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Thermally-integrated CO2 cycles for MW-scale power generation and storage

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Abstract. In an energy scenario driven by Renewable Energy Sources (RES), where more and more bulky quantities of RES should be introduced on the grid, the role of energy storage systems 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, whose 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. The peculiar features of supercritical CO_2 (sCO_2) make it the ideal candidate to act as working fluid for large scale PTES applications. sCO_2 cycles are indeed fully compatible with the temperature range of TES hot storage sources and sCO_2 has already been used in commercial HP solutions (even targeting high temperature HPs). In addition, sCO_2 allows energy storage to embody a compact design as well, making the whole PTES footprint smaller compared to technologies based on other working fluids. Nevertheless, sCO_2 based PTES solutions cannot achieve significant Round Trip Efficiency (RTE): a possible solution to such a limitation is represented by the exploitation of freely available heat sources (like thermal RES or waste heat), which could increase the COP of the charging cycle and, at the end, the electrical-based RTE, therefore in the so- called Thermally Integrated Pumped Thermal Energy Storage (TI-PTES). In the framework of the PRIN 2022 project ECO-SEARCHERS, specific cycle layouts of interest are studied and compared on a thermodynamic basis: the most promising solution is presented in this paper. Finally, the test rig that will be used for laboratory-scale validation of a small size radial bladeless turbine is described, providing a first glance to the technical challenges in realizing and managing sCO_2 cycles in practice.

1. Introduction

In recent times, electrical markets in Western countries are experiencing very low prices, or even negative where rules allow for it, during periods of high renewable power production, mainly wind and solar. In such a context, which might be augmented due to the energy transition towards clean but un-dispatchable power sources, , the development and deployment of large-scale (MW size) and long-duration (>8 h) energy storage (LDES) technologies will play a crucial role for the electrical grid regulation and sustainability [1], in order to shift the RES production from production peaks to consumption peaks as well as to augment grid flexibility and resilience. A part from electrochemical battery solutions, which may well fit small scale end-users, other technologies are needed at high voltage electrical line scale, targeting multi-MW installations and preferably adopting rotating machinery, which may serve to mitigate congestion in critical nodes of the grid as well as providing spinning reserve. Therefore, 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 for electrical transmission operators, offering significantly large energy and power storage capacity, excellent storage cyclability, rapid response time, limited dependency on geographical location (differently from pumped hydro storage) [3].

Among Carnot batteries (CB), pumped thermal energy storage (PTES) [4] works by turning electricity into heat using a large-scale heat pump (HP) and transfers heat from a “cold TES” to a “hot TES”; the latter is constituted by solid material, molten salts, or phase change materials (PCM) depending on the temperature. Whenever needed, such thermal potential is turned back to electricity using a power cycle based on a closed thermodynamic cycle. A recent paper presents a review on the most promising solutions based on CO_2 as working fluid [6], which allows to exploit waste heat at industrial scale with attractive thermo-economic features [7].

These thermodynamic cycles, however, demand the adoption of highly non-conventional turbomachinery, for both the charge and discharge phases. The machines operating in the cold part of the cycles (the charge expander and the discharge compressor) work with CO2 in the dense-gas condition and close to the critical point, implying low flow function and high-load coefficients, which typically suggest to select radial architectures [8]; moreover, the near-critical operation makes the fluid highly sensitive to the thermodynamic state and prone to phase change in local regions within the rotors [9]. The machines operating in the hot part of the cycles (the charge compressor and the discharge expander) feature completely different technological issues, as the high temperature and pressure of the fluid, associated to the small-scale of the channel sections, pose unprecedented issues in the design of the sealing system, in the aero-structural optimization of the blade sections, and in the selection of the proper material [10]. While these aspects have been partially addressed in the recent literature about turbomachinery for sCO_2 power systems (which are similar to the present discharge cycle), the turbomachinery design for the sCO2 charge cycle has to face completely new challenges and demands a high degree of novelty.

Most experimental research on sCO_2 expanders is focused on small-scale devices for the refrigeration sector. This focus arises from the significant pressure differences between the condenser and the evaporator in trans-critical CO_2 refrigeration cycles, which results in substantial throttling losses and, consequently, markedly reduce the COP of the system. To address this issue, numerous test benches have been employed to evaluate different types of volumetric expanders as substitutes of throttling valves. Baek et al. [11] developed a piston-type expander, reporting an isentropic efficiency lower than 10%. A radial piston reciprocating expander has been investigated by both Fukuta et al. [12] and Ferrara et Al. [13], achieving an overall 40% isentropic efficiency level.

