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PRELIMINARY ANALYSIS OF AN ELECTRO-THERMAL ENERGY STORAGE SYSTEM FOR GEOTHERMAL APPLICATIONS

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ABSTRACT

The shift towards renewable energy sources, particularly solar and wind, is reshaping the power generation sector. However, the intermittent nature of these sources necessitates their integration with flexible and dispatchable power generation systems and efficient energy storage solutions, to ensure grid stability and support Transmission System Operators (TSOs), enhancing both reliability and profitability. In response several different storage technologies are currently undergoing active research and development. The Horizon Europe SEHRENE project investigates electro-thermal energy storage (ETES) through the development of an innovative Carnot Battery system. During the charging phase, a high-temperature heat pump is used to store heat exploiting renewable excess electricity, while during the discharging phase an Organic Rankine Cycle (ORC) exploits the stored heat to flexibly produce electricity and respond to load variations. These two thermodynamic cycles are coupled by means of an advanced Thermal Energy Storage (TES) configuration featuring a single heat exchanger that utilizes a Phase Change Material (PCM) to achieve high energy density, integrating Kelvin cell structures to further enhance heat transfer efficiency. The project aims to demonstrate the ETES concept across three different use cases: (i) the Kızıldere II geothermal power plant in Turkey, (ii) a ceramic production facility in Alcora, Spain, and (iii) a Distribution System Operator (DSO) substation supporting an industrial park with photovoltaic energy production near Granada, Spain. The optimal system configuration is determined for each application according to the specific heat source availability and use-case requirements. This work focuses on the Kızıldere II geothermal power plant, presenting a preliminary ETES system design and detailing the selection criteria for the ETES working fluid. In the first part, aimed at working fluid selection, a simplified approach for PCM heat exchanger sizing is adopted. This study evaluates system performance, particularly focusing on the off-design of PCM heat exchanger and its influence on round trip efficiency (RTE). In the second part of the work, to improve calculation accuracy, the detailed design of the PCM heat exchanger is carried out considering the preselected working fluid, and RTE is recalculated. The findings of this work aim to numerically validate the feasibility of PCM heat exchanger technology and the potential of ETES systems for geothermal applications, also providing some valuable insights for optimizing similar systems in different industrial contexts.

1 INTRODUCTION

Electro-Thermal Energy Storage (ETES) is a novel energy storage concept embracing waste heat valorization, thermal energy storage (TES) and low temperature thermal energy conversion into electrical energy aiming to support a more sustainable energy system from both an economical and environmental point of view. This technology integrates electricity storage and low-temperature industrial waste heat recovery, taking advantage of a latent thermal energy storage system to flexibly meet the demand of the energy market, also helping to solve typical unbalancing and stability issues resulting from the increasing penetration of variable renewable energy plants in the current European power generation scenario.
The strength of this concept lies in its simplicity and potentially boundless application, as it does not require specific geographical conditions nor critical raw materials for its construction and operation.
Few scientific works dealing with thermal storage systems integrated in geothermal energy power plants

are currently available. Some works consider TES coupled with geothermal energy as an alternative energy source to provide heating and cooling to meet the energy demands in buildings [1]. On the other hand, regarding the implementation of Phase Change Materials (PCM) in heat exchangers, one interesting example presents a latent heat storage system to increase the energy efficiency and reduce the environmental impact of an energy-intensive industry by recovering and storing waste heat from combustion gases [2]. Moreover, few recent studies highlight the potential of PCMs as a promising solution for thermal energy storage in concentrated solar power (CSP) systems [3]. Conversely, PCM-TES applications in heat exchanger design for low-temperature (<250 °C) thermal-to-electric conversion in ORC systems remain largely unexplored, making Electro-Thermal Energy Storage a promising alternative to conventional Battery Energy Storage Systems (BESS). The successful operation of an ETES system requires an accurate simulation model and a sophisticated optimization procedure for its design parameters, considering many operating constraints, such as electricity prices, waste heat availability and ambient conditions, among others.

With the aim of future implementation in a digital twin, this study presents an ad-hoc numerical model for the performance assessment of both charging and discharging cycles, finally evaluating the system round trip efficiency. This study is performed in the context of the SEHRENE project, a Horizon Europe funded project which aims at demonstrating the PCM-based ETES technology TRL 4 with a 10 kW heat pump prototype and 20 kWh PCM TES prototype. This project will study the detailed design of the ETES systems for three different applications, and this work refers to the specific assumptions and boundary conditions defined for one of them, the Kızıldere II geothermal power plant, located in Turkey.

