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PERFORMANCE ANALYSIS OF PTES LAYOUTS EVOLVING SCO2 FOR INDUSTRIAL WHR INTEGRATION

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ABSTRACT

A consistent amount of renewable energy (RES) from non-predictable sources in the energy mix brings an increasing need of energy storage technologies to support grid stability. At the same time, electrification of industrial processes as well as the more and more common habit of industries to self-produce power via RES or CHP, can make industries a partner in disrupting grid stability. Thermo-mechanical storages can contribute through the use of traditional technologies (rotating machinery) employed in power plants, which are currently used to manage peak demand and grid services, and typically classify as hours-size storages, also capable of providing spinning reserve services to the electrical grid. Among such type of storages, Pumped Thermal Energy Storages (PTES) are a promising technology that enhance the concept of power-to-heat-to-power and long duration energy storage, and presents also different layouts and applications. This paper analyse the thermal performance of Pumped Thermal Electricity Storage (PTES) evolving supercritical CO2 (sCO2), comparing different layouts, while valorising waste heat (WH) sources, which are typically in temperature ranges of 100-400°C. WH temperature in this range are difficult to be exploited for traditional energy generation, but they are currently under investigation for the possibility to be valorised via High Temperature Heat Pump. In this sense this quality of Waste Heat could be valorised via PTES. In fact, the use of additional heat, otherwise dumped to ambient, may make the system capable of an apparent round-trip efficiency (RTE) higher than 100%. The use of sCO2 could enhance the techno-economic features of these systems, if compared to similar plants evolving steam or air. Starting from an identified reference case (a cement production plant with WH temperature to be valorized around 350°C), a sCO2-based PTES cycle is presented and analysed in this paper. The waste heat integration to the PTES system has been found to add an undeniable value in terms of RTE. The use of sCO2 enhances the techno-economic features of these systems, the independent charging and discharging system proposed in this study can also provide a keen sense of flexibility. At the same time, the valorisation of low temperature waste heat enables industries to enhance their energy efficiency, limit their operational costs and environmental impact, whilst becoming an active part in the regulation of the grid. Nevertheless, CAPEX of the proposed systems are still quite relevant and only a robust exploitation of the PTES in ancillary service market could attract industrial customers interest on sCO2 PTES.

INTRODUCTION

Waste heat recovery (WHR) is a direct way to increase industrial energy efficiency and promote EU industry decarbonization and it is already recognized as a best practice in many different industrial sectors particularly to valorise high temperature Waste heat (WH). Nevertheless, as shown in Figure 1, most of the discharged WH during industrial processes is qualified as low-grade heat (under 200 °C) which poses several technical challenges for its exploitation towards power production or internal re-use.

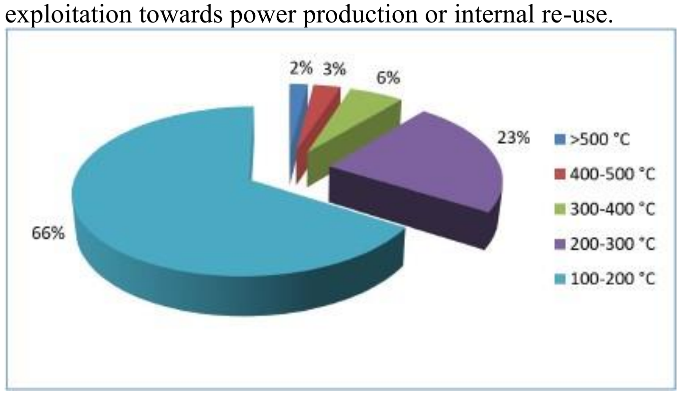

Figure 1: EU Industrial Waste Heat Potential Temperature distribution [1]

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A number of different technologies are available on the market depending on the source type, temperature range and end-use requirements, as shown in Figure 2.

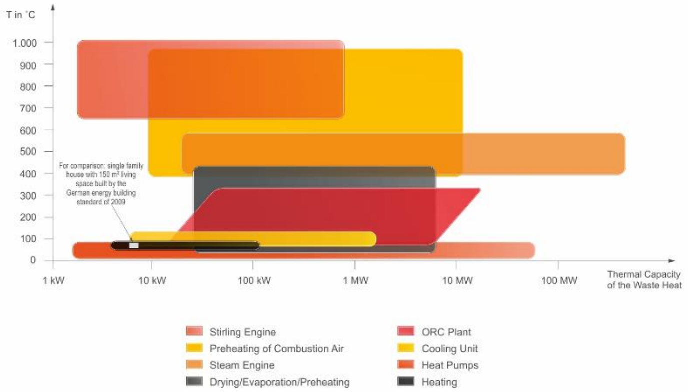

Figure 2: Waste Heat Recovery Technologies: temperature and thermal capacity classification [2]

Looking at medium grade WH (temperature ranges between 200-400 °C), which accounts for around 1/3 of EU WH, ORC (with quite low efficiency) seems to be the only way to valorise such WH. Nevertheless, sCO2 power cycles are gaining more and more interest as WH-to-Power [3], even if the higher conversion efficiencies are reached with WH temperature higher than 350-400°C [4] [5].

WH utilization can be better addressed via Heat Pumps (HPs), in which field CO2 (in trans critical and supercritical status) as a working fluid is being investigated for high temperature HPs [6]. The possibility to couple sCO2 HPs and power cycles for bulky energy storage [7] in so called Carnot Batteries [8], while integrating external heat inputs for example coming from Concentrated Solar Power (CSP) [9], has been recently more and more investigated.

Even if the possibility to exploit sCO2 power cycles for WHR applications is widely analysed [10], including different demonstration projects in US [11] and EU [12], the possibility of valorising WH via a sCO2 HP for Power-to-heat-to-power (P2H2P) purposes has not been investigated so far. Looking at the fact that: 1) there are more and more fluctuating/non-predictable RES that un-stabilize the grid; 2) self-generation of power via CHP and RES systems is becoming a best practice in different industrial sectors; 3) Electrification of industrial processes seems to be a relevant technological option to reduce fossil fuel consumption in industries, it is quite important to identify solutions that could make industries as grid flexibility actors, while enabling them the possibility to valorise local WH at this purpose.

sCO2 could make this possible, via WH P2H2P solutions exploiting sCO2 HPs and power cycles. Basing on previous thermoeconomic analysis of advanced sCO2 power cycles [13] and energy storage solutions [14] , in this paper an innovative layout and concept is proposed, aiming at valorising industrial WH and achieving attractive Round-Trip Efficiency (RTE).

PURPOSE OF THE STUDY AND CASE STUDY DESCRIPTION

In order to compare from a performance perspective the proposed WH driven P2H2P system with an existing sCO2 plant for WH2P system (CO2OLHEAT project [12]), a cement plant is considered as case study [15].