Conversely, research on single-phase sCO_2 expander test benches remains quite limited and frequently draws on solutions from the refrigeration industry [14]. While numerous authors have proposed studies detailing the design procedures for such test rigs [15][16], publications of experimental results in this area are still scarce.

In this paper, the most promising layout for a Thermally-Integrated Pumped Thermal Energy Storage (TI-PTES) concept with specific net power size of 5MW, both during the charging and discharging phases, is presented and performance discussed. Furthermore, the design space for turbomachinery is presented, including initial considerations on the design solutions related to radial machines, which are considered the most suitable for the power size analysed. Finally, the paper outlines the forthcoming research objectives, which include experimental validation of a small scale CO_2 bladeless turboexpander.

2. Cycle configuration

2.1 Selection of cycle layout

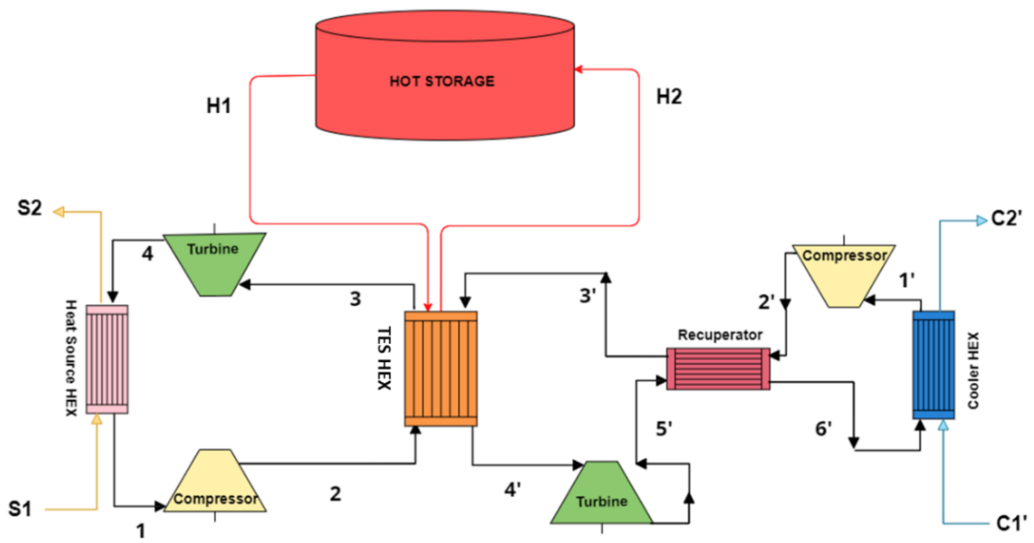

Figure 1 – Cycle layout of TI-PTES.

The TI-PTES system in this study is being investigated with a simple charging cycle and a recuperated discharging cycle connected by a thermal energy storage system as shown in Figure 1. The charging cycle (CC) works out as it follows, with reference to the main thermodynamic points: (1) the working fluid (which in this case is sCO_2), previously heated-up by the external heat source, passes through the "hot" compressor, where pressure as well as enthalpy are increased, (2) fluid then goes to the TES heat exchanger (HEX), of counter flow shell and tube type, where TES fluid (e.g. molten salts) enters at one end and sCO_2 enters the other, sCO_2 being the hot fluid, (iii) the "cold" sCO_2 enters a cold turbine, after expansion (4) sCO_2 is ready to recover heat from the external heat source HEX, and then repeats the cycle.

The thermal energy storages system (TES) is made up of thermal storage material tanks and the heat exchanger TES HEX, which transfers heat to and from the thermal energy storage material, depending whether the system is operating in charging mode or discharging mode, respectively. For the case of molten salts and thermal oils, the storage material itself is the TES fluid, as it flows from one tank to the other, exchanging heat passing through the TES HEX. In other cases, out of the scope of the present paper, the fluid can further transfer heat to a solid material (e.g. concrete), but introducing additional heat transfer irreversibilities.

