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Proceedings of the 7th International Seminar on ORC Power Systems

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TECHNICAL COMPARISON AND EVALUATION OF A PUMPED THERMAL ENERGY STORAGE WITH TWO DIFFERENT DESIGNS

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ABSTRACT

Increased worldwide demand for energy (especially electricity), rising energy costs, and heightened environmental concerns are factors that continually press for the adoption of technologies to exploit renewable energy sources (RES). In order to adopt these systems, large-scale energy storage systems are needed to ensure the security of supply in energy systems due to the increasing share of renewable energy sources in the energy mix. The paper proposes a new form of energy storage technology that utilizes a combination of a heat pump and an organic Rankine cycle, referred to as Carnot Batteries (CB), which can store thermal energy for several hours to a few days and transform it back to electrical energy. Two different configurations of Carnot Batteries were tested using models created in Matlab programming software, with six working fluids used for technical optimization. The second, more complex configuration increased the roundtrip efficiency mainly due to the better performance of the two-stage heat pump.

1 INTRODUCTION

Rising fossil fuel prices and environmental concerns have led to a growing interest in exploiting renewable energy sources. However, their intermittent nature presents a challenge to the stability and security of power systems and grids. Energy storage systems have emerged as a feasible solution to address this issue, and over the years, various energy storage processes have been developed and implemented (Jafari et al., 2022). Despite their gains, existing energy storage systems such as pumped hydro storage and compressed air storage technologies are limited by geographical features, while battery-based storage systems face limitations such as short lifespan and high economic cost (Gallo et al., 2016).

In recent years, Carnot battery energy storage, which is based on the Carnot cycle, has emerged as a promising approach to overcome the limitations of existing energy storage systems (Nadeem et al., 2019). The Carnot cycle is a thermodynamic cycle that has been used in the field of energy conversion for over a century. Various types of Carnot batteries exist based on the type of thermodynamic cycle used in the charge and discharge phases. Rankine, Brayton, transcritical CO_2, and other cycles are examples of thermodynamic cycles employed in Carnot batteries (Dumont et al., 2020). Furthermore, Carnot batteries can achieve greater versatility by utilizing various thermodynamic cycles for upstream and downstream cycles and employing different heat storage methods. One of the most commonly used and well-known Carnot batteries is a system based on an organic Rankine cycle (ORC) process, a thermal energy storage (TES), and a variable heat pump (VHP) (Weiss & Zinn, 2016).

There are several advantages to using CBs. They have a low cost per stored kWh, are long-lasting, and can be utilized for applications ranging from 100 kW to 100+ MW. Additionally, some types of CBs can be powered by both electric energy and heat, which is particularly important. The heat required can be obtained from renewable sources, such as solar or geothermal energy, or from waste heat recovery. This feature allows the technology to function as a hybrid between a storage unit and a power production system, with the added benefit of utilizing low-temperature heat ($T \leq 100$ °C). (Frate et al., 2020).

In this work, we will provide an in-depth analysis of Carnot Batteries, including their working principles, comparing different types of Carnot Batteries, highlighting the advantages and disadvantages of each. The aim of this paper is to examine and compare two different layouts of a Carnot battery based on ORC and VHP technologies for charge phase for discharging phase, respectively.

2 CARNOT BATTERIES

Carnot Batteries work by establishing a thermal gradient between a high temperature reservoir (HTR) and a low temperature reservoir (LTR). Electric energy is used to charge the system by forcing heat flow against the natural gradient, thus storing thermal exergy. In the discharge phase, the heat flows from the hot environment to the cold one, and this flow is used to produce work in a heat engine.

According to O. Dumont et al. (Dumont et al., 2020), a Carnot Battery (CB) is defined as an electrical energy storage (EES) technology where there is always an electric input, and an electric output. A thermal input (waste heat, geothermal water, concentrated solar power, …) may be used to improve the performance of the CB; however, its primary purpose remains the storage of electric energy. Similarly, the battery may output useful thermal energy, but the electric output must be comparable with the electric input. In practice, the reservoirs may be physical ones, such as water tanks or solid materials, or their role may be taken up by the environment (for example, the ambient air). Similarly, electric heat pumps or resistance heaters may be used for charging, and any heat engine (Rankine, Brayton, others) or even a thermoelectric generator may be used for discharging.