On a typical cement plant with a capacity of 5,000 t/day, the flue gas flow rate is 300,000 Nm^3/h with a temperature of 330°C and around 1/3 of exhaust air - "quaternary air" - representing 116,000 Nm^3 /h, which can be exploited thus having a WH source of around 10 MWth of maximum exploitable power at 330°C.

The idea is therefore to study a WH driven P2H2P system in which the waste heat acts as a heat source for the heat pump cycle that operates between the waste heat and the storage unit. The model is composed by: 1) a high temperature HP operating with sCO2 able to valorise available WH (CHARGING CYCLE); 2) a Molten-Salt (MS) High temperature Thermal Energy Storage (TES) able to store heat produced by the HP (STORAGE ASSET); 3) a sCO2 power cycle able to produce power once required exploiting the heat stored in the TES (DISCHARGING CYCLE). The goal of the study is to: 1) define sCO2 cycles operating conditions and design parameters considering the proposed test case, also investigating different WH2P sCO2 layouts (with or without recuperator); 2) analyse via a sensitivity analysis sCO2 cycles operating conditions and design parameters towards RTE maximization and WH optimal valorisation; 3) compare from a thermodynamic performance point of view the proposed WH driven P2H2P solution with "state of the art" sCO2 WH2P cycles.

PROPOSED CYCLE LAYOUTS

Figure 3 shows the charging cycle where the heat from the WH source of the cement plant is valorised via a heat pump increasing its temperature. Such heat is then stored in a Molten Salt TES (HITEC commercial molten salt). The arrows on the heat exchangers show the direction of transfer of heat. The heat is picked up from the waste heat recovery heat exchanger (WH HEX) and transferred to the thermal energy storage heat exchanger (TES HEX) to be then stored in the TES.

As a result of TES charging, the hot effluents from the cement process are released to the ambient at much lower temperature, for instance from 330°C down to 150-80°C, depending on the HP operating conditions.

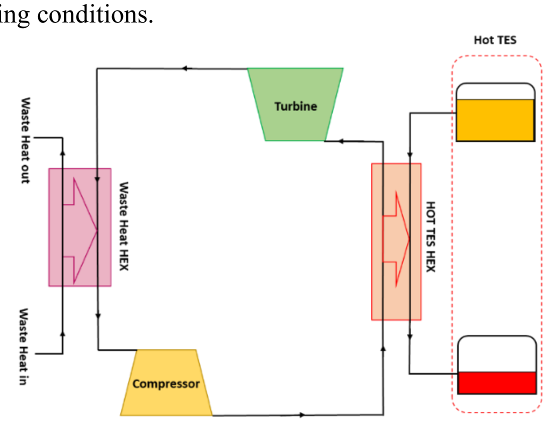

Figure 3: Charging cycle configuration

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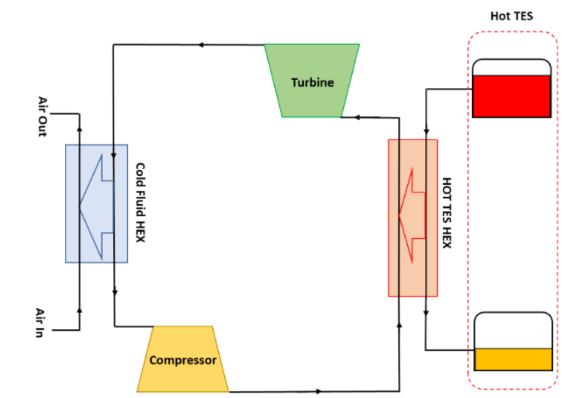
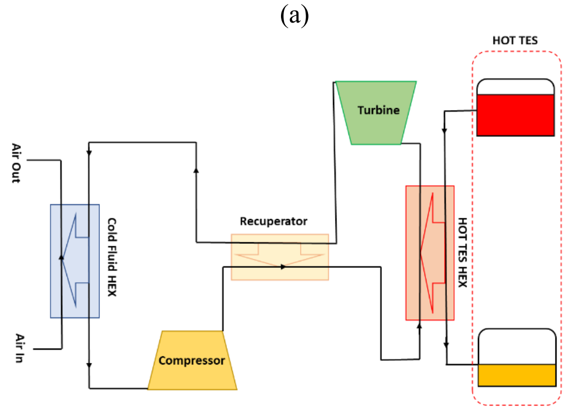
The charging cycle is followed by a discharging cycle. Figure 4 shows the three different configurations of the discharging cycle that will be studied to find out the most suitable configuration for this case study. Figure 4(a) shows a simple sCO2 discharging cycle valorising the heat stored in the TES: the sCO2 working fluid gets compressed and absorbs heat along TES HEX entering the sCO2 turbine, expanding and dissipating the remaining heat in the cooling HEX. The cooling HEX for the discharging operates at a much lower temperature than the WH temperature, thus significantly reducing the compressor work even if working with the same compression ratios.



(a)



(b)

Figure 4: Discharging cycle configurations: (a) simple discharge cycle (b) recuperated discharging cycle

In order to maximise the efficiency of the discharging cycle, a recuperated sCO2 cycle was analysed too. The recuperated heat is utilized right after the compressor, which makes the discharging heat utilization more efficient. The discharging cycle configurations were analysed searching for optimum performance in terms of electrical RTE.

MODELLING APPROACH DESCRIPTION

The modelling procedure used to get the thermodynamic properties of the cycles is mentioned in this section. Furthermore the economic assumptions and approach used for the component cost calculation are described in detail.

Cycle modelling technique and information flow

All the thermodynamic computations were done using a modified version of WTEMP-EVO, a component-based in-house thermo-economic simulation tool. It is developed in MATLAB®, integrating Coolprop [16] libraries for fluid properties, and it can simulate energy systems through the assembly of the desired layout, as explained in [17]. The tool 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.

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

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

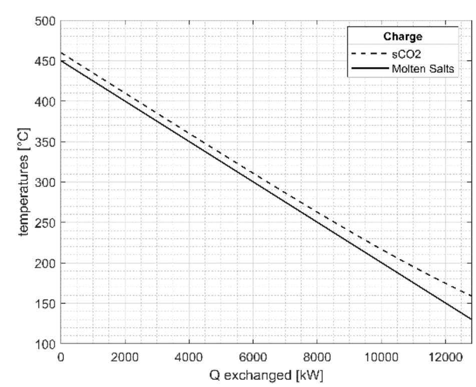
$$h_{Out} = h_{In} + \eta_{Turb} \cdot (h_{Out-isoentr} - h_{In}) \tag{3}$$

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

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

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

The maximum heat that can be exchanged by an heat exchanger ($Q_{HEX-max}$) is computed as the maximum amount of heat that can be transferred, from the hot to the cold fluid, in a counterflow heat exchanger that has an infinite area, thus leading to a temperature difference between the hot and cold fluid which is equal to zero in a certain point. If the properties of the fluids change throughout the heat exchanger, an internal pinch point can appear (e.g., when a peak is present in the value of the specific heat capacity of the hot fluid), and the computation of the maximum heat with this method allows to spot the temperature at which the internal pinch point can appear in the case of an ideal counterflow heat exchanger, and assign that value of that heat as the maximum possible.