The recuperated discharge cycle (DC) starts at (1') the cold compressor, which inlet conditions are nearly ambient and close to the CO_2 critical point to assist in power saving for compressor operation. After compressor, (2') sCO_2 passes through the recuperator and gets preheated before entering (3') the TES HEX to extract heat from the TES HEX, attaining the peak temperature; then (4') it goes through the hot expander, after which it (5') passes through the recuperator and gives off any extra heat (6') finally through the cooler, to come back to the compressor inlet conditions again. With the exception of the thermal energy storage system, which comprises of the storage material in the hot storage container and the TES HEX, all the components of the charging and discharging cycles are separate.

The main assumptions for thermodynamic calculations are summarised in Table 1.

The scale of the plant is selected to be 5 MW net for the charging (power absorbed) as well as the discharging (power produced) cycles, to ensure equal grid flexibility in terms of power taken and provided to the grid. The scale of the thermal storage is selected to be 50 MWh which means that we will have variable charging and discharging time based on the performance of the TI-PTES.

The TI-PTES is being evaluated at two temperature levels. For low storage temperature of 350°C utilizing low temperature waste heat at 250°C, whereas for the second configuration the storage temperature of 550°C is selected that utilizes the WH temperature of 450°C. Despite the availability of waste heat at 450°C is not so common, since it can well find other uses in an industrial context, it can occur in a few industrial processes where, for instance, pre-heating of combustion air is challenging due to restrictions on NOx emissions. Both of the configurations have a 100°C gap in between the source and the storage temperature. The 250°C seems to be more of a practical temperature level for waste heat recovery, however, the storage temperatures of 550°C and higher seems more appropriate for tailored applications, such as Concentrated Solar Power (CSP), and to achieve better performance, as later demonstrated.

The charging and discharging pressures are fixed based on previous studies. In particular, the following rationale has been adopted: (i) DC should remain supercritical, and therefore 83bar and 183bar are selected as extreme pressures, to keep some safety margin against two-phase region and to comply with HEX and recuperator structural constraints, respectively; (ii) CC is calculated consequently, to comply with the WH and TES temperature levels, allowing also for trans-critical cycle if necessary, but requiring no change of phase and maximum pressure or 200bar, for structural constraints.

Table 1 – Thermodynamic assumptions.

Parameter	Value	UoM
General		
Compressor isentropic efficiency	80	%
Turbine isentropic efficiency	85	%
HEX pressure losses	1	%
ΔT pinch	10	K
DC Recuperator effectiveness	80	%
Electrical and mechanical eff.	99	%
Discharging Cycle		
DC high pressure	183	bar
DC low pressure	85	bar
Coolant inlet temperature (Air)	25	°C
Coolant exit temperature (Air)	45	°C
Net power (- CC, + DC)	±5	MW
TES		
TES storage capacity	50	MWh
TES storage temperatures	350 & 550	°C
Charging Cycle:		
WH temperature	250 & 450	°C
CC high pressure	200	bar
Net power	5	MW

2.2 Cycle performance analysis

Cycle modelling was performed in EBSILON© for the abovementioned two different temperature combinations of waste heat and thermal storage temperatures. Both of the combinations have been scaled up for the 5 MW of power in charging and discharging modes, featuring 50 MWh of storage capacity.

The thermodynamic points for the combination 250°C waste heat temperature and 350°C storage temperature are reported in Table 2.

Since the PTES systems consist of two separate cycles, one for charging and one for discharging, the simulation separates the computation of the two cycles, indirectly connected by the temperatures of the TES fluid, and the amount of energy and mass stored in it. In practice, the model firstly computes the DC, and then uses the results to initialize the computation of the CC. For sake of clarity, since the discharging layout consists of a recuperated cycle, the minimum temperature of the TES depends, for both charging and discharging, on the inlet temperature of the TES fluid in TES HEX in the DC, and thus on the effectiveness of the recuperator. For this reason the DC is computed as first, and then the low temperature tank (TES fluid inlet temperature in the TES HEX in the DC) is used to set the boundary conditions in the CC.

Table 2 – Cycle design point for 250°C waste heat and 350°C storage.

