

CARNOT BATTERY INTEGRATION IN A GEOTHERMAL POWER PLANT: STUDY CASE OF ZORLU ENERJI

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Abstract. Waste heat generated by the industrial sector is a major source of unused energy. Integrating technologies into this process improves energy efficiency and contributes to the reduction of greenhouse gas emissions. In this scope, the SEHRENE project presents a novel approach to industrial waste heat recovery, increasing process flexibility and efficiency. The concept integrates an Electro-Thermal Energy Storage (ETES) system, including a High-Temperature Heat Pump (HTHP), a Phase Change Material (PCM) storage and an Organic Rankine Cycle (ORC). This study evaluates the optimal integration of an ETES in the largest Geothermal Power Plant (GPP) in Türkiye. The 80 MWe Kızıldere-2 GPP belongs to the Zorlu Enerji group. This plant is composed of a triple flash-binary (ORC) combined GPP. Several ETES architectures and refrigerants are analysed, to select the optimal configuration. This paper presents the results of this study case including system performance and a sensitivity analysis of various system parameters. These were then chosen to get the best design out of the trade-off between the potential performance and industrial feasibility of the machine. The main findings indicate that cyclopentane is the optimal working fluid for the given operating temperatures, where with different system architectures the round-trip efficiency (RTE) can reach values higher than 80%. Also, the performance sensitivity of the varied parameters was assessed. The most impactful varied parameters were the PCM melting temperature and the turbine isentropic efficiency. These results show the promising integration of ETES in geothermal power plants and all other processes with usable low and medium-temperature waste heat.

Keywords. High Temperature Heat Pump, Organic Rankine Cycle, Thermal Integration, Geothermal energy, ETES

Nomenclature

A	Heat transfer area (m ²)
$\dot{C}$	Heat capacity rate (W/K)
E	Energy (J)
F	LMTD correction factor (-)
h	Specific enthalpy (J/kg), Heat Transfer Coefficient (W/m ² ·K)
$\dot{m}$	Mass flow rate (kg/s)
$\dot{Q}$	Heat duty (kW)
t	Time (h)
T	Temperature (°C or K)
U	Overall heat transfer coefficient (W/m ² ·K)
v	Specific volume (m ³ /kg)
$\dot{V}$	Volumetric flow rate (m ³ /s)

Special characters

η	Efficiency (-)
Π_r	Pressure ratio (-)
Δ	Difference (-)
Acronyms	
COP	Coefficient Of Performance
ETES	Electro-Thermal Energy Storage
GWP	Global Warming Potential
(HT)HP	(High Temperature) Heat Pump
IWH	Industrial Waste Heat
KPI	Key Performance Indicators
LMTD	Log Mean Temperature Difference (K)
ORC	Organic Rankine Cycle
PCM	Phase Change Material
RTE	Round-Trip Efficiency
VHC	Volumetric Heat Capacity (J/K·m ³)

Subscripts

cd	Condenser
cp	Compressor
ev	Evaporator
ex	Exhaust
HX	Heat Exchanger
pp	Pump, Pinch point
s	Isentropic
sf	Secondary fluid
su	Supply
sto	Storage
tu	Turbine
wf	Working fluid
x1	Vapor quality of 1

1 Introduction

Research on Electro-Thermal Energy Storage (ETES) has gained increasing interest because of the recent need for electricity storage to damp renewable energies' intermittent production [1]. This specific storage solution depends less on geographical conditions [2] and on rare earth elements [3] and thus could be a suitable long-term solution. However, the complexity of these machines and the recency of the research interest are barriers to overcome. The SERHENE project aims to contribute to this field by presenting and deeply assessing novel thermally integrated designs. These designs contain High Temperature Heat Pumps (HTHP), Organic Rankine Cycles (ORC) and novel Phase Change Material (PCM) heat storages. They are also studied as to be integrated into real test cases, the one focused on here is the largest Geothermal Power Plant (GPP) in Türkiye from Zorlu Enerji. The Kızıldere-2 (KZD II) GPP has an installed capacity of 80 MWe and is composed of a triple flash-binary (ORC) combined GPP. This paper will be led as the following:

- Introduction of the case study and of the reference topology.
- Presentation of the thermodynamic model.
- Selection of the refrigerant and the topology.
- For the final topology, refinement of the parameters thanks to a sensitivity study.
- Pre-sizing of the components of the system.

2 Reference Case Definition

2.1 Case study description and reference topology

The reference Carnot battery design is depicted on Figure 1. Its HTHP contains an internal heat exchanger and its ORC includes a recuperator.

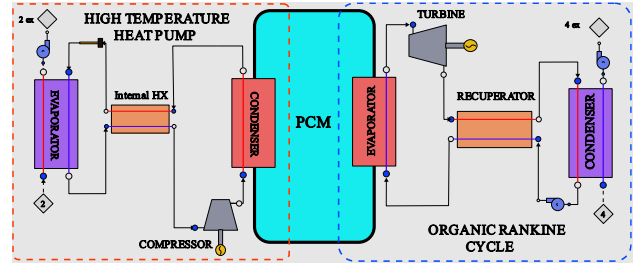


Figure 1: ETES reference configuration.

