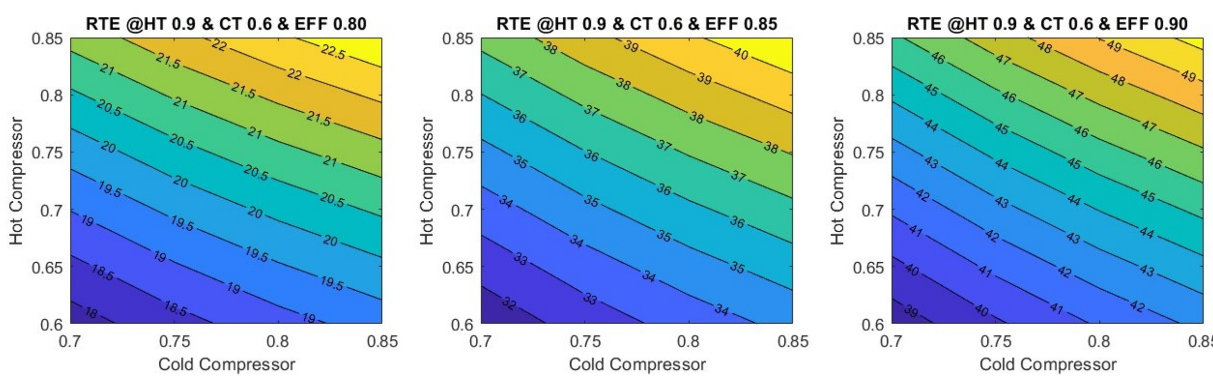

Figure 12 – RTE sensitivity on cold compressor and hot compressor performance

Figure 12 shows the sensitivity on hot compressor and cold compressor. The HT efficiency is fixed at 0.9 and the CT one is fixed at 0.6, which is their maximum allowable efficiencies. The effectiveness 0.9 shows comparatively thinner contours showing higher sensitivity of the RTE for change in both compressor efficiencies for higher effectiveness of the recuperator. Overall RTE seems to be more responsive to the changes in the hot compressor efficiency compared to the cold compressor efficiency. Increase in the recuperator effectiveness from 0.8 to 0.9 increases the responsiveness of RTE to the cold compressor.

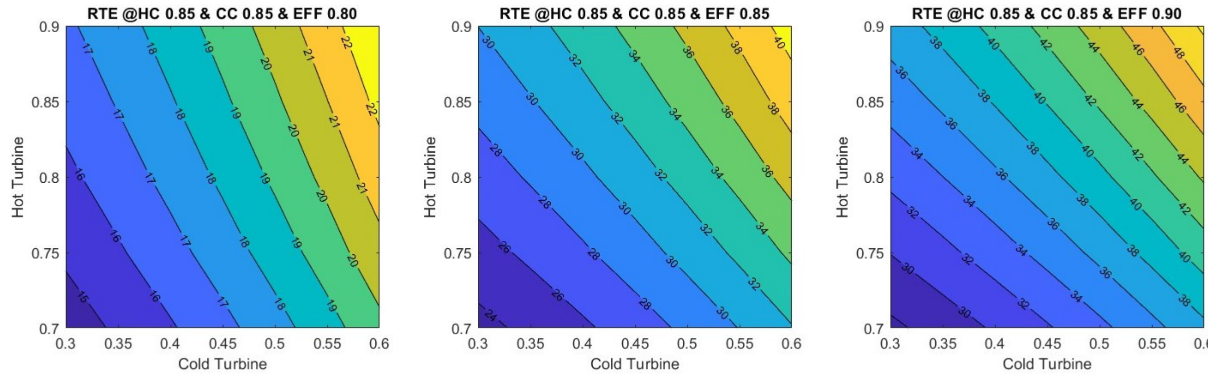

Figure 13 – RTE sensitivity on hot turbine and cold turbine performance

The RTE sensitivity to hot turbine and cold turbine is depicted in Figure 13 for the HC and CC efficiency of 0.85. The contour sections are thick for effectiveness 0.80 and become thinner for higher effectiveness, as it can be seen for 0.9, showing high sensitivity of RTE to the changing efficiency of both the turbines. For lower effectiveness, RTE seems to be more sensitive to the cold turbine whereas with the effectiveness becoming higher the sensitivity to the hot turbine efficiency also increases for the EFF 0.9 case.