The roundtrip efficiency of a PTES is the common parameter for evaluation the performance and calculated, generally, as the product of the COP of the heat pump with the heat engine efficiency (if thermal storage efficiency is not considered). Carnot Batteries offer roundtrip efficiencies (ε_{rt}) in a wide range depending on their boundary conditions (Dumont et al., 2020), (Dumont & Lemort, 2020; Steinmann et al., 2019; Weitzer et al., 2022), low energy costs (Jafari et al., 2022)(Dumont et al., 2020), and high lifetimes (Dumont et al., 2020). These systems are mostly competitive for electricity storage on the scale of several hours, in situations that demand low Power/Capacity (kW/kWh) ratios, with values of 1/4 h-1 and lower. They have a very low environmental impact, and no dependence on geographical conditions (Jafari et al., 2022), making them a suitable competitor to PHS and CAES, whose geographical constraints pose a considerable limit to their exploitability on a large scale. For local, small-scale implementations, Carnot Batteries may also present an adequate replacement for chemical batteries, which are often expensive and environmentally unsafe. 392

Additionally, an important advantage of Carnot Batteries is the ability to integrate additional thermal reservoirs (such as industrial waste heat), which act as additional exergy sources (Dumont et al., 2020). This thermal integration increases the ε_{rt} of the system, potentially to values greater than unity (>100%), by decreasing the work input or increasing the work output. In Table 1 a brief summary is given of the technical characteristics of some different Carnot battery technologies.

Table 1: Technical summary of Carnot Battery technologies (adapted from (Dumont et al., 2020)).

Cycle	Brayton Cycle	Electrical heater and Rankine Cycle	Heat Pump and Rankine Cycle
Power [MW]	Up to 100	Up to 100	Up to 10*
Energy [MWh]	Up to 400	Up to 400	Up to 40*
Temp [°C]	[-70:1000]	Up to 750	Up to 150
Compactness [kW/m^3]	25	~4	[0.05–1.72]
Compactness [kWh/m3]	200	~36	[0.2–207]
Self-discharge	medium	Very low	[30–73]
ε_{rt} [%]	[60-70]	[12-55]	[70-150]**
Price [$/kW]	[395-875]	~376	[272–468]
Price [$/kWh]	[55–198]	~94	[68–117]

Estimated TRL	5	9	7
Typical fluids	Argon, Air	Water	R1233zd(E), CO_2, NH_3, water
* Possible to extend by association in series ** Thermally integrated			

If a heat pump is used to charge the thermal storage, the system is designated Pumped Thermal Electrical Storage. These systems may use Brayton heat pumps (HP) with Brayton heat engines (HE), Vapor Compression Heat Pumps (VCHP) in conjunction with Rankine Cycles (RC), or Brayton heat pumps with Rankine Cycles (Frate et al., 2020). The presence of a heat pump for the charge cycle is advantageous if the Coefficient of Performance (COP) is higher than one. This thermal integration allows an increase in efficiency without complex modifications to the thermodynamic cycles and makes the PTES more flexible, as it may now receive two inputs (electric and thermal) instead of just one – in this sense, Thermally Integrated PTES (TI-PTES) may be seen as a hybrid energy storage and waste heat power plant.

Brayton PTES is usually comprised of a Brayton heat pump, based on the inverse Brayton cycle, and a Brayton heat engine, typically working between two sensible reservoirs (Dumont et al., 2020). The typical layout contains two thermal reservoirs and four machines (two compressors and two expanders), however in a reversible system this number could reduce to two. In a Brayton cycle, the working gas is compressed, heated and expanded, and then cooled before the next compression. As the work produced by expansion is greater than the work used in compression, the expander drives the compressor with a net positive work output (as with any heat engine, the driving force is a temperature gradient between the two reservoirs) – this is the cycle followed in the discharge process. For charging, the inverse process occurs – the gas is heated and compressed, drawing heat from the LT reservoir, and then cooled, storing heat in the HT reservoir, followed by expansion (Dumont et al., 2020).