According to the information provided by Zorlu Enerji regarding the Kızıldere II three-flash plant with a Binary cycle, various energetic vectors can be recovered or utilized for potential integration into the ETES system. Table 1 presents the properties of the different vectors that could be integrated into the ETES. Based on this data, the low-pressure separator brine point will be considered as the Industrial Waste Heat (IWH) for the HTHP evaporator. Using the fluid coming from KZD II wells would have been an option but it also would have decreased the energy produced by the current system. Meanwhile, the cooling water supply to the binary cycle will be used as the cold source for the ORC condenser.

Table 1: Available residual heat sources from KZD-II GPP.

Point	T [°C]	P [bar]	$\dot{m}$ [ton/h]	Fluid
From KZD II wells (1)	178.1	10.65	2678.5	97% steam/brine
Low-pressure separator brine (2)	113.1	1.59	3088.6	100% steam/brine
Condensate from binary Plant (3)	50.6	3	283.9	99.9% steam/brine
Cooling water supply to the binary plant (4)	24.1	3.63	9000	100% steam/brine

2.2 Thermodynamic Modelling

Various components are employed and modelled, in this first stage as a thermodynamic model based on different assumptions. These hypotheses are represented for each component involved in the possible configuration of each case study. The modelling of individual components is conducted using EES 2024 (Engineering Equation Solver).

2.2.1 Heat exchangers with phase change (Evaporator and Condenser): In these components the heat transfer process involves two fluids or mediums. In the evaporator, the refrigerant or working fluid (wf) undergoes a phase change from liquid (or two-phase) to saturated, or even superheated vapour, or even to superheated vapour. Conversely, in the condenser, the refrigerant transitions from vapour (or two-phase) to liquid

through condensation. The energy balance between the refrigerant and the secondary fluid (sf) is presented in Equation 1. This relation corresponds to the condenser, nevertheless, for the evaporator it is the same in the reverse way.

$$\begin{aligned} \dot{m}_{wf,cd} \cdot (h_{wf,ex,cd} - h_{wf,su,cd}) &= \dot{Q}_{cd}, \\ \dot{Q}_{cd} &= \dot{m}_{sf,cd} \cdot (h_{sf,su,cd} - h_{sf,ex,cd}) \end{aligned} \quad (1)$$

The saturation temperature is computed as the mean temperature between the one at the beginning and the one at the end of the two-phase zone. The equations of the pinch-point (*PPTD*) for the HP condenser are presented in Equations 2 and 3. These equations allow to determine by iteration of the pinch-point in the heat exchanger and set to determine the saturation state.

$$\dot{Q}_{pp,cd} = \dot{m}_{sf} \cdot (h_{sf,pp,cd} - h_{sf,su,cd}), \quad (2)$$

$$\dot{Q}_{pp,cd} = \dot{m}_{wf} \cdot (h_{wf,x1,cd} - h_{wf,ex,cd})$$

$$\Delta T_{pp,cd} = T_{wf,x1,cd} - T_{sf,pp,cd} \quad (3)$$

2.2.2 Sensible heat exchangers (recuperator, preheater and superheater): These heat exchanger models only transfer sensible heat between two fluids. The recuperator transfers heat from the turbine exhaust to the pump exhaust (ORC), and after the condenser with the exhaust of the evaporator (HTHP). The preheater preheats the fluid upstream of the ORC evaporator. The superheater serves to increase the gaseous temperature at the exhaust of the ORC evaporator. The assumption for modelling these components corresponds to considering an isobaric process, then no pressure drops. A counterflow heat transfer process is considered. Finally, energy and mass are conserved in both fluids. The same energy balances are considered as in the phase change heat exchanger, adding the heat transfer equation, which is presented in Equation (4).

$$\dot{Q}_{hx} = \dot{C}_{min,hx} \cdot \eta_{HX} \cdot (T_{hot,su,HX} - T_{cold,su,HX}) \quad (4)$$

2.2.3 Compressor: This component is employed in the HTHP to raise the pressure from the evaporator to the condenser. To avoid any kind of problem, superheating of the compressor suction due to refrigerant droplets must be ensured. The isentropic and volumetric efficiencies are presented in Table 2. The first is used to compute the machine real work. The second is used to compute the swept volume of the volumetric machines. The energy balance for computing the input energy for the compressor is

presented in Equation 5, whereas the calculation of the swept volume is presented in Equation 6.

$$\dot{W}_{cp} = \dot{m}_{wf} \cdot (h_{wf,s,ex,cp} - h_{wf,su,cp}) / \eta_{s,cp} \quad (5)$$

$$V_{cp} = \frac{\dot{m}_{wf}}{\eta_{v,cp} \cdot N_{cp} \cdot \rho_{wf}} \quad (6)$$

2.2.4 Turbine: This component is the machine used in the ORC to convert the fluid internal energy into electricity. This component and model can be considered identical to the compressor but with the energy transfer taking place in the opposite direction. The energy balance to compute the energy taken from fluid in the turbine corresponds to Equation 6.

$$\dot{W}_{tu} = \dot{m}_{wf} \cdot \eta_{s,tu} \cdot (h_{wf,su,tu} - h_{wf,s,ex,tu}) \quad (7)$$

2.2.5 Pump: The pump is the drive component in the ORC, which increases the pressure in the working fluid from the lower (condenser) to the upper pressure (evaporator). This component also consumes energy and to avoid any cavitation problems the working fluid is subcooled. The assumption for modelling these components corresponds to an isentropic process driving the fluid from the supply to the exhaust, where an efficiency of 65% was considered. The power required for the pump is computed according to Equation 7.