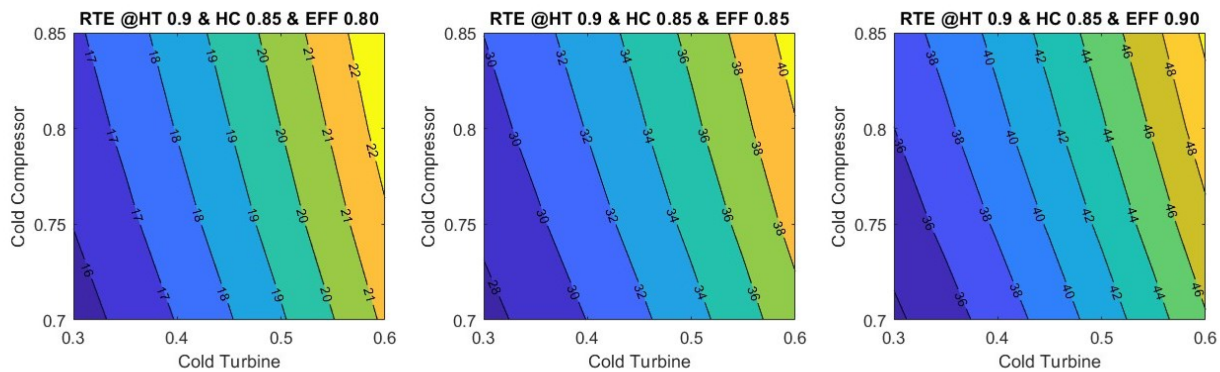

Figure 14 – RTE sensitivity on cold turbine and cold compressor performance

When comparing the cold turbine with the cold compressor in terms of RTE sensitivity, the HT is kept at 0.9 efficiency and HC is kept at 0.85 efficiency as shown in Figure 14. Cold turbine shows higher influence on RTE compared to the cold compressor. The influence does not change much with EFF.

After evaluating the RTE sensitivities for the changes in turbomachinery and effectiveness of the recuperator in the discharging cycle. It is also important to evaluate the impact of each component's efficiency on the overall RTE to find out the ranking of sensitivity of each component. For that purpose, by fixing all the other parameters to their highest possible values, the change in RTE with respect to the individual change in each component's parameter is computed at fixed intervals. To understand this trend the following formula is proposed:

$$\Delta SENS = \frac{RTE_{i+1} - RTE_i}{n_{i+1} - n_i} = \frac{\Delta RTE}{\Delta n} \tag{7}$$

where

- Δn is the Interval of change of the parameter
- ΔRTE is the Change in RTE over the Δn
- $\Delta SENS$ is the Sensitivity of RTE

Table 3 – Individual sensitivity assumptions

Component	Range of parameter	Interval of change (Δn)	Number of instances of $\Delta SENS$
Cold Turbine	0.3-0.6	5	6
Hot Compressor	0.6-0.85	5	5
Hot Turbine	0.7-0.9	5	4
Cold Compressor	0.7-0.85	5	3

Table 3 shows the range of parameters, interval of change and number of instances that can be got calculating $\Delta SENS$. (e.g. for Cold Compressor, as the range is from 0.7 to 0.85 and interval of change is 0.05, there are three instances over which the change in RTE is calculated – i) 0.75-0.7 ; ii) 0.8-0.75 ; iii) 0.85-0.8).

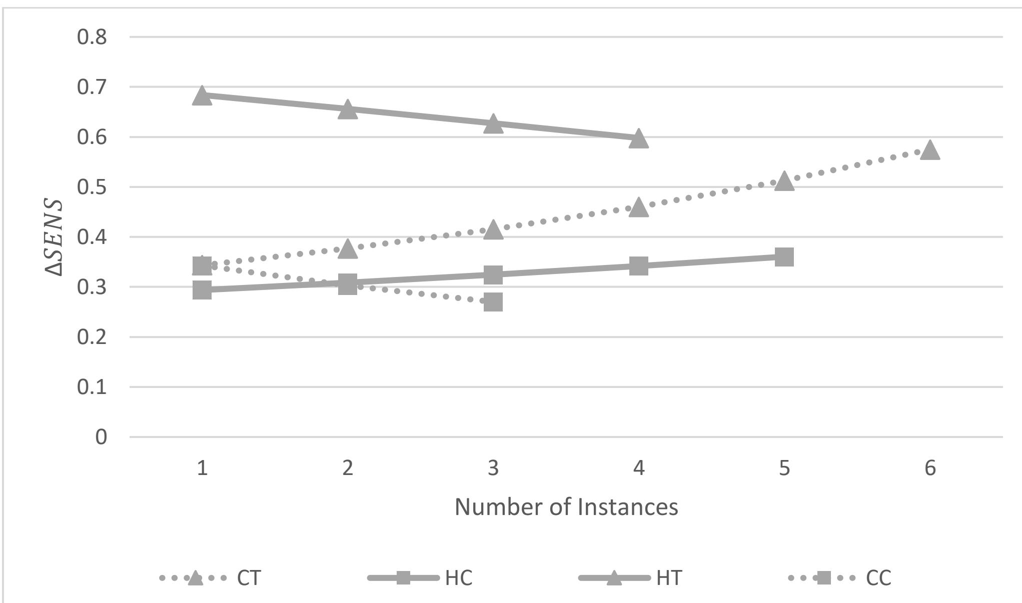

Figure 15 – RTE sensitivity for each individual component at fixed intervals.