The key component of the investigated ETES system is the PCM heat exchanger, as this component is shared between the charging and discharging systems, where it operates as condenser and evaporator, respectively. The charging system is made by a High Temperature Heat Pump (HTHP), which upgrades the heat coming from the unexploited liquid brine of the geothermal plant, before this stream is sent to the reinjection well. The upgraded waste heat is transferred to the PCM TES by the working fluid flowing and condensing inside the tubes of a single pass heat exchanger. Two additional heat exchangers are included in the charging circuit: the HTHP evaporator in which the heat source, namely the geothermal brine, delivers thermal power to the working fluid; and an internal heat exchanger used to superheat the working fluid vapor at compressor inlet using the working fluid residual heat at condenser outlet. Thanks to this feature, the compressor avoids any potential issue related to liquid droplets formation even when working with fluid characterized by a high molecular complexity. An expansion valve completes the heat pump equipment, expanding the working fluid down to evaporation pressure and closing the thermodynamic cycle.

The discharging system is based on a conventional saturated ORC cycle, with internal recuperation. The PCM heat exchanger during discharge acts as working fluid evaporator, to achieve saturated vapor conditions before entering the ORC turbine. An internal recuperator is used to recover the residual heat at turbine discharge to preheat the working fluid at pump outlet, improving cycle efficiency and minimizing heat transfer irreversibility. A water-cooled condenser is adopted thanks to the large water availability, as typical of several geothermal power plants. Downstream the condenser, a centrifugal pump increases the working fluid pressure up to the maximum value, before the fluid is sent to the cold side inlet of the internal recuperator. It is thus clear that, among all the components of the ETES system, the PCM heat exchanger is the most critical and challenging piece of equipment to design, since it directly influences both charging and discharging cycles.

In this study, firstly a simplified approach for PCM heat exchanger sizing is adopted for the purpose of a preliminary working fluid selection. Secondly, the simulation of the ORC discharging cycle and the detailed design of the PCM heat exchanger are carried out considering the preselected working fluid and the set of boundary conditions defined for the Kızıldere II geothermal plant. The preliminary design of the heat exchanger is performed considering the requirements of the ORC discharging cycle, as for this specific geothermal application it features a larger heat duty compared to the HTHP charging cycle. The preliminary sizing of the PCM-working fluid heat exchanger is crucial to improve the accuracy of the simulation of the charging HTHP cycle, since the PCM heat exchanger operates in off-design conditions during the charging cycle. Finally, the HTHP charging cycle performance is evaluated based on the PCM heat exchanger sizing, and the overall ETES round-trip efficiency is computed.

ETES layout is represented in Figure 1.

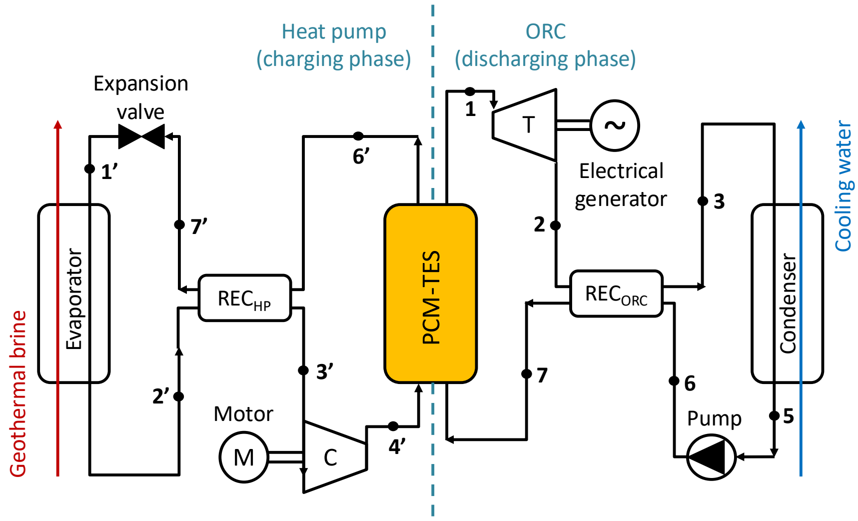

Figure 1: ETES concept layout. On the right, the heat pump cycle, working during low electricity prices and charging the PCM TES exploiting the electricity from the geothermal plant and the available heat from the geothermal brine before reinjection. On the left, the ORC unit, exploiting the thermal energy stored in the TES to produce additional power during high electricity demand periods.