$$\dot{W}_{pp} = \dot{m}_{wf} \cdot \left(v_{wf,su,pp} \cdot \frac{(P_{wf,ex,pp} - P_{wf,su,pp})}{\eta_{s,pp}} \right) \quad (8)$$

2.2.6 Phase Change Storage: PCM heat storage is the key component for the ETES which stores the energy from the HTHP and releases it in the ORC for producing electricity. The material has not been yet defined. The project's Key Performance Indicators (KPIs) require a PCM storage with an energy density corresponding to 90 kWh/m^3 . It is assumed that PCM can release a constant energy flow rate corresponding to the total energy storage divided by the discharging time. The energy storage in the PCM is computed according to Equation 8.

$$E_{sto,PCM} = \int_{t_{initial}}^{t_{final}} \dot{Q}_{cd,HP} dt \quad (9)$$

2.2.7 Expansion valve: This component considers an isenthalpic process which connects the HTHP condenser with the HTHP evaporator. It is considered that this component can maintain a constant evaporator superheating.

Table 2 summarises the pinch point and efficiencies mentioned in the equations 3 to 8.

Table 2: Assumed parameters for the reference case

Parameter	Value	Unit
$\Delta T_{pp,cd}, \Delta T_{pp,ev}$	3	K
η_{HX}	0.8	-
$\eta_{s,cp}, \eta_{s,tu}, \eta_{s,pp}$	0.65	-
$\eta_{v,cp}, \eta_{v,pp}$	0.95	-
T_{PCM}	160	°C
t_{charge}	4	h
$t_{discharge}$	2	h
V_{PCM}	322.22	m^3

2.3 Refrigerant Selection

This section focuses on selecting refrigerants suitable for use in the ETES system. A PCM temperature of 160°C is considered in this evaluation. For HTHPs, refrigerants such as R1233zd(E), R1234(Z), R600, R601, R1336mzz(Z) and R1224yd(Z) are considered promising candidates for this application [4]. Additionally, R1234ze(Z) and R1234ze(E) are highlighted as viable options for high-temperature applications [5, 6], along with hydrocarbon-based refrigerants [7].

A comprehensive study by Frate et al. [8] has assessed 27 potential refrigerants with a Global Warming Potential (GWP) lower than 1000, identifying options that achieve critical temperatures above 125°C and maintain a saturation pressure above 0.05 bar at 40°C. Furthermore, zeotropic refrigerant mixtures have been thermodynamically evaluated, demonstrating performance improvements of up to 46% [9].

The working fluids selected for potential integration in the ETES system include Acetone, Benzene, Cyclopentane, Cyclohexane, Ethanol, Isopentane, Isohexane, Methanol, n-pentane, R1233zd(E), R1234ze(Z), R245fa, R600, and Toluene. These fluids are evaluated based on two key performance metrics for HTHP: the Coefficient of Performance (COP) and the Volumetric Heat Capacity (VHC) [9]. In ORC, the fluids are assessed based on the cycle efficiency (η_{orc}). Additionally, for ETES, the Round-Trip Efficiency (RTE) is also considered, calculated as the product of the HTHP COP and the ORC cycle efficiency.

At first glance, refrigerants like R1234ze(Z), R245fa, and R600 are not suitable for operation at temperatures above 150°C due to limitations imposed by their critical pressures. Similarly, R1233zd(E) has a critical pressure near this

threshold, which can cause operational challenges when PCM temperatures exceed 160°C. Additionally, fluids such as ethanol, methanol, cyclohexane, and toluene exhibit pressures below 0.1 bar at 20°C, which extremely increases the risk of air leakage into the ORC condenser. These refrigerants are assessed in the configuration presented in Figure 1, which integrates the reference configuration for the HTHP [4] and ORC.

The main simulation results are summarized in Table 3, where refrigerants are evaluated at a HTHP heat capacity of 7 MW_{th}, and times of charge and discharge of 4 and 2 hours, respectively. The compressor and turbine are considered at a fixed speed of 50 Hz.

Table 3: Results from the simulation of the reference configuration of the pre-selected refrigerants

Fluid	COP	η_{orc}	RTE	VHC
Acetone	4.27	0.156	0.667	3.625
Benzene	4.47	0.165	0.739	1.92
Cyclopentane	4.28	0.163	0.699	3.57
IsoPentane	3.83	0.156	0.597	4.81
IsoHexane	4.173	0.163	0.682	2.63
n-pentane	3.93	0.158	0.620	4.29

Among the fluids tested, benzene achieves the highest RTE, exceeding 70%. However, benzene's low VHC necessitates larger components, resulting in higher investment costs compared to isopentane and n-pentane, which exhibit VHC values above 4. Acetone and cyclopentane emerge as optimal fluids, offering a balanced trade-off between RTE and VHC. Notably, cyclopentane achieves a 3.2% higher RTE and a 1.5% lower VHC compared to acetone. In conclusion, with the considered PCM temperature and the evaluated parameters, cyclopentane is determined to be the most suitable choice for the ETES system.

2.4 Architecture Selection

Since the HTHP and the ORC will never work at the same time, it is possible to imagine architectures where the IWH is used during the ETES discharge. The different possible configurations analysed correspond to:

- HP/ORC: Reference case (Figure 1).
- HP/ORC + PRE: Reference case with integration of preheater fed with IWH.
- HP/ORC + SUP: Reference case with integration of IWH to feed the evaporator and where the PCM HX is a superheater.