(a)

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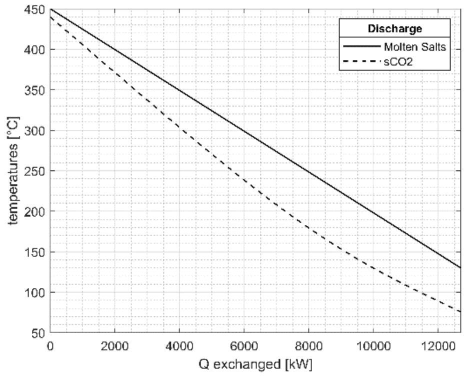

(b)

Figure 5: High temperature (TES HEX) Heat exchanger thermal exchange behaviour – (a) charging, (b) discharging

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.

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 [17].

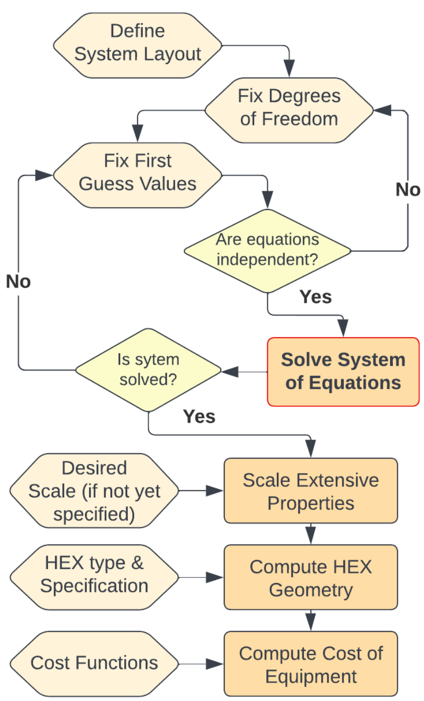

Figure 6: Algorithm flowchart of the modified WTEMP-EVO tool

Thermodynamic modelling assumptions

Since the PTES systems basically consist of two separate cycles, one for charge and one for discharge, the simulation use this approach of separating the computation of the two cycles, connecting the two by imposing the equivalence of the temperatures of the TES, and the amount of energy and mass stored in it.

In practice, the code firstly computes the discharge cycle, and then uses the results to initialize the computation of the charge cycle. To be more precise, since the discharge layout consists of a recuperated cycle, the minimum temperature of the TES depends, for both charge and discharge, on the inlet temperature of the Hot HEX in the discharging cycle, and thus on the effectiveness of the recuperator. For this reason the discharge is computed as first, and then minimum temperature of the TES is used to initialize the discharge cycle.

Modelling approach for this specific study consisted in separate calculations of the charging and the discharging cycles, in order to perform a sensitivity analysis on some identified parameters among the most relevant ones for each layout.

Considering WH available temperature, the commercial TES MS storage media HITEC XL, capable of operating temperatures between 130°C and 450°C [18], was identified giving thus priority to the utilisation of a commercial fluid.

In the charging cycle, the following variables has been selected to be varied in the sensitivity analyses: the two cycle operating pressures (maximum and minimum ones, considered at compressor extremes), and the maximum and minimum temperature of the TES material.

The assumptions for the charging and discharging cycle are shown in Table 1, while TES material properties are presented in Table 2.

Table 1 – Thermodynamic modelling assumptions

Assumptions	Value	UoM
TES Max temperature	450	°C
Recuperator Effectiveness	60 ; 80	%
Isentropic efficiency turbomachinery	80	%
Thermal losses of the TES	1	%
Electrical efficiency	98	%
Mechanical efficiency	98	%
Pressure loss in heat exchanger	2	%
Min ΔT Heat Exchangers	10	K
Compressor Inlet Temp.	35	°C
Ambient Temperature T_0	25	°C
Air Temperature Cooler Exit	45	°C
Waste Heat Temperature	330	°C
Waste Heat mass flow rate	38.6	kg/s

Table 2 – HITEC XL TES Material properties [19]

Maximum Temperature [°C]	450
Minimum Temperature [°C]	130
Density [kg/m³]	1877
Specific Heat [kJ/kgK]	1.426
Thermal conductivity [W/mK]	0.52
Cost [$/kg]	1.6

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The temperature and pressure levels of the discharging phase where chosen in order to maximise the behaviour of this cycle, and then a corresponding charging cycle was properly studied and selected. In fact, this was possible due to the fact that the charging and discharging cycles were considered fully decoupled, with respect to a standalone PTES, because of the possibility of integration with WH at high temperature.

In the following paragraphs, performance sensitivity analyses of charging and discharging cycles are presented, targeting the matching of the two cycles, and starting from the discharging phase.

Performance is judged mainly in terms of electrical RTE (i.e. accounting only for the electrical energy flows) and in terms of exergetic RTE (i.e. accounting for electrical and thermal exergy flows); the exergetic efficiency of a cycle η_{ex} is used for the evaluation of the direct utilisation of the waste heat. These parameters are defined by the following equations, where E is the energy, P the power, Δt is the charging and discharging time, T_0 is the ambient temperature, and $\dot{Q}_{WH}$ is the heat flow rate absorbed from the waste heat gases:

$$RTE_{el} = \frac{E_{DC}}{E_{CC}} = \frac{P_{DC} \cdot \Delta t_{DC}}{P_{CC} \cdot \Delta t_{CC}} = \frac{P_{DC}}{P_{CC}} \quad (1)$$

$$RTE_{ex} = \frac{P_{DC}}{P_{CC} + \dot{Q}_{WH} \cdot \left(1 - \frac{T_0}{T_{avg}}\right)} \quad (2)$$

$$\eta_{ex} = \frac{P}{\dot{Q}_{WH} \cdot \left(1 - \frac{T_0}{T_{avg}}\right)} \quad (3)$$

$$T_{avg} = \frac{\int_{in}^{out} T ds}{s_{out} - s_{in}} \cong \frac{h_{out} - h_{in}}{s_{out} - s_{in}}; \quad if\ p \cong const. \quad (4)$$

In fact, considering equal duration of charging and discharging phases, the electrical RTE can be easily calculated on the electrical total power (consumed and produced respectively) of the two cycles (instead of based on total electrical energy values), as the two cycles are obtained considering the same storage dimension and thus the same mass flow rate of the TES fluid, in this case. Similarly, the exergetic RTE can be derived basing on electrical power and exergy flows. The exergetic RTE is computed considering the actual WH input to the cycle and, since the heat exchange does not occur at constant temperature, the corresponding thermodynamic-average temperature T_{avg} at which the heat exchange can be considered to occur, computed as in Eq. 4 [20].