HP/RC systems combine Vapor Compression Heat Pumps with Rankine Cycles, where the VCHP charges the reservoirs with thermal exergy, and the Rankine Cycle produces work by harnessing the temperature difference between HT and LT reservoirs. By comparison with Brayton systems, Rankine-based PTES offers the advantages of high energy density and low temperature operation – these features allow for more compact storage, lower self-discharge, and potentially more efficient integration of waste heat. One of the advantages of HP/ORC systems is the use of commercially available equipment (pumps, compressors, expanders, heat exchangers, valves), which facilitates their construction (Dumont et al., 2020), (Frate et al., 2020), (Eppinger et al., 2021). 393

3 THERMODYNAMIC MODELLING

3.1 Roundtrip efficiency of the CB

For the study of Carnot Battery systems, the HP/ORC architecture was chosen due to its simplicity and practical feasibility, as well as its low temperatures that facilitate thermal integration; this is also one of the most widely studied types of Carnot Battery, so reference values are readily available.

To study the performance of HP/ORC PTES, each thermodynamic cycle was modelled in MATLAB, with resort to the REFPROP database to calculate fluid properties. The effectiveness of the Carnot battery process is evaluated through the roundtrip efficiency. The value of this variable is affected by the behaviors of the two sub-cycles of vapor compression heat pump (COP) and the organic Rankine cycle (η), and is expressed by:

$$\varepsilon_{rt} = \mathrm{COP} \times \eta \tag{1}$$

In this work, two different configurations of Carnot battery are proposed and identified in Figure 1. The first is a single stage heat pump and a simple organic Rankine cycle (left). The second, placed at right is a two-stage heat pump system for the charging process and a regenerative ORC for the discharge process. Figure 2 depict the thermodynamic cycle diagrams of the proposed configurations of the heat pump and organic Rankine cycle.

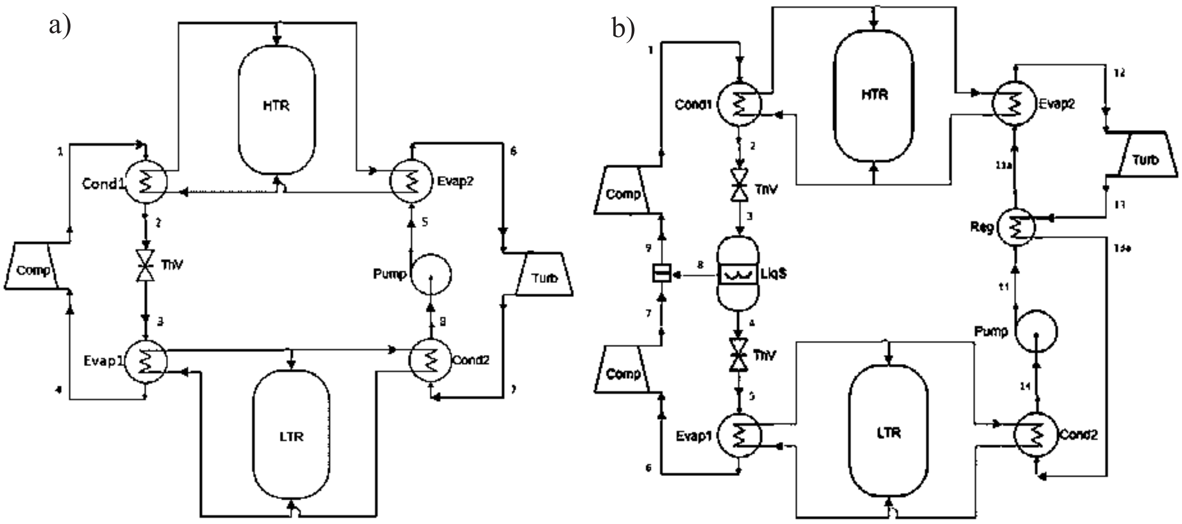

Figure. 1. Proposed Carnot battery, (a) single VCHP and ORC, (b) multi-stage VCHP and regenerative ORC.