- HP/ORC + 2 stages: Reference case with integration of a 2-stages ORC. The high-pressure turbine is supplied by the PCM HX and the low-pressure turbine by IWH.
- ORC standalone: Only ORC, directly in contact with the IWH.

For the architecture selection, a PCM temperature of 150°C is considered while all the other parameters remain the same as in Table 2. The results of the different assessed configurations are presented in Table 4. The better RTE corresponds to configuration 1, which can produce 2.168 MW_e during the discharging time. Another interesting option is configuration 3, which has lower RTE performance and higher electricity output. An important remark on this configuration corresponds to the required size of the ORC is 18 and 14 times higher than cases 1 and 2, respectively.

Table 4: Simulation results architecture selection.

Variables	HP/ ORC	HP/ORC + PRE	HP/ORC + SUP	HP/ORC + 2stages	ORC stand- alone
η_{orc}	15.9	15.7	10.9	12.5	12.2
COP	5.5	5.5	5.5	5.5	-
RTE	0.87	0.86	0.60	0.68	-
$\dot{W}_{cp}$	1.23	1.23	1.23	1.23	-
[MW]					
$\dot{W}_{tu}$	2.17	2.62	8.36	2.14	2
[MW]					

The best trade-off between the RTE and output power corresponds to configuration HP/ORC + PRE, which has a RTE lower of 1.15%, but the power output increases by 20.8%. For this reason, configuration HP/ORC + PRE is highlighted as the better configuration with the given pre-design conditions. In configuration HP/ORC + SUP, the ORC efficiency is the lowest due to the fact that the ORC working fluid mass flow rate is very high in order to satisfy the discharge time as the specific heat transfer is lower. This high mass flow rate results in a drop of secondary fluid temperature in the evaporator, lowering the saturation temperature. This decreases the pressure ratio, and thus the efficiency of the ORC. The representation of the chosen architecture is presented in Figure 2. This is the selected pre-design for the Zorlu use case, this will be further analysed in the parametric section. The idea is to get the optimal values for this configuration.

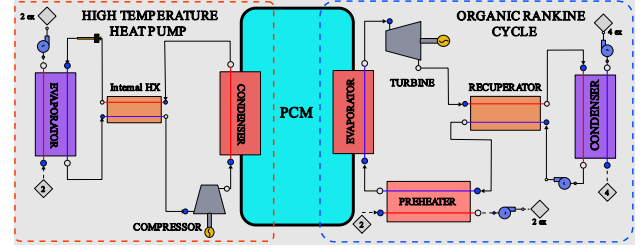


Figure 2: Zorlu configuration HP-ORC(Preheater)

3 Reference Case Refinement

Different constraints shall be added to give a more realistic view of the ETES performance potential. These constraints and the machine parameters shall then be varied to determine which ones are the most performance sensitive and then chosen/optimized in consequence. As shown in Figure 2, the heat pump and the ORC can be studied separately except for the integrated PCM heat exchanger. Note that the impact of each parameter variation was evaluated by only considering one variation at a time.

3.1 Constraints Addition

Many constraints are to be added, which are:

- Constraints on heat exchanger pinch point temperature differences (PPTD).
- Heat exchanger pressure drops.
- Secondary fluids and their auxiliaries.

First, varying **heat exchanger PPTD** has an impact on their working fluid saturation temperatures. This in turn impacts the saturation pressures, leading to higher/lower pressure ratios of the machine. Increasing PPTDs of HP heat exchangers increases its pressure ratio for a given condenser power, leading to lower COP. Increasing PPTDs of ORC heat exchangers decreases its pressure ratio for a given evaporator power, leading to lower efficiency. For example, the novel PCM-integrated heat exchanger still has to undergo experiments to determine its performance, a PPTD of 10K is therefore chosen as a first safety value.

Secondly, **heat exchanger pressure drops** lower the performance of Carnot Batteries. However, some are more important than others. For the heat pump, pressure drops in the high-pressure part show little impact on its performance as the valve does not recover any work. However, the low-pressure side pressure drops tend to increase the heat pump pressure ratio, leading to higher compressor work lowering the COP. The conclusions are similar for the ORC. The pressure drops in the low-pressure side lead to a higher pressure at the outlet of the turbine,

as the condensing pressure is constrained to be higher than a threshold value of 0.5 bar to prevent leakages. This decreases the pressure ratio of the machine, leading to lower work production and lower ORC efficiency. The high-pressure HX pressure drops still remove useful work but this impact is lower compared to the phenomenon explained above. Depending on their importance, the allowable pressure drops assumed in heat exchangers are listed in Table 5.

Table 5: HX assumed pressure drops [kPa].

Importance	Liquid	Gas	Evaporating	Condensing
Low	50	20	50	50
High	30	10	15	15

Finally, **secondary fluid auxiliary consumptions** need to be considered since they decrease the ETES performance as they impact the heat pump COP and the ORC efficiency as follows:

$$\eta_{ORC} = \frac{\dot{W}_{exp,orc} - \dot{W}_{pp,orc} - \dot{W}_{pp,sf}}{\dot{Q}_{ev,orc}} \quad (10)$$

$$COP_{HP} = \frac{\dot{Q}_{cd,HP}}{\dot{W}_{cp,HP} + \dot{W}_{pp,sf}} \quad (11)$$

However, this negative impact can be reduced through an optimization of the secondary fluid flowrate. The optimization can be determined in this case through the tradeoff between pump consumption and the efficiency of the heat exchanger.