2 THERMODYNAMIC MODELLING OF THE ETES

The assumption of this specific use-case is to size the ORC system for 2 hours discharging time, compared to 5 hours of HTHP charging time. Thus, due to the larger heat transfer duty associated to the discharging phase, the ORC power cycle is solved first, and the PCM heat exchanger is sized according to the discharging cycle operating conditions. The ORC cycle is designed to work with saturated vapor at turbine inlet to minimize heat transfer irreversibility and provide an optimal matching of the working fluid temperature profile with the PCM TES. For this reason, to attain a dry expansion process and avoid potential issues for the turbine operation, only working fluids with a sufficiently high molecular complexity and thus characterized by a dry expansion are investigated. Further details are specified in Section 2.4, dedicated to working fluid selection.

Once the ORC thermodynamic cycle is solved starting from the given input data and assumptions, the thermodynamic conditions and working fluid properties at the inlet and outlet of the ORC evaporator are defined. These values are then provided as input data to the PCM heat exchanger design routine.

The PCM heat exchanger is sized for a specified setpoint, based on a target minimum pinch-point temperature difference and a target pressure drop in nominal operating conditions.

The HTHP thermodynamic cycle is initially solved by considering a first guess pinch-point temperature difference of the PCM heat exchanger equal to the value assumed for the ORC operation. This pinch point temperature difference is then recursively updated, solving the HTHP power cycle at each iteration until convergency is attained considering the off-design operation of the PCM heat exchanger.

2.1 Modelling assumptions

ORC and HTHP power cycles are solved considering the following assumptions and boundary conditions:

- The geothermal brine is modelled as pure water and its inlet and outlet conditions are reported in Table 1, along with the cooling water conditions of the ORC condenser;
- Recuperative cycle configuration is considered for both ORC and HTHP cycles;
- Pinch-point temperature differences and target pressure drops for each heat exchanger are set equal to the values presented in Table 3;
- Fixed isentropic efficiency of turbomachines, mechanical and electrical efficiency are considered (ORC turbine and pump; HTHP compressor) adopting the reported values:

$$\eta_{is-TURB} = 0.85;\ \eta_{is-PUMP} = 0.75;\ \eta_{is-COMP} = 0.80;\ \eta_{mech} = 0.97;\ \eta_{el} = 0.95$$

- For both thermodynamic cycles, saturated liquid and saturated vapor are considered at condenser and evaporator outlet, respectively.

Table 1: Thermodynamic states of geothermal brine (waste heat source for HTHP evaporator) and cooling fluid (ORC condenser)

Fluid	T_{in} [°C]	T_{out} [°C]	p_{in} [kPa]	p_{out} [kPa]	$\dot{m}$ [kg/s]	Operating mode	Heat exchanger
Water	113.1	110.3	159	159	500	HP	EVA
Water	24	28	101.3	101.3	900	ORC	COND

PCM temperature is assumed to be constant and equal to 141°C for this preliminary study. Four possible candidates have been selected as PCM in this temperature range, namely Glucose, KNO_3–$NaNO_2$ (56%–44%), KNO_3-$NaNO_3$-$NaNO_2$ (HITEC), Dimethyl terephthalate (DMT). PCM final selection depends on corrosion and ageing performance, which is still under assessment. Melting temperature and latent heat of the PCM candidates are summarized in Table 2.

Table 2: PCM main thermal properties (melting temperature and latent heat)

PCM	Glucose	KNO_3–$NaNO_2$ (56%–44%)	HITEC	DMT
T_m [°C]	135/142	141	143	142
H_m [kJ/(kgK)]	49	97	117	165

The final ORC evaporation and HTHP condensation temperatures are then set considering the pinch-point temperature differences presented in Table 3.

Table 3: Main assumptions for the design of the heat exchangers of the ETES system.

Heat exchanger	PCM	REC (low pressure)	REC (high pressure)	COND
ΔT_{pp} [K]	5	8	8	5
Δp [kPa]	50	8	50	8