Cost assumptions

In order to evaluate the CAPEX of the system, further than HEX cost functions as presented in [18-19], typical sCO2 power cycle components (turbine – compressor – recuperator) cost functions have been considered here, properly correcting them (particularly once studying "hot compressor" and "cold turbines" in HP/charging cycles) according to literature "correction factor" approach [21] in order to take into account different materials used to manufacture components in operating conditions that are different than usual ones. Cost function presented in [22] were therefore multiplied and divided by a correction factor of 2.035 and 1.764 respectively to evaluate compressor and turbine CAPEX, considering the different material/operating temperature once such components are operating in HP charging cycle.

DISCHARGING CYCLE (DC) - SUPERCRITICAL CO2 POWER CYCLES (WITH/WITHOUT RECUPERATOR)

For what concerns the discharging cycles, typical values for sCO2 where selected, leading to a maximum pressure considered of 250 bar (this value has been identified by the authors following their experiences in previous study [13], for technological reasons in order not to face too much challenging operation conditions – e.g. manageable compression ratios, use of pressurized HEXs…). Moreover, considering aforementioned assumptions, the maximum CO2 temperature was selected to be 440°C, while a typical value of 35°C was selected for the compressor inlet temperature, to ensure a stable behaviour of the compressor itself.

The objective of the analysis was to investigate the points in which a match with the charging cycle (to be presented hereafter) can lead to a maximisation of the electrical RTE.

Figure 7 shows the electrical power achievable by a simple CO2 cycle given a constant mass of molten salts as heat source, with respect to the minimum pressure of the cycle and the minimum TES temperature. From this analysis, while keeping constant all the other variables, the best point for the discharging cycle is identified targeting maximisation of produced power: not surprisingly, the best minimum pressure corresponds closely to the CO2 critical pressure.

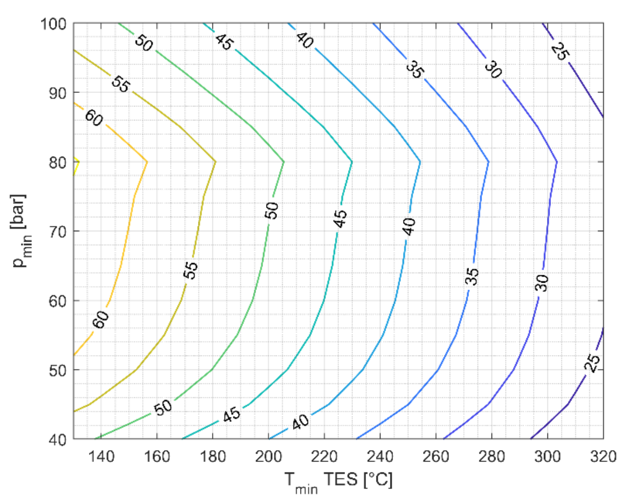

Figure 7: Total net power that can be achieved by a simple cycle for a 1 kg/s mass flow rate of the molten salts.

Looking at a recuperated cycle, the value of the minimum temperature of the TES is dependent mainly on the grade of recuperation chosen for the cycle.

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The following figures represent the values of this temperature (Fig. 8) and the values of efficiency (Fig.9) and net power (fig.10) that can be achieved by the systems.
The best results are achieved near 80 to 85 bar. While the efficiency of the cycle increases with the recuperator effectiveness, this parameter does not influence much the total power that can be extracted by the TES source, given a constant total mass.
Two extreme values of the recuperator effectiveness are then chosen to represent the behaviour of the recuperated layout: 60% and 80% effectiveness. From those, an optimal value of minimum pressure ensuring the maximum achievable power was chosen equal to 83 bar, being in the above mentioned range as well as guaranteeing a proper compressor operation according to authors' experience [13].

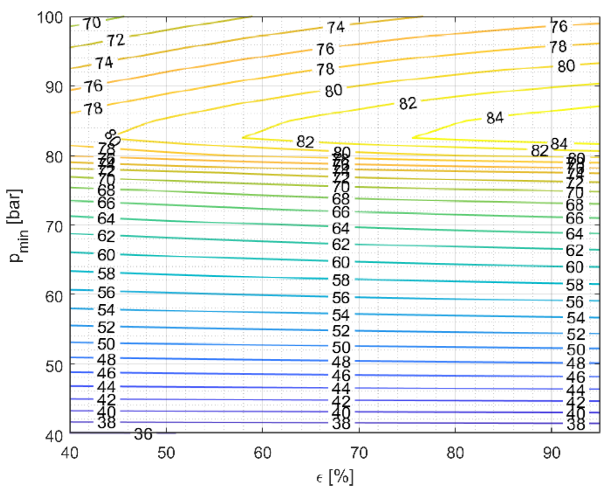

Figure 8: Total net power that can be achieved by a recuperated cycle for a 1 kg/s mass flow rate of the molten salts; abscissa is the effectiveness of the recuperator.

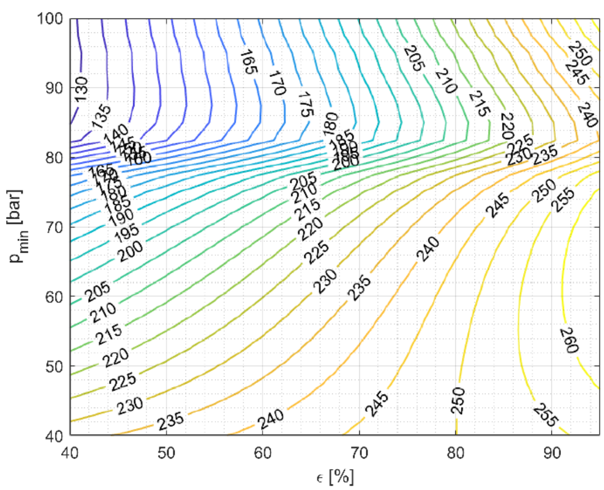

Figure 9: Heater inlet temperature in a recuperated cycle; abscissa is the effectiveness of the recuperator.