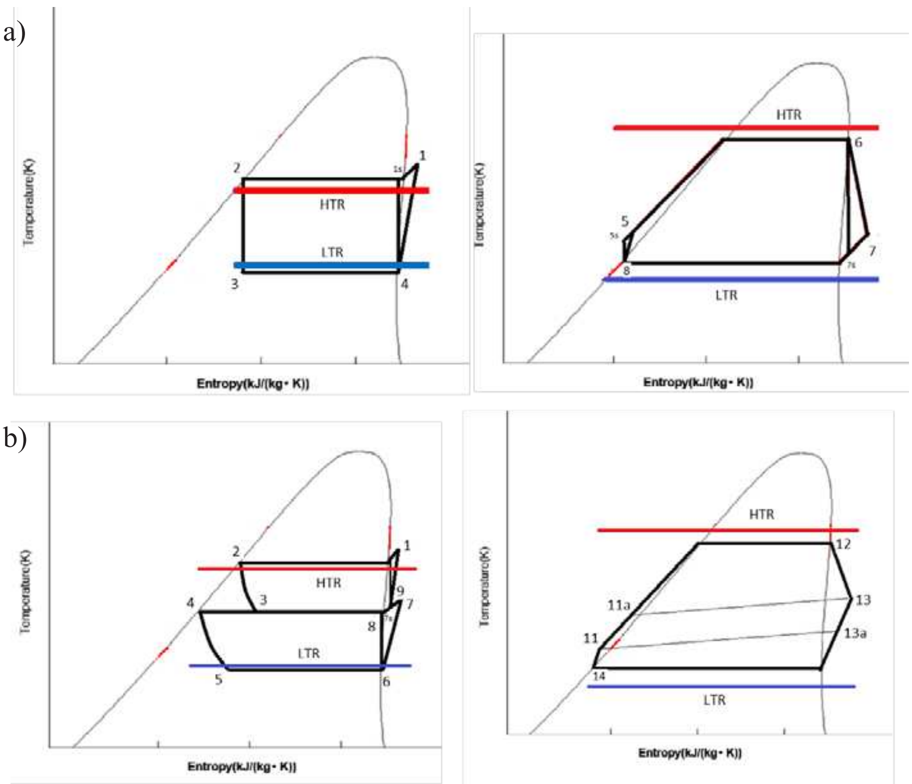

Figure. 2. T-s diagram of the proposed thermodynamic cycles, (a) single VCHP and ORC, (b) multi-stage VCHP and regenerative ORC.

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3.2 Vapor Compression Heat Pump (VCHP)

The standard VCHP cycle is shown in Figure 3 In this cycle, almost all processes involve changes in enthalpy in which case, by definition, heat and/or mechanical work (associated with pressure and volume changes) are being exchanged, neglecting changes in kinetic and potential energy of the fluid. From points 4 to 1, the enthalpy of the vapor is increased through compression, and the specific work performed on the system is equal to the enthalpy variation:

$$\dot{W}_{comp} = \dot{m}_f (h_1 - h_4) \tag{2}$$

Between 3 and 4, the fluid cools and condenses in a condenser, releasing heat into the HT reservoir at the condensation temperature:

$$\dot{Q}_{cond1} = \dot{m}_f(h_2 - h_1) \quad (3)$$

Between 3 and 4, the liquid is expanded into the two-phase region. If this happens in a throttling valve, the process is isenthalpic as heat exchanges can be neglected, and no work is performed on the surroundings. This process is highly irreversible, so it can never be isentropic.