2.2 ETES discharging cycle assessment – Organic Rankine Cycle

Based on the assumptions reported in Section 2.1, the thermodynamic discharging cycle is solved for seven different working fluids, selected based on their critical temperature and environmental performance, namely their limited Global Warming Potential (GWP) and zero Ozone Depletion Potential (ODP). These fluids and their main thermodynamic and environmental properties are reported in Table 4. For each working fluid, the ORC power cycle is solved starting from the evaporator and condenser components, and for the given input data and assumptions for the PCM and cooling water, the temperatures and pressures of the working fluid are defined. Later, adopting the isentropic efficiency value reported in Section 2.1, the outlet conditions for the pump and the turbine are computed. Finally, by imposing the energy balance and the pinch point temperature difference at the recuperator, all the power cycle thermodynamic streams conditions are defined.
The PCM heat exchanger surface is preliminary estimated by dividing the component in two different subsections, namely the preheater and the evaporator. Using the mean logarithmic temperature differences and a preliminary guess of the heat transfer coefficients in each subsection, the overall PCM heat exchanger surface area is calculated as the sum of the two calculated surfaces.
The PCM heat transfer coefficient is considered equal to 1000 W/m^2K by adopting a lattice matrix used to enhance the material conduction to the heat exchanger tubes. The working fluid heat transfer coefficient is supposed to be equal to 1500 W/m^2K for single-phase liquid, 600 W/m^2K for single-phase vapor and 2000 W/m^2K for 2-phase conditions [5]. Among the working fluid candidates, n-pentane and cyclopentane show the best performance in terms of ORC efficiency, with cycle conversion efficiencies equal to 15.87% and 15.85%, respectively. The T-s (temperature-specific entropy) and p-h (pressure-specific enthalpy) diagrams of the power cycle working with cyclopentane are here reported in Figure 2.

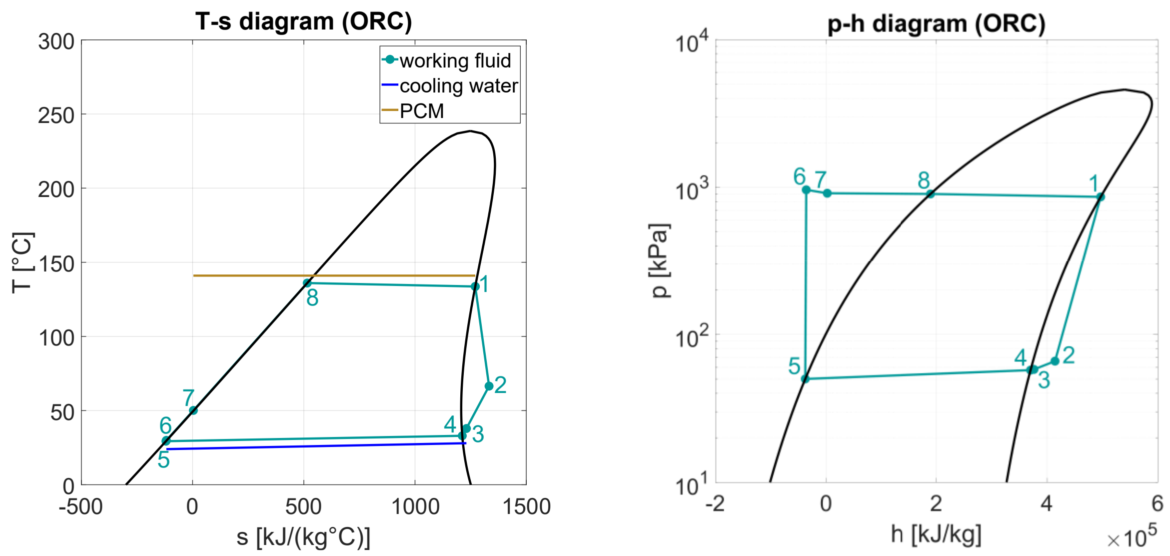

Figure 2: T-s (left) and p-h (right) diagrams of the power cycle working with cyclopentane.

2.3 ETES charging cycle assessment – High Temperature Heat Pump

Charging (HTHP) cycle is initially solved considering the same pinch-point temperature difference of the PCM heat exchanger assumed for the ORC operation (5 K).
PCM heat exchanger surface areas are thus calculated by dividing the heat exchanger in two main sections, namely de-superheater and condenser. The overall PCM heat exchanger surface area is calculated as the sum of these two heat transfer surfaces, following the same procedure adopted for the ORC power cycle. The PCM heat exchanger calculated surface is then compared to the area previously estimated for the ORC power cycle: the pinch point temperature difference assumed for HTHP operation is varied until convergence is reached between these two computed surfaces. The problem is handled in MATLAB adopting the *fsolve* routine and using the default algorithm to determine the solution. Once a feasible solution is found, the HTHP Coefficient of Performance (COP) is computed as the ratio between the thermal power discharged to the PCM TES and the compressor electrical power consumption.