3.2 Heat pump parameter choice

Considering the HP, the first parameters to set are the **allowable heat exchanger pressure drops**. These are set according to values in Table 4 and the discussion above it.

Another parameter to vary is the **PCM heat exchanger PPTD**. From discussions with manufacturers, it was decided to fix it first to a value of 10K instead of the 3K one that was first considered. However, note that this specific heat exchanger shall be shared between the two machines. This implies that the areas computed in Section 4.1 for these two virtually independent heat exchangers (heat pump condenser and ORC evaporator) shall be similar. The biggest area is then selected because it is the most restrictive and the other PPTD is then adjusted to fit a similar area value. The PPTD for the heat pump condenser is thus adjusted from 10 K to 5 K.

In addition, the **heat pump secondary fluid at the evaporator** has been considered. From a sensitivity study on the COP with respect to its flowrate, a value of 500 kg/s (previously 858 kg/s) was determined as optimal and set.

Finally, the last parameter to consider is the **PCM temperature**. Lowering the PCM temperature allows the heat pump COP to increase but the ORC efficiency to decrease. In this case, the impact on the HP COP is dominant on the global system performance. Decreasing the PCM temperature increases the RTE. It was decided to vary it from 150 (in section 2.4) to 141°C. This is the lowest fusion temperature among the PCM selected for this study.

Figure 3 shows the effect of the parameter variations on the heat pump COP. It can be concluded that the PCM melting temperature change has by far the most impact on performance. This is to be expected as it has a direct impact on the pressure ratio, leading to a change of compressor work and COP. The second most important factor is the increase of the PPTD at the integrated-PCM heat exchanger as it counteracts the lowering of the PCM temperature. Thirdly, the secondary fluid pump work shows a non-negligible impact on performance as it englobes the flowrate and the pressure drop effects. Finally, as already mentioned in Section 3.1, the impact of heat exchanger pressure drops highly depends on the working fluid and the pressure level that the heat exchanger is working on. The low-pressure side is the most sensitive.

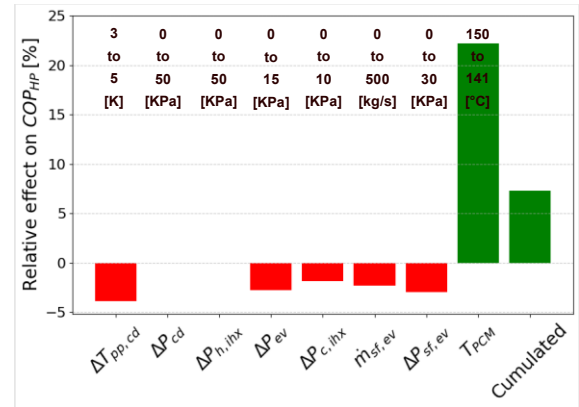


Figure 3: Relative effect of parameter variation on Heat Pump Performance.

In conclusion, the change in parameters had a positive cumulated impact on heat pump performance thanks to the lowering of the PCM temperature. The COP of the HP improved by 0.4 (+7.3% from the previous value).

3.3 ORC parameter choice

For the ORC, the same hypotheses as for the heat pump were considered for **allowable heat exchanger pressure drops**.

Concerning the **PCM heat exchanger PPTD**, the value of 10K mentioned by manufacturers was considered.

In addition, there are two secondary fluid related heat exchangers for this ORC architecture. For the sake of completeness, both of them shall be considered. The **preheater secondary fluid** flowrate and related pressure drop have been respectively to 60 kg/s and 50 kPa. Concerning the **condenser secondary fluid**, it was chosen to set its flowrate to 900 kg/s and its pressure drop to 30 kPa. These values result from a sensitivity study on ORC performance.

The **PCM melting temperature** shall be identical to the heat pump's one. The positive impact on the heat pump COP is higher than the negative one on the ORC efficiency when lowering the PCM temperature. Even though the ORC efficiency will suffer from it, the PCM temperature is set to 141°C.

Other parameters have to be considered specifically for the ORC. First, the **ORC condenser PPTD** shall be varied to maintain the pressure before the pump above 50 kPa. Its value was set to 5K. Secondly, as active charge management is meant to be studied in the project, the **ORC condenser subcooling** was changed from 3K to 1K to get closer to a system with a liquid receiver. Finally, the ORC features a 2MWe turbomachine. The value of 65% for its isentropic efficiency seems underestimated. It was then decided to fix it to 80%, which is closer to what can be seen in the industry while not being too optimistic.

Figure 4 shows the effect of the parameter variations on the ORC efficiency. It can be concluded that the most impactful change was to increase the turbine isentropic efficiency as it increases the net generated work significantly for the same turbine inlet conditions. In the ORC case, the second most impactful varied parameter type is the refrigerant heat exchanger pressure drops (again the low-pressure side ones) as they impact directly the pressure ratio of the turbine, reducing its output work. The PCM temperatures and heat exchanger PPTD had similar impacts on the ORC performance. These parameters reduce the cycle temperature difference,

lowering the pressure ratio and thus, the efficiency. The difference in relative importance is mainly due to the change in Kelvins. Finally, the impact of secondary fluids is lower in the case of the ORC (especially for the preheater since the flowrate is very low).