$$h_3 = h_2 \quad (4)$$

Finally, the two-phase mixture receives energy in the evaporator and the fluid returns to a gas state, removing heat from the LT reservoir at the evaporation temperature:

$$\dot{Q}_{evap1} = \dot{m}_f(h_3 - h_4) \quad (5)$$

The COP of the heat pump in heating mode is given by the ratio between the heat released from the condenser and the energy consumed in compression. This value is generally greater than unity.

$$COP = \frac{\dot{Q}_{cond1}}{\dot{W}_{comp}} \quad (6)$$

In reality, compression may be non-isentropic and there may be superheating of the vapor at point 1, as well as subcooling of the liquid at point 3. When the compression process is non-isentropic, which is the case with all real compression processes, the specific entropy of the fluid increases from 1 to 2 – this translates into higher compression work for the non-isentropic process. The isentropic efficiency of a compressor $\eta_{s,c}$ can thus be defined as the ratio between the specific work of the compressor in the isentropic scenario and that of the real scenario:

$$\eta_{s,comp} = \frac{(h_{4,s} - h_1)}{(h_4 - h_1)} \quad (7)$$

3.3 Organic Rankine Cycle (ORC)

In the ORC, all processes involve enthalpy changes, as heat and/or mechanical work are exchanged at every step. Between 1 and 2, the vapor is expanded, generating mechanical work – the enthalpy variation is equal to the specific work produced by expansion:

$$\dot{W}_{Turb} = \dot{m}_f(h_7 - h_8), \quad (8)$$

From 2 to 3, the working fluid is condensed, rejecting heat to the LT reservoir:

$$\dot{Q}_{cond2} = \dot{m}_f(h_8 - h_5) \quad (9)$$

Subsequently, the liquid is pumped to the high-pressure level. The enthalpy variation is equal to the specific work performed by the pump; however, this value is very low as the entire process happens in the liquid phase.

$$\dot{W}_{pump} = \dot{m}_f(h_6 - h_5), \quad (10)$$

Finally, the pressurized liquid is heated and vaporized in the evaporator, drawing heat from the HT reservoir and returning to the state of point 1.

$$\dot{Q}_{evap2} = \dot{m}_f(h_7 - h_6) \quad (11)$$

The efficiency of the ORC is given by the ratio between the net energy output, and the heat input at the evaporator:

$$\eta = \frac{\dot{W}_{turb} - \dot{W}_{pump}}{\dot{Q}_{evap2}} \quad (12)$$

As with the vapor compression cycle, the real Rankine cycle involves irreversibilities, some of the most significant being the non-isentropic behavior of the expander and the pump. The isentropic efficiencies of these machines are defined the same way as with the compressor – a ratio between an ideal and a non-ideal amount of work between two states:

$$\eta_{s,turb} = \frac{(h_7 - h_8)}{(h_7 - h_{8,s})} \quad (13)$$

$$\eta_{s,pump} = \frac{(h_{6,s} - h_5)}{(h_6 - h_5)} \quad (14)$$

3.4 Multi-stage VCHP

The two-stage heat pump can be seen as a double loop with two stages of compression and expansion where the top loop considers all working fluid and the bottom loop only partially, as described next

$$\dot{m}_4 = \dot{m}_3 - \dot{m}_8 \quad (15)$$

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In the two-stage heat pump it is considered that the pressure ratio in both expanders is equal. The intermediate pressure is then easily calculated by the expression 16 knowing the higher and lower pressures of the cycle.

$$p_3 = \sqrt{(p_1 \cdot p_6)} \tag{16}$$

After the first expansion in the throttle valve, a liquid separator are included to assure that only liquid flows to the second throttle valve while the vapor merges with the working fluid from the bottom loop between the two compressors. The energy balance in the liquid separator is given by:

$$\dot{m}_8(h_3 - h_8) = \dot{m}_4(h_4 - h_3) \tag{17}$$

Similarly, the mixture chamber between the two compressor joins the working fluid from the bottom loop and the saturated vapor after the first expansion.