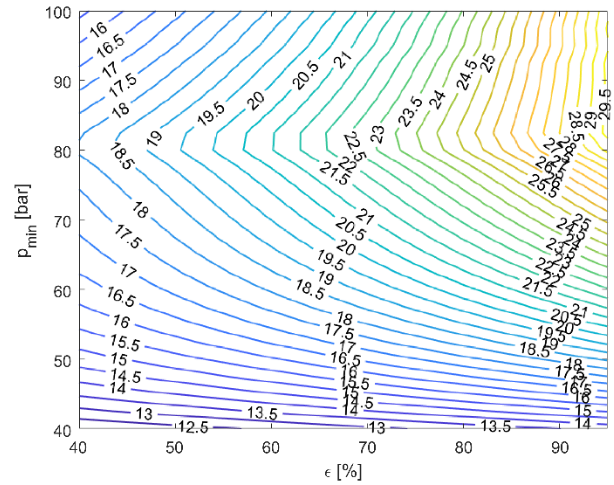

Figure 10: Thermal efficiency in a recuperated cycle; abscissa is the effectiveness of the recuperator.

CHARGING CYCLE (CC) – SUPERCRITICAL CO2 HEAT PUMP CYCLES

The analyses of the discharging cycles (targeting net power maximisation) allowed to select three values of minimum temperature of the TES to be analysed in the charging cycle.
For each discharging case, a corresponding charging one was analysed, fixing the minimum temperature of the TES at 130°C for the first case, related to the simple discharging one. For the second and third charging cases, corresponding respectively to discharging cycles with low (60%) and high (80%) effectiveness of the recuperator, the TES minimum temperature was assumed equal to 183°C and 222°C, respectively. Since the behaviour of the analysed parameters is similar in all the cases, only the simple discharging and the 80% recuperated discharging cycles results are presented in the following paragraphs.
A common operating performance value to be monitored during the charging phase for all these cases is the compressor outlet temperature . Such temperature, having set the values of the other temperatures, varies with the pressures, but only the values higher than or equal to 460°C were taken into account, given the initial assumptions related to the identified MS TES storage media. In all the following figures the line related to the minimum temperature is drawn in red, separating the not-acceptable values (above) and the real values (below), while the blue region represents a zone with pressure ratio too close to 1 or even lower.

Thus, it is clear that the initial hypothesis on the temperature strongly affects the behaviour of the charging cycle HP and a sensitivity on different values of temperature were needed. Moreover, this behaviour is also highly related to the assumptions on the machinery efficiency, which affect the performance of the heat pump cycle.

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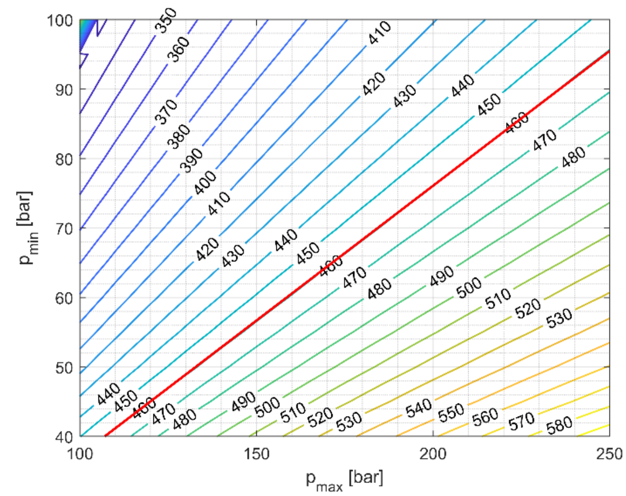

Figure 11: sCO2 Maximum temperature in the charging cycle. The minimum allowable value is 460 °C.

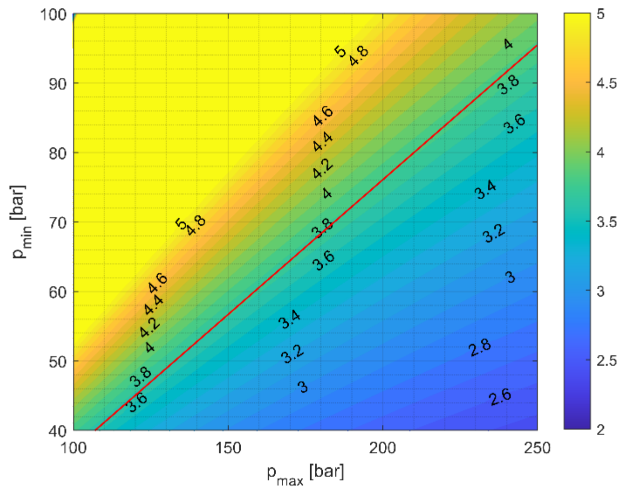

Figure 12: COP of a charge with min Temp of the TES equal to 130°C (acceptable area is below the red line).

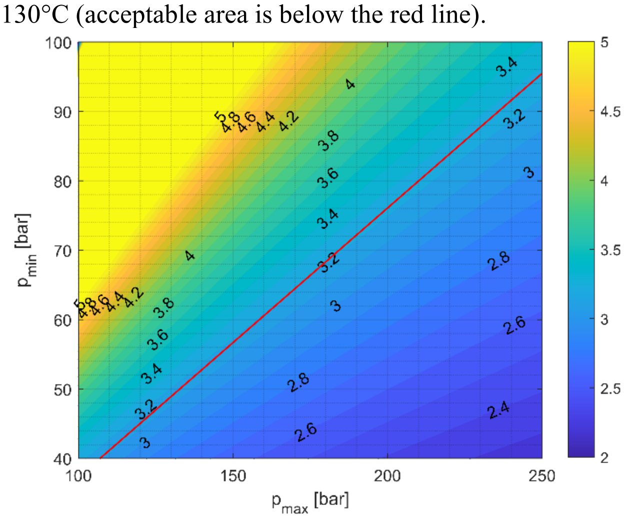

Figure 13: COP of a charge with min Temp of the TES equal to 222°C (acceptable area is below the red line).

Analysing the figures representing the HP net power consumption trend, it is worth highlighting that, as for the COP behaviour, power values are quite aligned with temperature values. For example, a value close to 460°C (lowest acceptable sCO2 temperature) foresees a pressure ratio close to 2.6, and leads to a COP between 3.7 and 3.9.

This is interesting mainly for two reasons: first, a simple thermodynamic optimum would lead to choose high pressure, while a thermo-economic point of view could move the selected point towards lower pressures only for the charging cycle; second, operating at low pressure can increase the influence of the variation of thermophysical properties of sCO2, and in particular of the specific heat, potentially leading the heat exchangers to an internal pinch point lower than the assumed values of temperature difference at the extremes. This would imply a more detailed analysis of the heat exchangers, potentially involving introduction of multiple heat exchanger and TES units, carrying different mass flows to better couple the CO2 heat capacity rate variation.