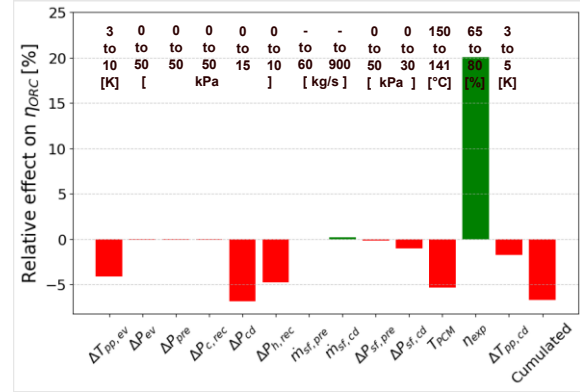


Figure 4: Relative effect of parameter variation on ORC Performance.

In conclusion, the change in parameters had a negative cumulated impact on ORC performance because of low-pressure side heat exchanger pressure drop and the lowering of PCM melting temperature. The ORC efficiency dropped by 1.04 [%] (-6.72% relative to the previous value).

3.4 Global system performance

To sum up, many sources of losses were added to the ETES to refine the study case. Their effects were first analysed through a sensitivity analysis together with some previously set parameters. Then, values were set (or re-set) for all these parameters. Most added parameters have a negative impact on ETES performance as they are sources of irreversibility that were not considered previously. To keep a similar RTE, the PCM temperature was decreased, and the turbine isentropic efficiency was increased. The performance results after this study are shown in Table 6. Figure 5 shows the ETES architecture with all fluid thermodynamic states.

Table 6: Results for **HP/ORC + PRE** architecture before (B) and after (A) study case refinement.

	η_{orc} [%]	COF [-]	RTE [%]	$\dot{W}_{cp}$ [MW]	$\dot{W}_{net}$ [MW]	T_{PCM} [°C]
B	15.7	5.5	86	1.23	2.62	150
A	14.4	5.9	85	1.19	2.51	141

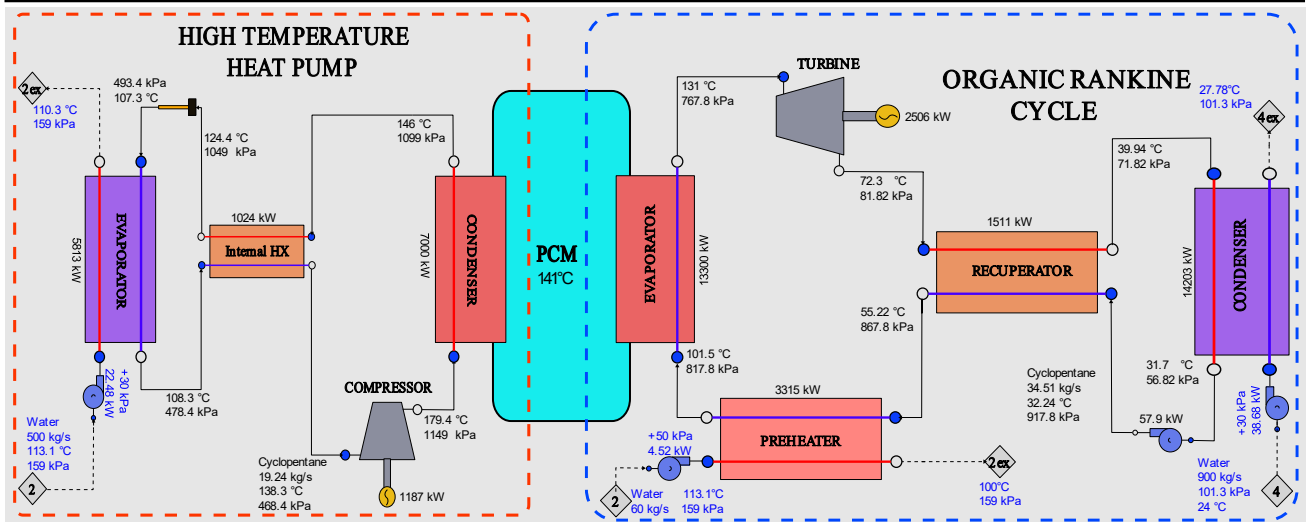


Figure 5: Chosen ETES architecture with thermodynamic states after parametric analysis.

4 Pre-Size of Components

This section aims to provide a pre-design of the main ETES components.

- For the heat exchangers: The technology and the heat transfer area. These are the pre-size parameters.
- For the turbomachines: The work, the inlet volumetric flowrate and the pressure ratio. These are the pre-size requirements.

4.1 Heat Exchangers

The heat exchanger area is pre-sized using the LMTD (Logarithmic Mean Temperature Difference) method. This method is commonly used for design and is based on temperatures at the inlet and outlet ports of a heat exchanger. It assumes that the heat duty ($\dot{Q}$) can be linked to the LMTD as follows:

$$\dot{Q} = U_{HX} \cdot A \cdot LMTD \cdot F \quad (11)$$

$$U_{HX} = \frac{1}{1/h_{cold} + 1/h_{hot}} \quad (12)$$

Where A [m²] is the heat exchanger area (to be determined), U_{HX} [W/(m² × K)] is the global heat transfer coefficient and F is the LMTD correction factor depending on the heat exchanger technology/configuration (computed using EES pre-built dedicated functions). U_{HX} is computed by considering the convective effects of both fluid sizes, meaning that the conduction resistance in the heat exchanger wall is neglected.