Thermodynamic point	Pressure (bar)	Temperature (°C)	Mass flow rate (kg/s)
1	77.6	240	62.9
2	200	365	62.9
3	198	183	62.9
4	78.4	103	62.9
H1	1.20	171	62.9
H2	1.19	350	62.9
S1	1.20	250	79.3
S2	1.19	113	79.3
1’	83.0	33.0	102
2’	183	54.8	102
3’	182	161	102
4’	180	340	102
5’	84.7	266	102
6’	83.9	75.4	102
H2	1.20	350	92.5
H1	1.19	171	92.5
C1	1.20	25.0	927
C2	1.19	45.0	927

Table 3 – Performance results for 250°C waste heat and 350°C storage.

Parameter	Column heading
COP	3.22
Efficiency	0.207
RTE	0.667
Charging Hours	3.10
Discharging Hours	2.11

Table 4 - Cycle design point for 450°C waste heat and 550°C storage.

Thermodynamic point	Pressure (bar)	Temperature (°C)	Mass flow rate (kg/s)
1	67.4	390	48.5
2	200	560	48.5
3	198	316	48.5
4	68.0	215	48.5
H1	1.01	306	38.1
H2	1.00	550	38.1
S1	1.01	400	52.0
S2	1.00	225	52.0
1’	83.0	33.0	65.2
2’	184	54.8	65.2
3’	182	299	65.2
4’	180	540	65.2
5’	83.7	453	65.2
6’	82.8	105	65.2
H2	1.01	550	50.4
H1	1.00	306	50.4
C1	1.01	25.0	706
C2	1.00	45.0	706

Table 5 – Performance results for 450°C waste heat and 550°C storage.

Parameter	Column heading
COP	2.84
Efficiency	0.259
RTE	0.739
Charging Hours	3.44
Discharging Hours	2.60

The inlet pressure of the compressor is around 77 bar, which keeps the cycle in the super critical state avoiding any two-phase flow. As the upper side pressure is fixed, the lower pressure of the compressor is subject to change depending on WH or TES temperature levels. For these thermodynamic points, the performance of the combination 250°C waste heat and 350°C TES temperature is reported in Table 3.

As the WH elevates the temperature from approx. ambient to 250°C, the CC compressor has to do less work to raise the temperature up to 350°C for storage. Therefore the COP is higher. The temperature of the TES, however, is in the low range of sCO_2 plants for the discharging cycle (power cycle) and along with 183 bar of upper side pressure on the discharging side, the first principle efficiency of the discharging cycle is 20.7% only. The TI-PTES system can provide therefore 2.11 hours of discharging for 3.1 hours of charging, because of the component irreversibilities, and therefore determining an RTE < 1.

For the combination of 450°C WH and 550°C TES temperatures, the lower side pressure is 67.38 bar, which brings the charging cycle into transcritical operation, although the CC still remains of Brayton cycle type. The charging cycle lower pressure for both the combinations is lower than the one of the discharging cycle. The related thermodynamic points are reported in Table 4.

It can be seen that for same power flexibility of 5 MW and thermal energy storage of 50 MWh, the higher temperature configuration of 450°C WH and 550°C TES requires lower mass flow rate, which can lead to lower size plant and better capital costs than the low temperature combination. As the temperatures are higher, the COP is a bit lower, however due to higher temperature of storage the efficiency as well as the RTE is higher, bring the charging and discharging time to 3.44 hours and 2.6 hours, respectively. Specific considerations on thermal storage size and suitable high temperature materials, such as Syltherm, Yara Salt, HitecXL, Solar salt and Concrete, can be found in [7].

3. Machinery preliminary design

3.1 Machine design space

The integrated sCO_2-based storage energy system introduced in previous section involves four machines, whose technical features and efficiency are crucial for the success of the entire technology. While for a preliminary cycle analyses reasonable assumptions were made regarding compressor and turbine efficiency, advanced thermo-economic analyses demand the use of reliable estimates on turbomachinery efficiency and size. Since in the present system the machines operate with supercritical CO2, classical performance correlations cannot be exploited and dedicated designs are required. Moreover, the turbomachinery designed for the charge cycle of the storage system have to operate in totally unconventional conditions and novel configurations must be conceived. A first step in the integration between the thermodynamic design of the cycle and the fluid-dynamic design of turbomachinery is presented in this section, where global operation parameters are evaluated in light of the findings of previous section.