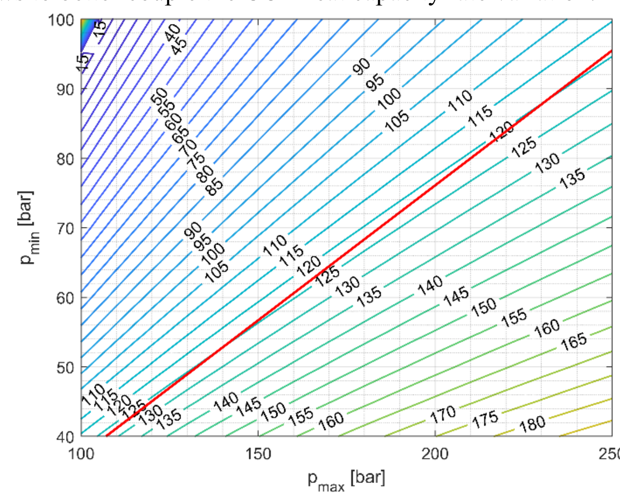

Figure 14: Net Power absorbed in a charge with min Temp of the TES equal to 130°C (acceptable area is below the red line).

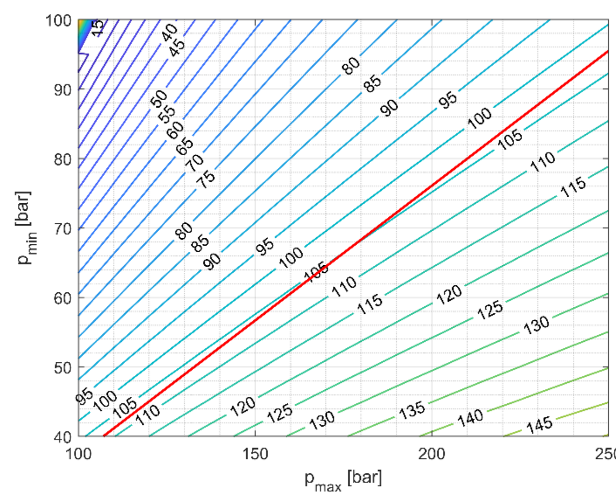

Figure 15: Net Power absorbed in a charge with min Temp of the TES equal to 222°C (acceptable area is below the red line).

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WH DRIVEN P2H2P SYSTEM

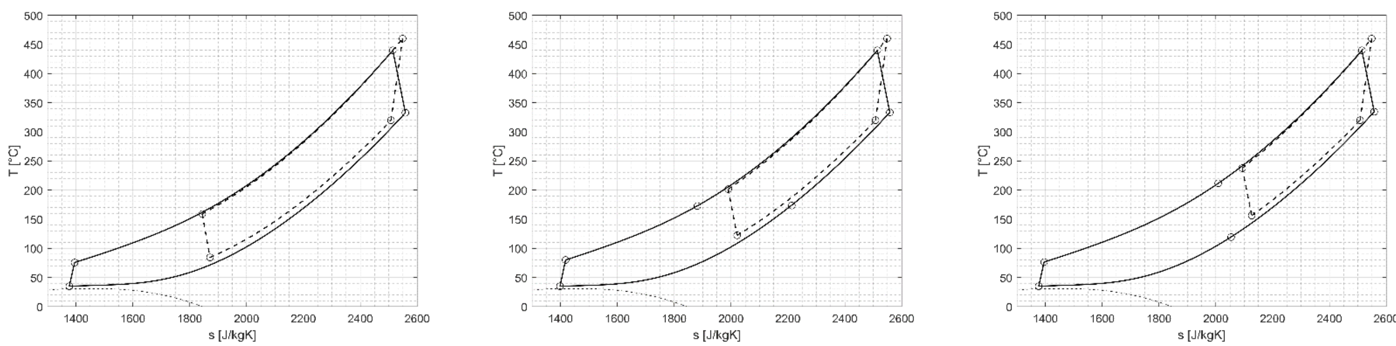
All the previous analyses conducted for the discharging and the related charging cycles led to the definition of the most effective P2H2P cycle operating points (Figure 16), starting from the initial assumptions and targeting the maximisation of the apparent electrical RTE, thus not considering the amount of the used WH.

The results obtained are then presented in the following tables, always presenting three cases for the discharging cycles:

I) simple cycle

II) recuperated cycle with 60% recuperator effectiveness

III) recuperated cycle with 80% recuperator effectiveness



(a) (b) (c)

Figure 16: T-s Diagrams for CASE I (a), II (b), III (c)

First, Table 3 presents the values of minimum pressures and TES temperature identified as the best operating parameters. The maximum optimal pressure resulted to be 250 bar for all the cycles (both in charging and discharging phase), i.e. the maximum allowed. Table 4 instead presents the values obtained in CC for the absorbed heat power from waste heat and the resulting temperature at which it is then released, and the values, for DC, of the temperature at which the heat is released. These temperatures are all above 100°C and this is due to the fact that the cycle layouts analysed rely on a Brayton-like configuration and thus they release heat to the cold source in a sensible way. These values suggest a possible further utilisation as process heat, or even the investigation of different layouts here not considered.

Table 3 - Minimum pressures and minimum TES temperature, for the best operating conditions

CASE	Layout	CC p_{min}	DC p_{min}	TES Tmin
I	CC + SDC	95.5 bar	83 bar	130 °C
II	CC + RDC (60%)	95.5 bar	83 bar	183 °C
III	CC + RDC (80%)	95.5 bar	83 bar	222 °C

Table 4 - Main parameters of heat exchange with the cold sources

CASE	CC WH Power	CC TES HEX Power	CC WH min Temp	DC T_in Cooler
I	9.35 MW_{th}	12.8 MW_{th}	94.5 °C	332.7 °C
II	7.87 MW_{th}	11.2 MW_{th}	132 °C	173.5 °C
III	6.54 MW_{th}	9.6 MW_{th}	166 °C	119 °C

The values of electrical RTE (presented in Table 5) show low performance for the case I, the one coupling a simple charging cycle (CC) and a simple discharging one (SDC). Better results are instead achieved using a recuperated discharging cycle (RDC), leading to values even higher than 70%. The utilisation of WH in a PTES is thus confirmed to allow reaching values of electrical RTE otherwise impossible with standalone PTES configuration, considering similar general assumptions. This comes at the cost of using the source of heat, and thus the exergetic RTE better represents the actual use of all the energy inputs involved in the process. Between the solutions investigated, the highly recuperated one shows to be most efficient exergetic RTE, despite the result does not show a good utilisation of the energy inputs, reaching values lower than 40%.

Table 5 – Net Powers and electrical RTE for the optimum points applied to the specific case study

CASE	DC Net Power	CC Net Power	RTE (electrical)	RTE (exergetic)
I	1.63 MW	3.34 MW	49.0%	24.8 %
II	2.10 MW	3.16 MW	66.4%	34.0 %
III	2.18 MW	2.98 MW	73.3%	38.8 %

Finally, it is worthy to underline the relevance of heat exchanger assumptions (Fig.5), since their design performance significantly impacts on cycle performance and should take into account the sCO2 real-gas behaviour. This aspect would deserve a separate dedicated investigation.