2.4 Working fluid selection

Performance results in terms of ORC efficiency, HTHP COP, and overall RTE are summarized in Table 4 for each working fluid, along with the HP compression ratio and the main fluid parameters. The RTE is computed as the product between the ORC efficiency and the HTHP COP, in which a factor 0.95 is introduced to take into account for TES thermal losses. This simple preliminary approach to mutualize the PCM heat exchanger between the ORC and HTHP operating modes resulting in PCM heat exchange surface area matching between the two opposite modes, produces a first approximated ETES performance outcome that may be used for the working fluid selection. All fluid thermodynamic and transport properties are computed with CoolProp routines.

Table 4: Working fluid performance comparison in ORC mode

Working Fluid	η_{ORC} [%]	COP_{HTHP} [-]	RTE [%]	PR_{comp} [-]	GWP [-]	T_{cr} [°C]
Cyclopentane	15.85	6.83	97.70	2.37	5	236.6
Acetone	14.90	6.79	91.31	2.55	0.5	235.0
Cyclohexane	14.26	6.48	83.40	2.88	2	280.5
Isopentane	15.84	6.49	92.78	2.17	5	187.2
Isohexane	15.55	6.62	92.90	2.52	<1	224.6
N-pentane	15.87	6.61	94.67	2.25	<25	196.5
R1233zd(E)	14.80	6.07	81.08	2.17	1.37	165.6

The analysis highlights cyclopentane as the best option among the considered working fluid candidates as it properly balances the charging (HTHP COP) and discharging (ORC efficiency) efficiencies, leading to the maximum RTE among the pool of investigated fluids. Thus, cyclopentane is considered for the next step of the analysis, increasing the accuracy of the ETES performance assessment with a more detailed PCM heat exchanger design, which is presented in Section 3.

3 PHASE CHANGE MATERIAL HEAT EXCHANGER DESIGN

The PCM-working fluid heat exchanger is designed considering ORC operating mode, as it features a higher heat duty with respect to the HTHP operation. This heat exchanger is defined as a single pass with parallel tubes of a defined internal diameter, set to 23.3 mm for this study. Tube thickness is set to 3.6 mm, according to preferred dimensions suggested in EN 10216-2:2024. The design procedure adopts the mean logarithmic temperature approach splitting the heat exchanger in 600 sections with the same thermal duty, a trade-off between good accuracy and a reasonable computational time. Each heat exchanger segment is solved considering the calculated heat transfer coefficient at the inlet of each section. Convective heat transfer coefficients are calculated adopting Gnielinski correlation for Nusselt number calculation in single-phase flow regime [4], while for evaporating two-phase flow the Liu and Winterton correlation [6] for boiling flow inside tubes is adopted. Finally, for in-tube flow condensation heat transfer the procedure employed for heat transfer performance prediction is proposed by Cavallini et al. [7]. Pressure drop is determined for single-phase and two-phase sections considering the frictional contribution. A constant heat transfer coefficient of 1000 W/m^{2}K is assumed for the external heat transfer coefficient considering a lattice cell structure to improve the conductivity on the PCM side.

It is worth mentioning that the PCM properties assumed in the model, such as the heat transfer coefficient, have a quite high uncertainty. This uncertainty will be numerically quantified through detailed modelling and experimental validation of this crucial component. The full PCM heat exchanger design procedure is summarized in the simplified flow-chart of Figure 3.