Considering a 5 MW capacity of the system, intended as both the net power absorbed by the charging cycle and the one released by the discharge cycle, two levels of maximum TES temperature have been considered, namely 350 °C and 550 °C, which correspond to a maximum temperature of the integrated WH recovery equal to 240° and 400 °C respectively. As result of the different conditions at fixed power of 5 MW, different flow rates result for the charge and discharge cycles, as well as between the two TES temperature levels, as reported in Table 6.

Table 6 – Preliminary design space for turbomachinery sizing.

TES: 350° C - charge: m = 68.9 kg/s	Pressure (bar)	Temperature (°C)
Compressor inlet	77.6	240
Compressor outlet	200	365
Turbine inlet	198	183
Turbine outlet	78.4	103

TES: 350° C - discharge: m = 102.5 kg/s	Pressure (bar)	Temperature (°C)
Compressor inlet	83.0	33
Compressor outlet	184	55
Turbine inlet	180	340
Turbine outlet	84.7	266

TES: 550° C - charge: m = 47.5 kg/s	Pressure (bar)	Temperature (°C)
Compressor inlet	67.4	390
Compressor outlet	200	560
Turbine inlet	198	316
Turbine outlet	68.0	215

TES: 550° C - discharge: m = 65.4 kg/s	Pressure (bar)	Temperature (°C)
Compressor inlet	82.0	33
Compressor outlet	184	56
Turbine inlet	180	540
Turbine outlet	83.7	453

A first inspection of the data reported above indicate that these conditions are all characterized by machines with low-mid specific speeds, namely in the range 0.5 - 0.9 for compressors and 0.35 - 0.6 for the turbines (considering a maximum acceptable angular speed of

30000 rpm), which suggest to consider radial compressors and turbines for both the discharge and charge phases in the whole design space identified. The design of the radial compressors, in particular, pose the highest challenges, for the near-critical state of the fluid in the (cold) compressor of the discharge cycle, and for the high temperature and load of the (hot) compressor of the charge cycle. Preliminary estimates indicate that two-stage compressor layouts may be advantageous, especially for the charging machine.

4. Component validation

Experimental validation of the whole cycles are out of the scope of the PRIN 2022 ECO-SEARCHERS project. However, validation of a laboratory-scale CO_2 expander based on bladeless technology, or Tesla expander, will be performed.

The experimental test rig to be used for the tests of the downscaled Tesla turbine prototype of ECO-SEARCHERS project will be located at the Department of Industrial Engineering (DIEF) of the University of Florence. It consists of a trans-critical CO_2 heat pump, which has been suitably modified for laboratory applications. The test rig is designed to be extremely versatile in its use, allowing for the extraction of the working fluid under various pressure and temperature conditions.

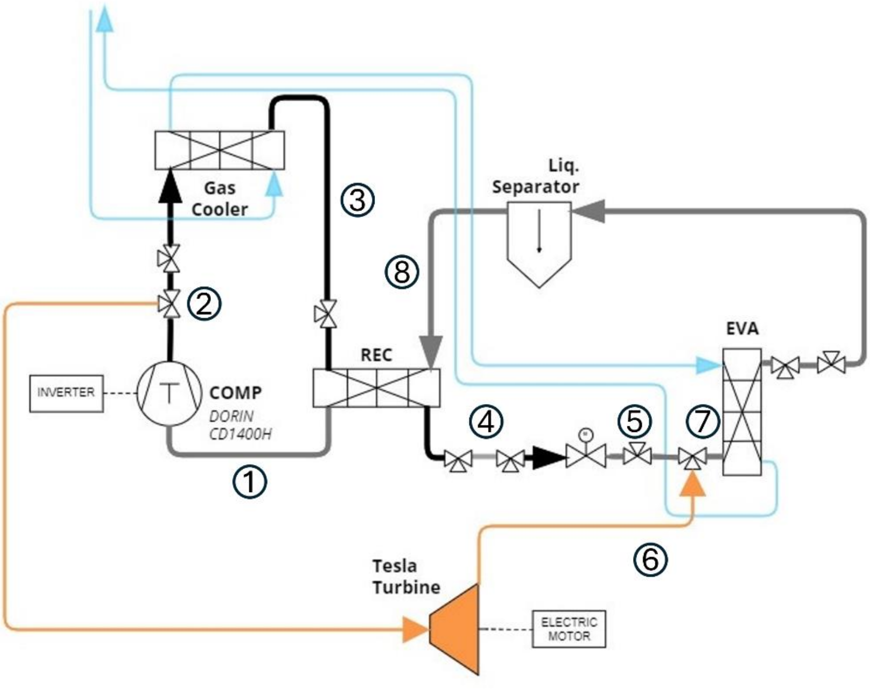

Figure 2 – Schematic layout of CO_2 test rig.