CAPEX ESTIMATIONS

Table 6 shows the capital cost of the components needed for the three analysed cases. All the values are in M$ and they are

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calculated using the cost functions described in previous sections as well as components sizes from thermodynamic calculations. A 100 MWh TES have been considered. Cost of the compressor and turbine depends on the power output. Heat exchangers costs are based on the UA. The UA value was calculated for the counter flow for all the heat exchangers using log mean temperature difference (LMTD) method, even though this can cause an under-sizing of the heat exchangers where a consistent change of fluid properties occur. According to the power requirement internally geared (IG) centrifugal compressor were used for both charging and discharging while axial turbines were used in both the cases.
The highest cost for all the components is of the TES System, both in terms of TES capacity and TES HEX.

Table 6 – Estimated CAPEX

Components	**I CC + SDC**	**II CC + RDC (60%)**	**III CC + RDC (80%)**	**WH2P (RC 80%)**
CC Compressor	4.58	4.66	4.69	\
CC Turbine	0.11	0.13	0.15	\
WH_HEX	1.58	1.39	1.21	1.59
DC Compressor	1.08	1.25	1.24	1.33
TES HEX	1.29	1.80	1.60	\
DC Turbine	0.29	0.34	0.35	0.33
Air Cooler HEX	0.36	0.48	0.54	0.65
Recuperator	/	0.16	0.31	0.35
TES	4.34	4.34	4.34	/
OVERALL	13.65	14.55	14.43	4.26

This is evident by the figure 5, which shows the internal features of the heat exchanger. The temperature difference between the hot side fluid and cold side fluid remain more or less 10K for the whole duration of heat transfer, which tend to make the area of the heat exchanger unusually large. This TES heat exchanger referred in table 6 as TES HEX is similar for the charge and discharge cycle. Although the pressures are not similar for charge and discharge, the UA is calculated for the larger value of pressure which belongs to the discharge cycle. The parameters of pressure and mass flow rate are different, whereas the temperature difference between the inlet and the outlet of TES HEX are same for the charging and the discharging cycle.
Although reversible machinery is being tested for lab scale PTES systems [23], large scale reversible sCO_2 machinery are not technologically ready yet for large scale plants. Therefore separate turbomachinery for charging and discharging are considered-

BENCHMARKING WITH OTHER WHR ENERGY SYSTEMS SOLUTIONS

A first comparison of the proposed WH driven P2H2P with direct utilisation of the WH for power production can be done basing on one of the cycle layouts used for the P2H2P solution, in particular the simple recuperated cycle. In fact, despite a recuperated cycle does not maximise the WH exploitation, it has been proven to be a viable solution on a thermo-economic point of view [17]. As it can be seen by comparing the tables, the utilisation of the P2H2P system leads to an increase of the net power in discharging phase, if compared to a WH2P sCO2 system, due to the fact that sCO2 TIT is higher.

Table 7 - Results of recuperated cycles applied to the case-study waste heat source

CASE	p_{min}	Net Power	Efficiency	WH T_out	Exergetic efficiency
RDC (60%)	83 bar	1.71 MW	16.2 %	137 °C	42.6 %
RDC (80%)	83 bar	1.76 MW	18.6 %	157 °C	47 %

Looking at exergy utilisation as a term of comparison for the two different solutions applied to the available waste heat, it can be seen that the direct WH2P solutions are able to achieve higher values for the case study analysed. Thus, this can lead to the conclusion that the system analysed is not competitive with a WH2P utilisation on a purely thermodynamic point of view.

It is also worthy to mention that such a WH2P cycle (if whose costs are estimated via the same costs functions as reported in Table 6) could have a CAPEX around 4.26 M$ which is significantly lower (at least around -60% of CAPEX) if compared to PTES CAPEX expressed in table. Nevertheless, it is relevant to highlight that such type of plant could not be operated in a flexible way on the electric market, thus not featuring any grid support service which are usually more remunerative power production revenue lines. A more detailed thermo-economic analysis needs to be carried out for a more comprehensive comparison.

CONCLUSIONS

This work analyses, from a thermodynamic performance and CAPEX point of view., WH driven sCO2 P2H2P cycles layouts at on-design conditions, presenting a specific case study on cement industry. Sensitivity analyses were made in order to explore the performance features of the charging and the discharging cycles that constitute a PTES system. Performance has been judged mainly in terms of electrical RTE (i.e. accounting only for the electrical energy flows) and in terms of exergetic RTE (i.e. accounting for electrical and thermal exergy flows). In particular, attention is paid to the electrical RTE enhancement potential when the P2H2P system is coupled with industrial waste heat recovery
Based upon the results obtained from such analyses, the most effective operating conditions (targeting electrical RTE maximisation) for the integrated system are presented, for each of the combinations investigated.

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Electrical RTE higher than 70%, are obtained envisaging the use of a recuperated cycle in the discharging phase, higher than what can be achieved with standalone PTES systems working in similar conditions. Moreover, a recuperated solution achieves a better result with a limited increase of the TES minimum temperature and thus of its dimension. Actually, the presented solution, leveraging on availability of WHR, can also achieve the elimination of a TES at low temperature, necessary in standalone PTES configurations (thus bringing to a CAPEX saving of the PTES). These results come of course at the cost of the utilisation of WH together with the net power of the charging phase: exergetic RTE shows that even the best of the analysed configurations cannot achieve a result higher than 45%.

For the same WHR case study condition, a simple WH2P configuration using the same recuperated cycle layout achieve exergetic efficiencies of 45-50%, showing a better exploitation of its exergetic inputs with respect to the PTES system, for the same case study. This suggest further analyses to be conducted to investigate different temperature levels (also at high Temperature TES level), or even different cycle layouts, to achieve a more comprehensive comparison of the solutions.

On the other side, it is worthy to remark that the utilisation of WH PTES is able to decouple the “power production” and the “power utilization/storage” with respect to a traditional WH2P solution, . Therefore, despite higher CAPEX, the attractiveness of the more complex WH PTES solution against the conventional WH2P solution might become clear from a more focused and dedicated thermoeconomic analysis, considering the additional possibility of selling services to the electrical grid and/or the additional flexibility in modulating the electrical power flows depending on the electrical market prices (night/day price profiles).

NOMENCLATURE

CC	Charging Cycle
COP	Coefficient of Performance
DC	Discharging Cycle
HEX	Heat Exchanger
HP	Heat Pump
MS	Molten Salt
P2H2P	Power to Heat to Power
p_{max}	Maximum Pressure
p_{min}	Minimum Pressure
PTES	Pumped Thermal Energy Storage
RDC	Recuperated Discharging Cycle
RES	Renewable Energy Sources
RTE	Round Trip Efficiency
TES	Thermal Energy Storage
WH	Waste Heat
WH2P	Waste Heat to Power
WHR	Waste Heat Recovery

REFERENCES

[1] C. Haddad, C. Périlhon, A. Danlos, M.-X. François, and G. Descombes, “Some Efficient Solutions to Recover Low and Medium Waste Heat: Competitiveness of the Thermoacoustic Technology,” *Energy Procedia*, vol. 50, pp. 1056–1069, Jan. 2014, doi: 10.1016/j.egypro.2014.06.125.