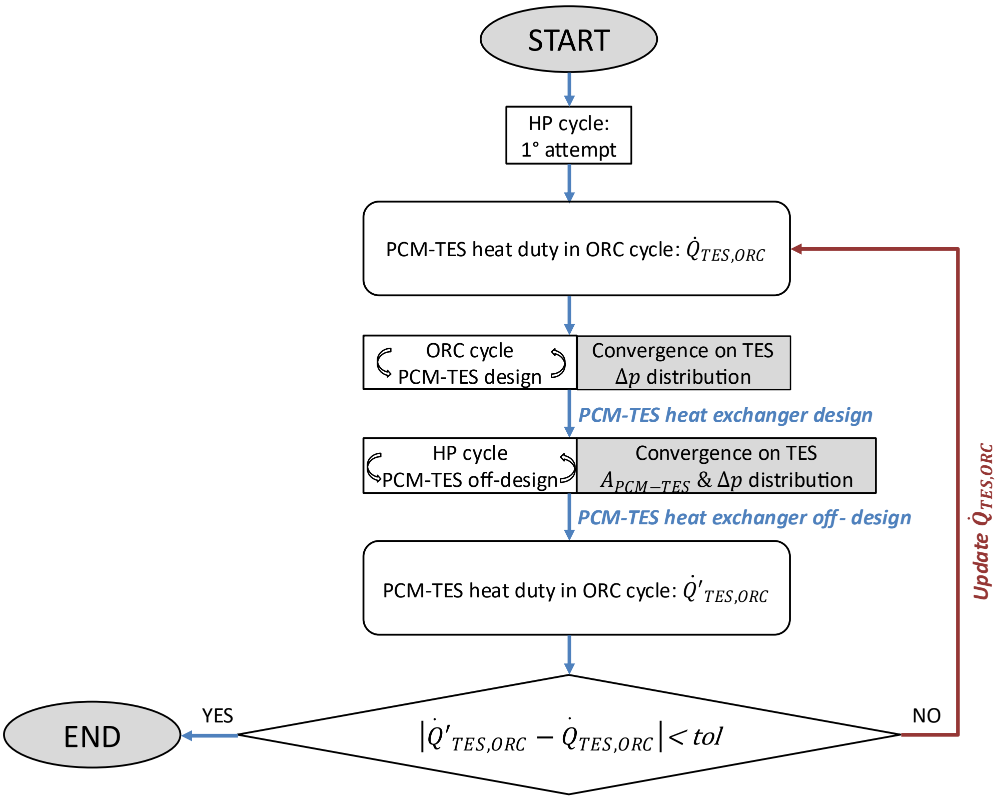

Figure 3: Simplified PCM heat exchanger design procedure and solving scheme of the HP and ORC cycle to ensure correct pressure drops estimation and heat transfer performance.

3.1 Heat exchanger design in ORC power mode

The design routine is carried out to obtain a target working fluid pressure drop across the heat exchanger, set to 50 kPa, and a target value of heat duty, defined in the ORC cycle thermodynamic analysis. The independent variables used to match these objectives are the number of heat exchanger parallel tubes and their length, considering a fixed tube diameter and thickness. Since the working fluid mass flow rate is univocally determined by the solution of the ORC thermodynamic cycle, setting the number of parallel tubes leads to a specific value of working fluid velocity, and consequently a well-defined result in terms of heat transfer coefficient and pressure drops per unit length. *Matlab lsqnonlin* embedded function is adopted to determine these two values, acting on the minimization of the residual functions. The PCM-working fluid heat exchanger results are here reported in Table 5 along with the representation of T-Q (temperature-thermal power) and U-Q (global heat transfer coefficient-thermal power) diagrams.

Table 5: Detailed PCM heat exchanger design

Total tube length [m]	Total internal tube surface area [m^2]	Number of parallel tubes [-]
30356	2222	316

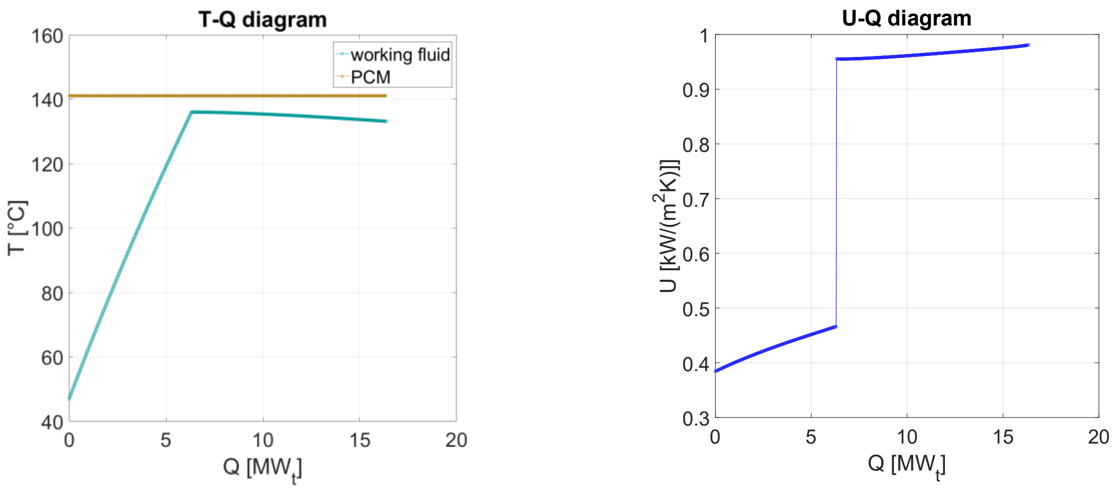

Figure 4: T-Q (left) and U-Q (right) diagrams of PCM Heat Exchanger in ORC mode (evaporator)