A schematic layout of the test bench is shown in Figure 2, where the connection with the Tesla expander for the experimental campaign is highlighted. As previously pointed out, the reference thermodynamic cycle is a modification of a typical recuperated reverse cycle. In this case, part of the working fluid is extracted downstream of the compressor and expanded into the turbine (charging cycle reference operating condition). The remaining fraction of the fluid is cooled down into the gas cooler, then it transfers part of its heat to a recuperative heat exchanger and, finally, its pressure is reduced through a lamination valve to achieve the desired value at the evaporator

inlet. The fraction of cold gas from the throttling process is then mixed with the hot one coming from the turbine upstream the evaporator, which provides the fluid with the necessary enthalpy to reach the desired conditions at the compressor inlet.
A thermodynamic representation of the test bench inverse cycle is illustrated in Figure 3, where the thermodynamic transformations of the individual components are plotted on a p-h diagram.

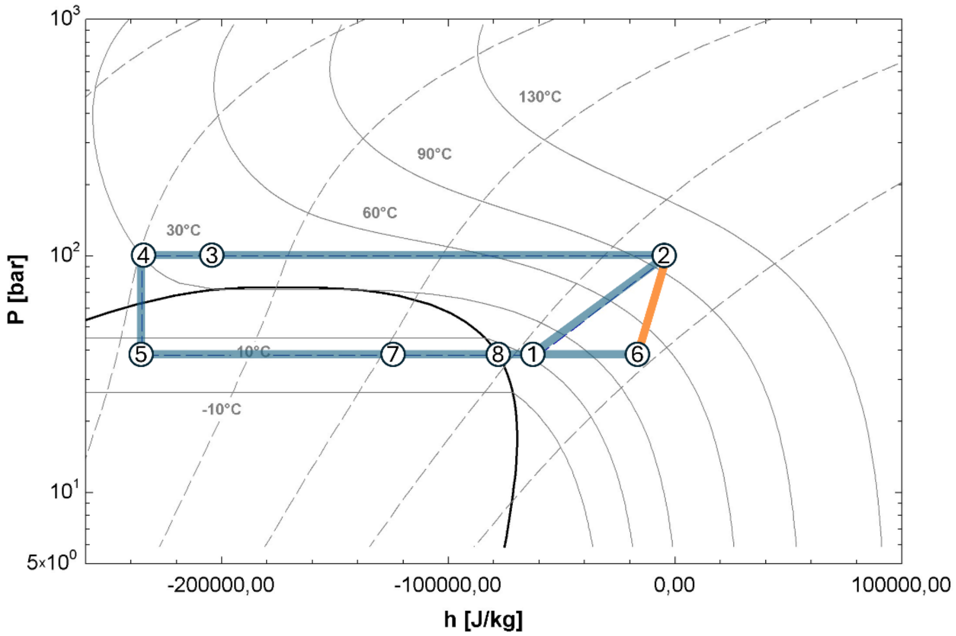

Figure 3 – p-h diagram of CO2 test rig.

The expansion process into the Tesla turbine begins at point 2, at supercritical pressure. To test the turbine in a single-phase region, it is essential to regulate the flow rate and thermodynamic conditions of the fluid to ensure that the turbine output falls outside the saturation dome. A wide range of operating pressures and mass flow rate conditions for the turbine can be arranged by the proposed test bench. However, some operating limits have been established as it follows:

- The maximum and minimum pressure conditions available in the turbine are primarily determined by the selection of the compressor. The test bench is equipped with a DORIN CD1400H, a semi-hermetic volumetric compressor designed for trans-critical CO2 applications. The operational map of this component is shown in Figure 4 (a), indicating the operating limits in terms of condenser pressure and evaporator temperature. Safety valves are installed to ensure that the pressure does not exceed 120 bar in the high-pressure branch of the machine and 60 bar in the low-pressure one.
- The maximum flow rate allowed by the turbine is primarily determined by the test rig control system. Specifically, it is crucial to ensure that the fluid entering the evaporator (point 7), which is obtained by mixing the hot fluid at the turbine output (point 6) with the cold fluid at the throttling valve outlet (point 5), is kept under a two-phase condition. Failure to achieve this condition could bring relevant difficulties in determining the minimum pressure and a significant increase in pressure drop through the expander. Figure 4 (b) illustrates the maximum flow rate which can be extracted from the test bench cycle as a function of the

condenser pressure and evaporator temperature. In this case, the worst-case scenario for turbine operation (e.g. an isenthalpic expansion) has been considered.

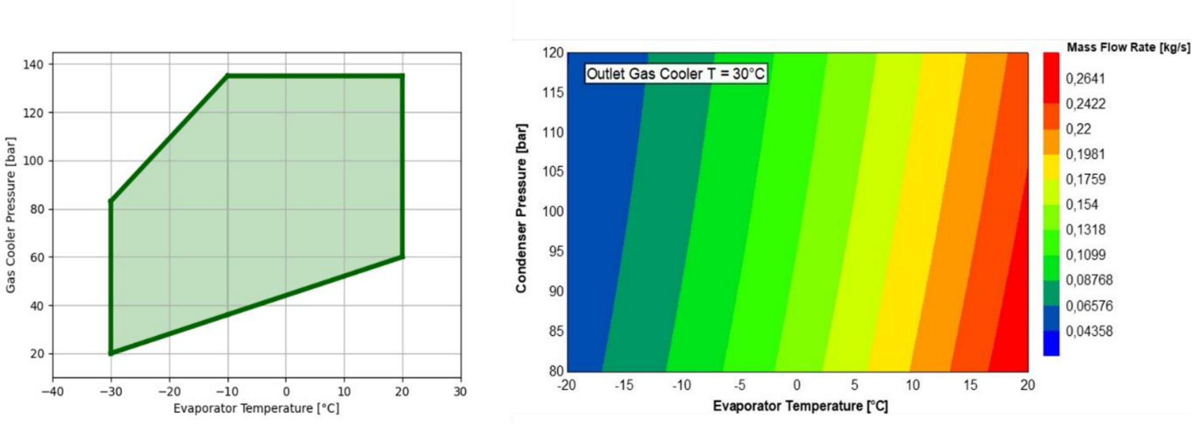

Figure 4 – a) Compressor operating limits b) Maximum turbine mass flow rate.

Conclusions

In this paper, innovative cycle layouts for Thermally-Integrated PTES systems are presented and analysed for the reference size of 5MW net power, both during charging as well as discharging cycles, including preliminary detailed consideration on turbomachinery. The work main conclusions are the following:

- TI-PTES cycles represent a viable solution for large scale energy storage at utility scale, employing rotating machinery and therefore serving also as spinning reserve. In such cycles, the use of CO_2 working fluid is promising for enhanced performance and compact turbomachinery, with the disadvantage of requiring high operating pressures.
- The proposed cycle layouts analysed in two configurations of WH and TES temperature levels (250°C WH and 350°C TES, 450°C WH and 550°C TES), show an electrical RTE of 68% and 74%, respectively, which might be promising for long duration energy storage applications
- Radial machines have been identified as the most promising technology for the ECO-SEARCHERS cycles for mid power size (5 MW), and will be subject of more detailed investigation.

In the forthcoming months the Authors will proceed to finalise a bladeless expander prototype to be experimentally validated in a laboratory-scale CO2 loop in operating conditions relevant for the charging cycle operation, and will provide initial evidences on component operation in such kind of energy storage cycles.