Figure 15 presents the effect of single machines' efficiencies on RTE (solid line for the hot components, dotted for cold ones; triangles for the turbines, squares for the compressors). Results from the figure can be interpreted as it follows, showing this ranking on the influence:

1) Hot turbine: The recuperator has a major impact on the RTE as we have seen in the sensitivity analysis. As the hot turbine outlet conditions determine the inlet conditions of the recuperator, indirectly η_{HT} has the major influence on the thermodynamic aspects of the discharging cycle thus impacting on overall RTE (obviously considering a top-efficient charging cyvle).

2) Cold Turbine: Cold turbine outlet conditions are responsible for the amount of waste heat recovered. Variations in the cold turbine efficiency influence the turbine outlet temperature which corresponds to waste heat outlet temperature and WHR along with the pressure ratio of the charging cycle having a dominant effect on the COP and RTE.

3) Hot Compressor: For a fixed thermal gradient on the hot compressor side (as inlet temperature of the compressor is fixed by WH source and TES temperature is fixed in this analysis) hot compressor efficiency influences only pressure ratio of the charging cycle, thus influencing the RTE. As showed by authors in [10], varying hot compressor temperature glide has a relevant impact on RTE.

4) Cold Compressor: similarly, cold compressor efficiency influences the pressure ratio of the discharging cycle influencing the RTE

As already highlighted by authors in [10] , Figure 15 shows that the increase in RTE becomes less significant as the efficiency of the turbomachinery becomes higher in the discharging cycle and inverse is true for the charging cycle, thus showing the higher impact of a most efficient charging cycle on the overall RTE.

CONCLUSION

This paper introduces an innovative concept of a sCO2 TI-PTES system that will be validated in a first of its kind laboratory rig to be located at University of Genova (UNIGE) premises, exploiting locally available heat from the Tirreno Power combined cycle power plant. The proposed solution has the potential to be applied to valorise different freely available heat sources (solar, WH, biomass heat…) improving energy storage RTE and reducing CAPEX by utilizing the heat source and avoiding the cold thermal energy storage.
The paper presents the first analysis performed by the authors to identify the design of the test-rig to be realized in the framework of the EU funded project SCO2OP-TES. A sensitivity analysis was conducted to explore the performance characteristics of the charging and discharging cycles of such a PTES system, focusing on the influence of turbomachinery efficiency and recuperator effectiveness. The performance evaluation is mainly based on electrical RTE (accounting for electrical energy flows) targeting its maximisation. From the results summarized in Table 3, it is possible to infer a rank of components from the highest to the lowest influence on RTE, thus also providing relevant guidelines to technology developers:

1. Hot Turbine
2. Cold Turbine
3. Hot Compressor
4. Cold Compressor

Despite its low power capacity and being the lowest efficiency component (due to its bladeless expander nature), the cold turbine efficiency of the charging cycle has shown the highest sensitivity on RTE: this is probably related to the fact that it has a direct influence on the COP of the HP-charging cycle and, at the same time, its irreversibility cannot be partially compensated (on the contrary, the hot compressor irreversibility, resulting in higher output temperature, may be partially recovered during the discharging cycle, thanks to a higher hot turbine inlet temperature). As already shown by the authors the HP COP [10] has highest relevance on the final RTE. This statement is confirmed by the fact that charging cycle hot compressor efficiency has a higher relevance than the cold compressor efficiency of the discharging cycle one.
This is even more relevant particularly if the effectiveness of the recuperator of the discharging cycle does not allow higher power cycle efficiency.

By proposing a sCO2 Carnot battery/power cycle sizing modeling tool, to be later validated thanks to the planned test-rig, this study sets the ground for the first project analysis, providing relevant information about where to focus component and system design optimization activities towards maximization of RTE. Considering the low scale of the machines and the pressure/temperature in place, it is worthy to highlight that a RTE up to 0.5 could be achievable in laboratory, while for future upscaled cycles benefitting from higher efficiency machines values exceeding 0.7 seems possible at pressures up to 250 bar.

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