3.2 Heat exchanger off-design operation for HP mode

Once the PCM heat exchanger design is defined by the ORC power cycle, its off-design operation is considered to simulate the HTHP cycle and estimate its performance. In this case the target setpoint is the same length and number of tubes selected for the PCM heat exchanger design of the ORC power mode and the independent variable is the pinch-point temperature difference of the PCM heat exchanger working as a condenser in the HTHP operating mode. For the detailed modelling, the pressure drops of the working fluid in the condenser are iteratively updated by maintaining the same number of tubes in parallel computed during the ORC mode. The resulting pinch-point temperature difference which allows convergence is 4.34 K. T-Q and U-Q diagrams, showing the variation of Temperature and Global heat transfer coefficient along the heat exchanger longitudinal dimensions are here reported in the Figure 5.

T-s and p-h diagrams related to ORC and HTHP power cycles are reported in Figure 6. Operating parameters, efficiency indicators and power flows updated after PCM heat exchanger design at ORC side and off-design applied to HTHP charging cycle are reported in Table 6.

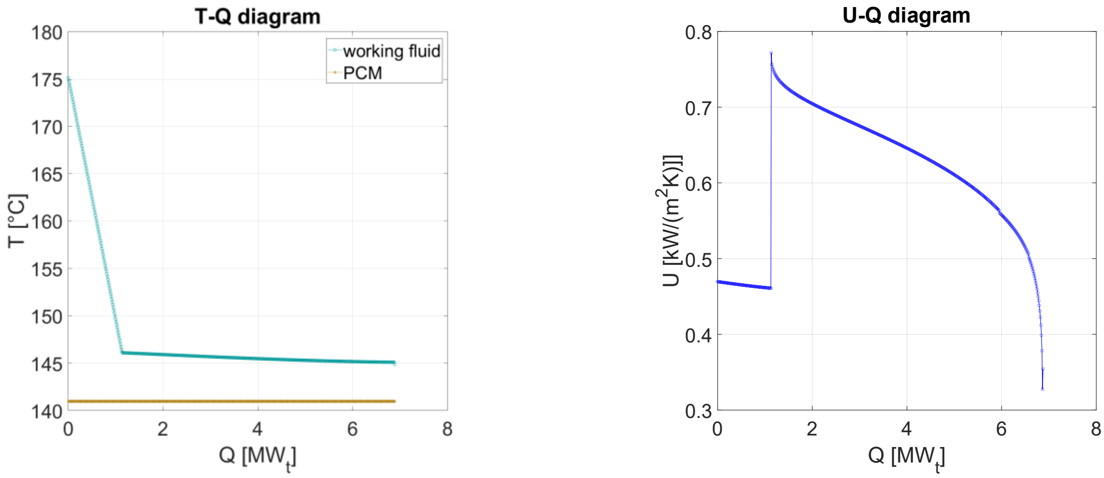

Figure 5: T-Q (left) and U-Q (right) diagrams of PCM Heat Exchanger in HTHP mode (condenser)