$$\dot{m}_9 \cdot h_9 = \dot{m}_8 \cdot h_8 + \dot{m}_7 \cdot h_7 \tag{18}$$

The COP of this configuration can be calculated by expression 19.

$$COP = \frac{(h_1 - h_2)}{(h_1 - h_9) + (\dot{m}_4/\dot{m}_3)(h_7 - h_6)} \tag{19}$$

3.5 Regenerative ORC

The inclusion of the intermediate heat exchanger (regenerator), requires to include the energy balance equation in the regenerator, which adds points 11a and 13a.

$$\dot{m}_f(h_{11a} - h_{11}) = \dot{m}_f(h_{13} - h_{13a}) \tag{20}$$

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3.6 Thermodynamic models

The steady-state models were developed in MATLAB software to compare the system performance of the different cycles. The calculations are based on energy and mass conservation with the following assumptions:

- Pressure drops and heat losses in piping and heat exchangers are neglected, as they are much lower compared with the pressure variation of the cycle.
- Heat exchangers are ideal with a zero-approach temperature difference (pinch = 0°C).
- The evaporation and condensation temperatures of the refrigerant are the same as the temperatures of the heat source (HTR) and sink (LTR), meaning that no superheating and subcooling are used.
- All expansion valves operate adiabatically and provide isenthalpic expansion processes.
- All compressors operate adiabatically with a specific isentropic efficiency of $\eta_{comp} = 0.8$.
- The turbines operate adiabatically with a specific isentropic efficiency of $\eta_{turb} = 0.7$, typical value for volumetric expanders in ORC systems.
- The pumps operate adiabatically with a specific isentropic efficiency of $\eta_{pump} = 0.5$.
- The separators have perfect separation efficiencies and the fluid outlets are saturated vapors and liquids.
- The higher and low temperature reservoir are assumed as latent heat storage type and the temperature remains constant in charging and discharging processes

4 RESULTS AND DISCUSSION

The model exploration started by doing a parametric exploration varying the LTR temperature from 20°C and HTR temperature from the LTR up to the critical temperature of each working fluid tested with a 2°C step. Figure 3 present the results for the working fluid Isopentane, for the COP of the heat pump, the η of the organic Rankine cycle and the last for the roundtrip efficiency for the single VCHP and ORC (left) and multi-stage VCHP and regenerative ORC (right).