The heat exchanger technologies can be chosen based on their inlet fluid states and their heat duty. Due to the presence of mainly liquid secondary fluids, all

heat exchangers (except the ones with PCM) were chosen to be shell-and-tube heat exchangers. The internal heat exchanger and the recuperator could feature finned tubes.

From Kakaç et. al. [11], generic heat transfer coefficient values were assumed for cyclopentane (considered between a light and a medium organic fluid) and water in shell-and-tube heat exchangers. Their values are summed up in Table 7.

Table 7: Typical Film Heat Transfer Coefficients for Shell-and-Tube Heat Exchangers.

Heat Transfer Mode	Fluid or Material	U $\left[\frac{W}{m^2 \times K} \right]$
Sensible Liquid	Water	5000
Latent solidification	PCM	1000
Latent Fusion	PCM	1000
Sensible Liquid	Cyclopentane	1500
Sensible Gas	Cyclopentane	600
Latent Evaporation	Cyclopentane	2000
Latent Condensation	Cyclopentane	2000

For the specific case of PCM-related heat exchanger, an equivalent heat transfer coefficient ($U_{eq,PCM}$) guess proved to be challenging. Because of the discussion on the pinch point value for one mode or the other in Section 3.2, the impact of the choice of this value on the areas of both modes had to be investigated. A sensitivity analysis on the ratio of required area for both modes (HTHP condenser and ORC evaporator) with respect to this parameter was performed. The results show that the value of $U_{eq,PCM}$ does not impact the area ratio significantly. Indeed, when varying the parameter from 500 to 5000 W/(m² × K), the required area of the heat pump condenser remained between 55 and 60% of the area

of the ORC evaporator. So, the effect on the pinch point temperature difference when trying to fit the two areas shall not depend significantly on this assumption. From a short literature review [12-14], the value for $U_{eq,PCM}$ for both solidification and melting processes were assumed at $1000 \text{ W}/(\text{m}^2 \times \text{K})$. This consists in a first approximation for the design point when lacking experimental data. Table 8 shows the results of heat exchanger pre-size from these discussions.

From this table, it can be concluded that except for the PCM-related heat exchanger, the biggest heat exchanger is the HTHP evaporator. Because of the small pinch point temperature difference imposed for this heat exchanger, its heat flux ($= 5.27 \text{ kW}/\text{m}^2$) is much smaller than the one of the ORC condenser ($= 13.14 \text{ kW}/\text{m}^2$). This means that this heat exchanger is the first one on which the pinch point temperature difference value could be increased to reduce the total heat exchanger area of the machine. However, its size is quite close to the one of the ORC condenser. Another solution to reduce heat exchanger costs might be to mutualize these heat exchangers.

Table 8: Heat Exchangers pre-size results.

Heat Exchanger	$\dot{Q}$ [kW]	Hot inlet fluid	Cold inlet fluid	HX Technology	A [m ²]
HTHP Evaporator	5813	Liquid (Shell)	Liquid (Tube)	Shell & Tube	1103
HTHP IHX	1024	Gas (Shell)	Liquid (Tube)	Shell & Tube	262.6
HTHP Condenser	7000	Liquid (Tube)	PCM	Kelvin Cell	1525
ORC Evaporator	13300	PCM	Liquid (Tube)	Kelvin Cell	1618
ORC Preheater	3315	Liquid (Shell)	Liquid (Tube)	Shell & Tube	147.9
ORC Recuperator	1511	Gas (Shell)	Liquid (Tube)	Shell & Tube	374.1
ORC Condenser	14203	Liquid (Tube)	Liquid (Shell)	Shell & Tube	1081

4.2 Turbomachines

Turbomachines are usually pre-sized according to their work ($\dot{W}$), inlet volumetric flowrate ($\dot{V}$) and pressure ratio (Π_r). These can be deduced from the data in Figure 5. They are listed in Table 9.

Table 9: Turbomachines pre-size requirements.

Turbomachine	$\dot{W}$ [kWe]	$\dot{V}$ [m ³ /s]	Π_r [-]
HTHP Compressor	1187	1.85	2.45
ORC Pump	57.9	0.047	16.22
ORC Turbine	2506	1.95	9.38

From this table, it can be confirmed that heat exchanger pressure drops have a large impact on the turbine pressure ratio. This can be explained by the quite low temperature at the turbine inlet (131°C), lowering the evaporator saturation pressure. The compressor and ORC consumed/produced energy during a battery cycle (4h charge - 2h discharge) is also quite similar (4748 kWh for the compressor and 5012 kWh for the turbine). This shows a good charge/discharge balance in the machine parameters and operating conditions.