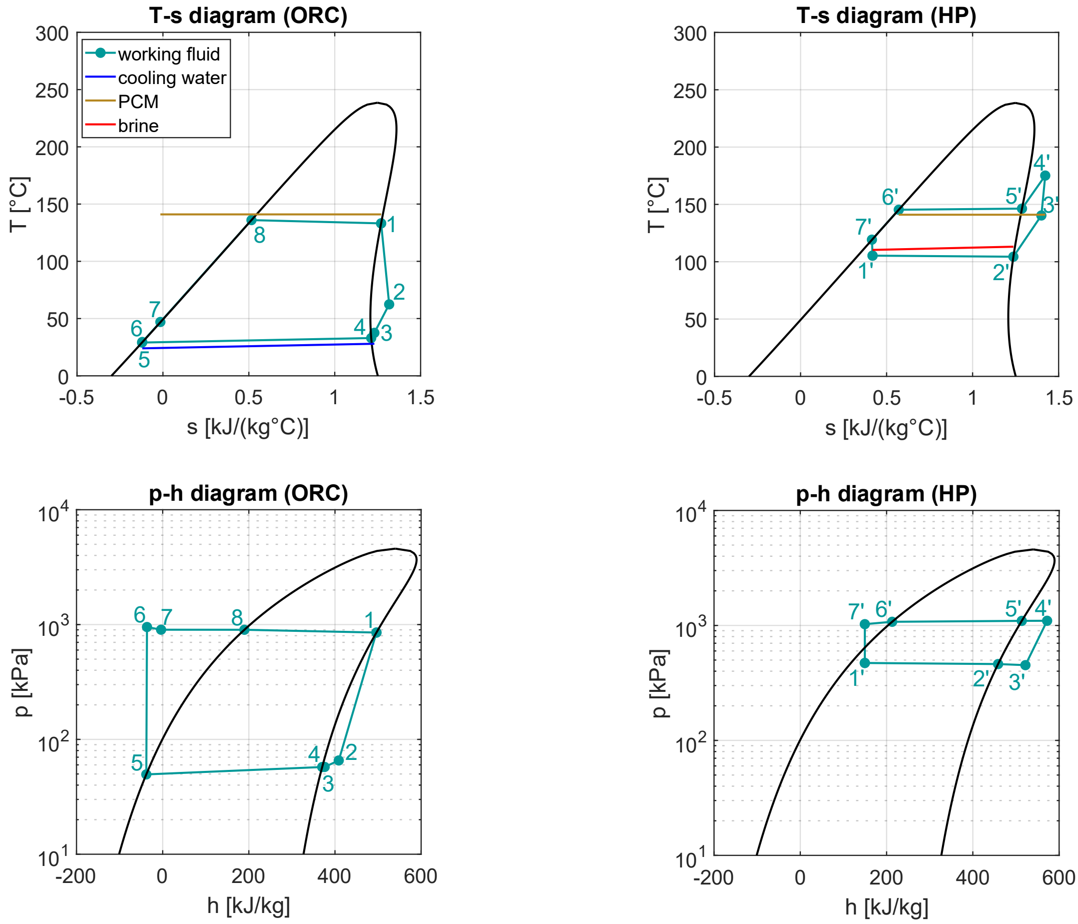

Figure 6: Optimized ORC and HTHP power cycles after PCM heat exchanger detailed design

Table 6: Final ETES performance assessment with detailed PCM heat exchanger ($t_{ch} = 5h;\ t_{dis} = 2h$)

η_{ORC} [%]	$P_{el_{ORC}} [MW_{el}]$	COP_{HTHP} [-]	$P_{el_{HTHP}} [MW_{el}]$	RTE [%]
15.82	2.59	6.58	1.05	94.01

4 CONCLUSIONS

The concept of electro-thermal energy storage represents a flexible and cutting-edge solution for both waste heat recovery and electricity storage within the EU energy system, as it can be possibly applied to several industrial and power generation sectors where process heat is currently wasted.

This study investigates the integration of ETES into the Kızıldere II geothermal power plant, one of the three use cases explored within the SEHRENE project, funded under the Horizon Europe research programme. In this context, ETES emerges as a promising energy storage solution for enhancing the flexibility of geothermal plants.

Preliminary results highlight the high potential of ETES technology to improve the dispatchability of Kızıldere II, positioning it as a competitive candidate in the ancillary services market. By enabling flexible operation, the ETES system contributes to making geothermal power investments more attractive from an economic standpoint, allowing traditionally base-load plants to participate in energy arbitrage strategies and improve long-term profitability.

The integration of PCM thermal energy storage coupled to saturated ORC cycles plays a key role in enabling this transition, supporting the shift from base-load coverage to dispatchable geothermal power generation. The initial assessment indicates a potential round-trip efficiency exceeding 90% for a thermally integrated solution, and an increase in the plant power output of up to 2.59 MW.

The ETES technology is a promising alternative solution to BESS, since it potentially achieves high round trip efficiency without critical raw materials, being at the same time a geographically independent solution. Future developments will include the detailed design of all ETES components, to be integrated into the simulation model and digital twin of the system as the SEHRENE project progresses. Further techno-economic assessment will be carried out to investigate the profitability of this technology, and LCA analysis will be performed to compare the environmental performance of this solution with conventional energy storage technologies, such as BESS. Simulation results will be validated using a laboratory-scale prototype, while further refinements will be made based on the actual performance of the PCM-TES component currently under development and testing.

NOMENCLATURE

Acronyms

BESS	Battery Energy Storage System
COP	Coefficient of Performance
CSP	Concentrated Solar Power
ETES	Electro-Thermal Energy Storage
EU	European Uninoon
GPP	Geothermal Power Plant
HTHP	High Temperature Heat Pump
KZD-II	Kizildere II geothermal power plant
LCA	Life Cycle Assessment
ORC	Organic Rankine Cycle
PCM	Phase Change Material
RTE	Round Trip Efficiency
SEHRENE	Store Electricity and Heat foR climatE Neutral Europe
TES	Thermal Energy Storage

Subscript

el	electric
t	thermal

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