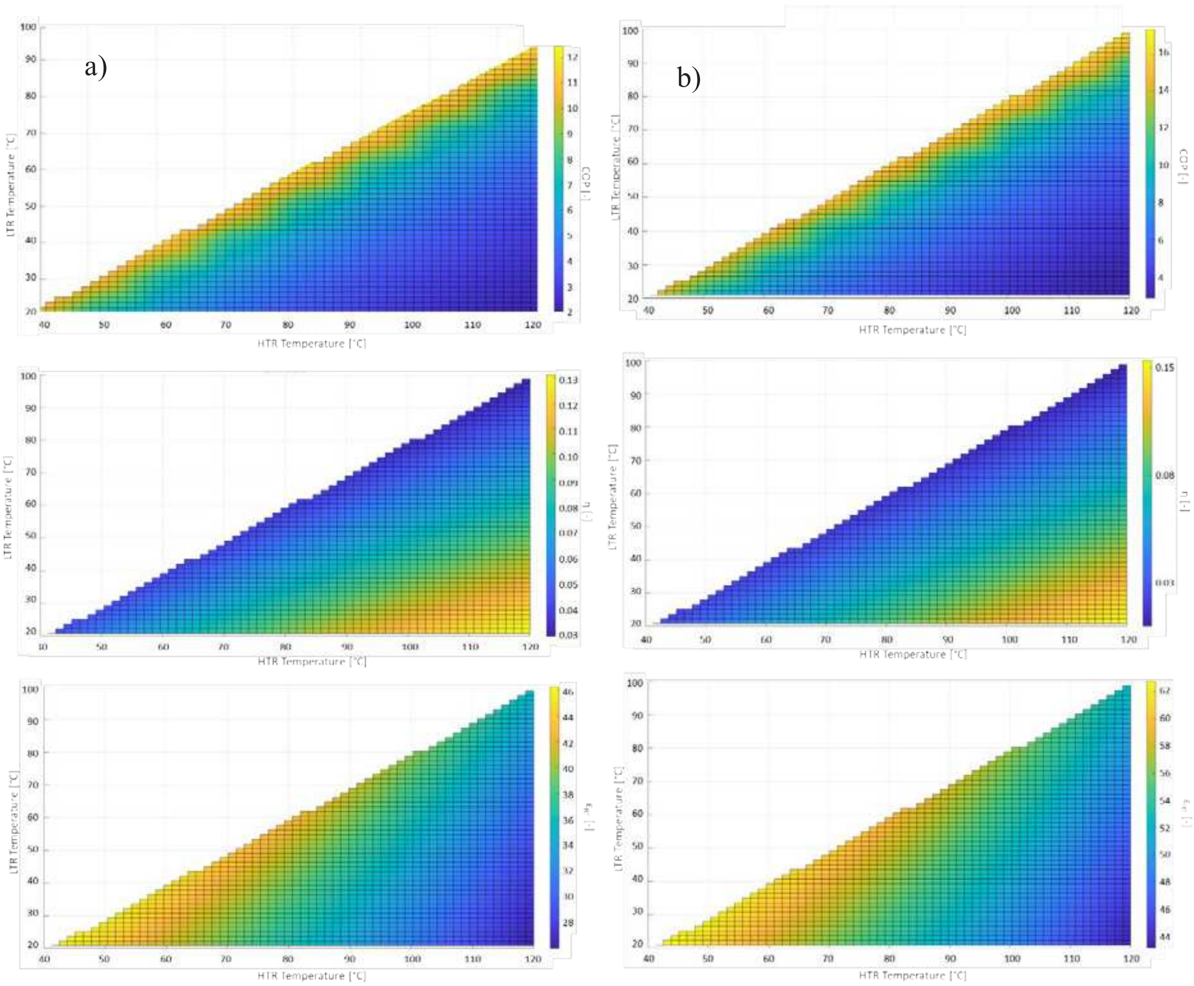

Figure. 3. Exploration results for COP, η_{ORC} and ε_{RT} with several temperatures combinations of HTR and LTR for configuration (a) single VCHP and ORC, (b) multi-stage VCHP and regenerative ORC.

Figure 3 indicates that the COP of the heat pump increases as the temperature difference between the two reservoirs decreases. Specifically, when the hot reservoir temperature is 120 °C and the cold reservoir temperature is 100 °C, the heat pump achieves its highest COP. Conversely, the efficiency of the Organic Rankine cycle is greater when there is a larger gap between the cold and hot reservoir temperatures. The maximum P2P is achieved at lower temperatures and a smaller temperature difference due to the interplay between the COP of the heat pump and the efficiency of the organic Rankine cycle. This relationship is linked to the specific enthalpy variation with temperature, which affects the compression and expansion of the cycles.
The improvement between the previous simple model and the sophisticated model, is notorious, the COP rises from values of 13 to values around 17 which indicates that the multi-stage heat pump can be an addition for this system. The addition of a regenerator in the organic Rankine cycle it is proved to be an improvement to the cycle efficiency, increasing the efficiency from 0,13 to 0,15. The biggest improvement is on the roundtrip efficiency where the increment is 15%.
Finally, a minimum temperature condition of 20°C was added between the two reservoirs and the condition that maximized the roundtrip efficiency of the cycle was obtained. The results for the 6 working fluids considered in this work are showed in Table 2.