[2] https://www.waste-heat.eu/about-waste-heat/waste-heat-technologies - Accessed: 11 Jan. 2023. [Online].

[3] M. T. White, G. Bianchi, L. Chai, S. A. Tassou, and A. I. Sayma, “Review of supercritical CO2 technologies and systems for power generation,” *Appl. Therm. Eng.*, vol. 185, p. 116447, Feb. 2021, doi: 10.1016/j.applthermaleng.2020.116447.

[4] D. Alfani, M. Binotti, E. Macchi, P. Silva, and M. Astolfi, “sCO2 power plants for waste heat recovery: design optimization and part-load operation strategies,” *Appl. Therm. Eng.*, vol. 195, p. 117013, Aug. 2021, doi: 10.1016/j.applthermaleng.2021.117013.

[5] M. Marchionni, G. Bianchi, and S. A. Tassou, “Review of supercritical carbon dioxide (sCO2) technologies for high-grade waste heat to power conversion,” *SN Appl. Sci.*, vol. 2, no. 4, p. 611, Mar. 2020, doi: 10.1007/s42452-020-2116-6.

[6] C. Mateu-Royo, C. Arpagaus, A. Mota-Babiloni, J. Navarro-Esbrí, and S. S. Bertsch, “Advanced high temperature heat pump configurations using low GWP refrigerants for industrial waste heat recovery: A comprehensive study,” *Energy Convers. Manag.*, vol. 229, p. 113752, Feb. 2021, doi: 10.1016/j.enconman.2020.113752.

[7] P. Tafur-Escanta, R. Valencia-Chapi, M. López-Guillem, O. Fierros-Peraza, and J. Muñoz-Antón, “Electrical energy storage using a supercritical CO2 heat pump,” *Energy Rep.*, vol. 8, pp. 502–507, Jun. 2022, doi: 10.1016/j.egyr.2022.01.073.

[8] T. Liang *et al.*, “Key components for Carnot Battery: Technology review, technical barriers and selection criteria,” *Renew. Sustain. Energy Rev.*, vol. 163, p. 112478, Jul. 2022, doi: 10.1016/j.rser.2022.112478.

[9] “Supercritical CO2 Heat Pumps and Power Cycles for Concentrating Solar Power: Preprint,” *Renew. Energy*, p. 13, 2020.

[10] L. Liu, Q. Yang, and G. Cui, “Supercritical Carbon Dioxide(s-CO2) Power Cycle for Waste Heat Recovery: A Review from Thermodynamic Perspective,” *Processes*, vol. 8, no. 11, Art. no. 11, Nov. 2020, doi: 10.3390/pr8111461.

[11] Alex Kacludis, Sean Lyons, Dan Nadav, and Edward Zdankiewicz, “Waste Heat to Power (WH2P) Applications Using a Supercritical CO2-Based Power Cycle.,” in *Proceedings of Power-Gen International 2012*, Orlando, FL U.S.A., Dec. 2012.

[12] “Home,” *CO2OLHEAT*. https://co2olheat-h2020.eu/ (accessed Oct. 27, 2022).

[13] Guédez, R, Barberis, S, Maccarini, S, López-Román, A, Milani, A, Pesatori, E, Oyarzábal, U, & Sánchez, A. "Design of a 2 MW Molten Salt Driven Supercritical CO2 Cycle and Turbomachinery for the SOLARSCO2OL Demonstration Project." Proceedings of the ASME Turbo Expo 2022: Turbomachinery Technical Conference and

DOI: 10.17185/duepublico/77306

Exposition. Volume 9: Supercritical CO2. Rotterdam, Netherlands. June 13–17, 2022. V009T28A008. ASME. https://doi.org/10.1115/GT2022-82013

[14] M. Manzoni, A. Patti, S. Maccarini, and A. Traverso, “Analysis and comparison of innovative large scale thermo-mechanical closed cycle energy storages,” *Energy*, vol. 249, p. 123629, Jun. 2022, doi: 10.1016/j.energy.2022.123629.

[15] “Waste heat recovery - FIVES PILLARD - PDF Catalogs | Technical Documentation | Brochure.” https://pdf.directindustry.com/pdf/fives-pillard/waste-heat-recovery/196705-752096.html (accessed Oct. 27, 2022).

[16] Welcome to CoolProp — CoolProp 6.4.3 documentation.” http://www.coolprop.org/ (accessed Jan. 04, 2023).

[17] G. Baglietto, S. Maccarini, A. Traverso, and P. Bruttini, “Techno-Economic Comparison of Supercritical CO2, Steam, and Organic Rankine Cycles for Waste Heat Recovery Applications,” *J. Eng. Gas Turbines Power*, vol. 145, no. 4, Dec. 2022, doi: 10.1115/1.4055727.

[18] A. Caraballo, S. Galán-Casado, Á. Caballero, and S. Serena, “Molten Salts for Sensible Thermal Energy Storage: A Review and an Energy Performance Analysis,” *Energies*, vol. 14, no. 4, Art. no. 4, Jan. 2021, doi: 10.3390/en14041197.

[19] Deliverable D3.3_2019.01 "Thermal storage for improved utilization of renewable energy in steam production " - HighEFF- Centre for an Energy Efficient and Competitive Industry for the Future - Norway's Centre for Environment-friendly Energy Research (FME) - Project co-funded by the Research Council of Norway and Industry partners

[20] Bejan, A., Tsatsaronis, G. and Moran, M. (1996) Thermal Design & Optimization. John Wiley & Sons Inc., New York.

[21] N. T. Weiland, B. W. Lance, and S. R. Pidaparti, “sCO2 Power Cycle Component Cost Correlations From DOE Data Spanning Multiple Scales and Applications,” presented at the ASME Turbo Expo 2019: Turbomachinery Technical Conference and Exposition, Nov. 2019. doi: 10.1115/GT2019-90493.

[22] Richard Turton, Richard C. Bailie, Wallace B. Whiting, Joseph A. Shaeiwitz, *Analysis Synthesis and Design of Chemical Processes 5th Edition*. Pearson Education.

[23] D. Steger, M. Feist, and E. Schlücker, “Using a screw-type machine as reversible compressor–expander in a Carnot Battery: An analytical study towards efficiency,” *Appl. Energy*, vol. 316, p. 118950, Jun. 2022, doi: 10.1016/j.apenergy.2022.118950.

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