Acknowledgements

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References

[1] Jafarizadeh, H., Soltani, M., Nathwani, J., 2023, A Novel Analysis of Energy Density Considerations and Its Impacts on the Cost of Electrical Energy Storage (EES) Plants, Energies, Vol. 16, p. 3330.
[2] Novotny, V., Basta, V., Smola, P., Spale, J., 2022, Review of Carnot Battery Technology Commercial Development, Energies, Vol. 15, p. 647.
[3] Alfani, D., Binotti, M., Macchi, E., Silva, P., Astolfi, M., 2021, sCO2 power plants for waste heat recovery: Design optimization and part-load operation strategies, Applied Thermal Engineering, Vol. 195, p. 117013.
[4] Kaushik, E., Prakash, V., Mahela, O.P., Khan, B., El-Shahat, A., Abdelaziz, A.Y., 2022, Comprehensive Overview of Power System Flexibility during the Scenario of High Penetration of Renewable Energy in Utility Grid, Energies, Vol. 15, p. 516.
[5] Zhou, C., Li, Y., Wang, F., Wang, Z., Xia, Q., Zhang, Y., Liu, J., Liu, B., Cai, W., 2023, A Review of the Performance Improvement Methods of Phase Change Materials: Application for the Heat Pump Heating System, Energies, Vol. 16, p. 2676.
[6] Shamsi, S.S.M., Barberis, S., Maccarini, S., Traverso, A., 2024, "Large scale energy storage systems based on carbon dioxide thermal cycles: A critical review", Renewable and Sustainable Energy Reviews, Vol.192, pp.114245_1-24.
[7] Barberis, S., Maccarini, S., Shamsi, S.S.M., Traverso, A., 2023, "Untapping Industrial Flexibility via Waste Heat-Driven Pumped Thermal Energy Storage Systems", Energies, Vol. 16(17), pp. 6249.
[8] Lettieri, C., Baltadjiev, N. D., Casey, M., and Spakovszky, Z. S., 2014,"Low-Flow-Coefficient Centrifugal Compressor Design for Supercritical CO2," ASME Journal of Turbomachanery, Vol. 136(8), p. 081008.
[9] G. Persico, P. Gaetani, A. Romei, L. Toni, E.F. Bellobuono, R. Valente, 2021, "Implications of Phase Change on the Aerodynamics of Centrifugal Compressors for Supercritical Carbon Dioxide Applications", ASME Journal of Engineering for Gas Turbines and Power, Vol. 143(4), p. 041007.
[10] S.D. Cich, J. Moore, J. Mortzheim, R. Singh, 2022, "Aeromechanical design of a 10 MWe SCO2 turbine", sCO2 Power Cycles Symposium, February 21-24, 2022, San Antonio, TX-USA.
[11] Baek, J. S., Groll, E. A., & Lawless, P. B., 2005, Piston-cylinder work producing expansion device in a transcritical carbon dioxide cycle. Part I: experimental investigation. International Journal of Refrigeration, Vol. 28(2), pp. 141-151.
[12] Fukuta, M., Anzai, F., Motozawa, M., Terawaki, H., & Yanagisawa, T., 2014, Performance of radial piston type reciprocating expander for CO2 refrigeration cycle, International journal of refrigeration, Vol. 42, pp. 48-56.
[13] Ferrara, G., Ferrari, L., Fiaschi, D., Galoppi, G., Karellas, S., Secchi, R., & Tempesti, D., 2016, Energy recovery by means of a radial piston expander in a CO2 refrigeration system, International Journal of Refrigeration, Vol. 72, pp. 147-155.
[14] Anselmi, E., Bunce, I., Pachidis, V., Zachos, P., & Johnston, M., 2018, An overview of the Rolls-Royce sCO2-test rig project at Cranfield University, 6th International Supercritical CO2 Power Cycles Symposium, Pittsburgh, USA.
[15] Vesely, L., Dostal, V., & Hajek, P., 2014, Design of experimental loop with supercritical carbon dioxide, proceedings of International Conference on Nuclear Engineering, ASME Paper ICONE22-30798, Vol. V003T05A023.
[16] Hasuike, H., Yamamoto, T., Fukushima, T., Watanabe, T., Utamura, M., & Aritomi, M., 2010, "Test plan and preliminary test results of a bench scale closed cycle gas turbine with super-critical CO2 as working fluid", ASME Paper GT2010-22171.
[17] http://www.sit-tesla-technologies.com/, last access 15/05/2024.
[18] Renuke, A., Pascenti, M., Traverso, A., Reggio, F., Silvestri, P. , 2024, Performance investigation of stator-less and blade-less radial expander, Journal of Engineering for Gas Turbines and Power, Vol. 146, pp. 011014.