5 Conclusion

A novel Electro-Thermal Energy Storage (ETES) featuring a high temperature heat pump (HTHP), an Organic Rankine Cycle (ORC) and a Kelvin cell-based Phase Change Material (PCM) thermal storage was studied in this work. A preliminary investigation of this system as a thermally integrated electricity storage solution for the 80MWe Kızıldere-2 geothermal power-plant from Zorlu Enerji was performed. This investigation consisted in:

- **The choice of the working fluid:** 14 refrigerants were investigated. Cyclopentane was chosen because it offers a good compromise between performance (through a high round-trip efficiency (RTE)) and system compactness (through a high volumetric heat capacity), allowing better prospects for system profitability. It also satisfies the system requirements in terms of critical temperature and air leakage risks at ambient temperature.
- **The choice of architecture:** 5 architectures were tested. It showed that the reference case (HTHP with an internal heat exchanger together with an ORC featuring a recuperator) had the best RTE. Moreover, the addition of a preheater slightly decreased the efficiency (-1.15%) but increased the output work by 20.8%. This justified the choice of such an architecture. The ETES with these choices could reach a RTE of 86% with an input work of 1.23MWe and an output work of 2.62 MWe. The RTE can reach that high-value thanks to the use of waste heat from the plant process.
- **The introduction and assessment of impact on the performance of irreversibility factors.** This study allowed to choose coherent values for allowable heat exchanger pressure drops, secondary fluids use and heat exchanger pinch point temperature differences. This while optimizing the pre-established parameters for better performance. It shows that the pressure drops in low-pressure side heat exchangers were the most important for both machines. However,

the most important parameter for the whole system is the storage PCM melting temperature. The result of the addition of irreversibility factors and the tuning of ETES parameters is a system with a RTE of 85% with an input work of 1.19MWe and an output work of 2.51 MWe.

- The **pre-size** of the heat exchangers together with the **requirements** for the sizing of turbomachines.

This investigation shows positive prospects of the system performance, feasibility and profitability. However, these aspects still need to be proved by modelling the system in more details. Future work will consist in the detailed sizing of the components to verify their performance assumptions, the dynamic study and charge optimization of the whole system together with the study of an energy management system featuring this machine.

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References

- [1] "IEA: Grid-scale Storage", [Online]. Available: <https://www.iea.org/energy-system/electricity/grid-scale-storage> [Accessed 15 November 2024]
- [2] M. Gimeno-Gutiérrez and R. Lacal-Arán, "Assessment of the European potential for pumped hydropower energy storage based on two existing reservoirs", [Online]. Available: <https://doi.org/10.1016/j.renene.2014.10.068>
- [3] Fan et. al., "Life cycle assessment of electric vehicles' lithium-ion batteries reused for energy storage", [Online]. Available: <https://doi.org/10.1016/j.est.2023.108126>
- [4] Arpagaus et. al., "High temperature heat pumps: Market overview, state of the art, research status, refrigerants, and application potentials", *Energy*, vol. 152, pp. 985–1010, Jun. 2018, [Online]. Available: <https://doi.org/10.1016/j.energy.2018.03.166>
- [5] Longo et. al., "Experimental assessment of the low GWP refrigerant HFO-1234ze(Z) for high temperature heat pumps", *Experimental Thermal and Fluid Science*, vol. 57, pp. 293–300, Sep. 2014, [Online]. Available: <https://doi.org/10.1016/j.expthermflusci.2014.05.004>
- [6] Fukuda et. al., "Low GWP refrigerants R1234ze(E) and R1234ze(Z) for high temperature heat pumps", *International Journal of Refrigeration*, vol. 40, pp. 161–173, Apr. 2014, [Online]. Available: <https://doi.org/10.1016/j.ijrefrig.2013.10.014>
- [7] Bamigbetan et. al., "Review of vapour compression heat pumps for high temperature heating using natural working fluids", *International Journal of Refrigeration*, vol. 80, pp. 197–211, Aug. 2017, [Online]. Available: <https://doi.org/10.1016/j.ijrefrig.2017.04.021>
- [8] Frate et. al., "Analysis of suitability ranges of high temperature heat pump working fluids", *Applied Thermal Engineering*, vol. 150, pp. 628–640, Mar. 2019, [Online]. Available: <https://doi.org/10.1016/j.applthermaleng.2019.01.034>
- [9] Gómez-Hernández et. al., "Carbon dioxide and acetone mixtures as refrigerants for industry heat pumps to supply temperature in the range 150–220 °C", *Energy*, vol. 269, p. 126821, Apr. 2023, [Online]. Available: <https://doi.org/10.1016/j.energy.2023.126821>
- [10] Bertinat et. al., "Fluids for high temperature heat pumps", *International Journal of Refrigeration*, vol. 9, no. 1, pp. 43–50, Jan. 1986, [Online]. Available: [https://doi.org/10.1016/0140-7007\(86\)90152-0](https://doi.org/10.1016/0140-7007(86)90152-0) [Accessed 15 November 2024].
- [11] Kakaç et. al., "Heat Exchangers: Selection, Rating, and Thermal Design", 3rd edition. Taylor & Francis Group, 2012.
- [12] Esteves et. al., "Evolution of global heat transfer coefficient on PCM energy storage cycles", [Online]. Available: <https://doi.org/10.1016/j.egypro.2017.10.318> [Accessed 15 November 2024].
- [13] Esteves et. al., "Test of Two Phase Change Materials for Thermal Energy Storage: Determination of the Global Heat Transfer Coefficient", [Online]. Available: <https://doi.org/10.3390/chemengineering2010010> [Accessed 15 November 2024].
- [14] Horbalyuk et. al., "Comparative Heat Transfer Data for Solid-Liquid Phase Change of D-Mannitol and Adipic Acid", [Online]. Available: https://www.scirp.org/pdf/epe_2022112315073062.pdf [Accessed 15 November 2024].