Table 2: Best case scenario of roundtrip efficiency for both configurations.

			single VCHP and ORC			multi-stage VCHP and regenerative ORC		
Fluids	T_{HTR} [°C]	T_{LTR} [°C]	COP [-]	η_{ORC} [%]	ε_{RT} [%]	COP [-]	η_{ORC} [%]	ε_{RT} [%]
R1233zd	40	20	11,15	4.15	46,35	14,87	4,17	62,14
R1234yf	40	20	10,39	3.80	39,43	14,03	3,76	52,86
R11	40	20	11,36	4,23	48,09	15,05	4,20	63,35
R236ea	40	20	10,89	4,06	44,18	14,64	4,11	60,32
R245fa	40	20	11,07	4,13	45,77	14,80	4,15	61,50
Isopentane	40	20	11,14	4,16	46,30	14,90	4,22	62,92

The most favorable outcome is obtained using R11, which yields a roundtrip efficiency of 48% and 63% when the hot reservoir temperature is at 40 degrees and the cold reservoir temperature is at 20 degrees for configurations a) and b), respectively. However, due to its prohibition from use, R11 has only been tested for research purposes. Therefore, the optimal choice of working fluid is either Isopentane or R1233zd, with the main distinction being that the former is flammable.
It is very interesting that the maximum roundtrip efficiency is obtained for the lowest temperature lift allowed. It can be concluded that the COP of the heat pump has more influence than the Rankine cycle efficiency on this parameter. Moreover, this result should be even more noticeable if the heat losses in the reservoirs and temperature differences in the heat exchangers were included in the model as the losses increase with higher temperature differences for the ambient which would further decrease the overall efficiency for the higher temperatures of the reservoirs and be minimal for the temperatures obtained in this optimization.

5 CONCLUSIONS

In this study, a preliminary assessment of a Carnot batteries as an innovative electrical energy storage system was made. The study converged in several points with the bibliography already made and presented an interesting new perspective for future studies.
The main conclusions reached in this assessment were:

- The best working fluid (excluding R11) for CBs configurations a) and b) is R1233zd and the isopentane, respectively;
- The addiction of a new stage to the heat pump increased the COP by 22%;
- The ORC efficiency slightly improved with the addiction of the regenerator by 3%;
- The second proposed CB configuration got an improvement on the roundtrip efficiency of 26%.

The COP of the heat pump increases as the temperatures of the thermal reservoirs are closer while the ORC efficiency increases for larger spread of the reservoir's temperatures. Moreover, in all fluids and both configurations the COP of the heat pump surpasses the importance of the ORC efficiency as the highest roundtrip efficiency obtained coincides with the highest COP condition.
Future work will be to sophisticate this model to include temperature differences between the exchangers and reservoirs, taking into account the pitch point, model the components with variable efficiencies instead of assuming constant efficiencies, and include the temperature change in the reservoirs over time at charging and discharging, thus making the model transient.

NOMENCLATURE

CAES	Compressed Air Energy Storage	
CB	Carnot battery	
COP	coefficient of performance	(-)
EES	Electrical Energy Storage	

Subscript

cond	condenser
comp	compressor
evap	evaporator

h	specific enthalpy	(kJ/kg)
HTR	high temperature reservoir	
HE	Heat Engines	
LTR	Low temperature reservoir	
$\dot{m}$	mass flow rate	(kg/s)
ORC	Organic Rankine Cycle	
PHS	Pumped Hydro Storage	
Q	Heat flux	(kW)
RES	Renewable energy source	
RT	Roundtrip	
TES	Thermal Energy Storage	
VCHP	Variable Capacity Heat Pump	
W	Work	(kW)
η	ORC efficiency	
ε	RT efficiency	

f	working fluid
pump	pump
s	isentropic
turb	turbine

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The ORC conference, organized biennially, stems as the only conference that is specific to ORC technology, therefore gathering a diverse community whose affiliation spans across all the interested stakeholders, not only in this particular technology but also and in a broader context, in the energy transition. Original equipment manufacturers, professional associations, end-users, investors, policy makers, academics, scientists feel at home at ORC 2023.

The almost 100 proceedings in this book cover a wide variety of topics, from fundamentals to system integration through component design, accounting for thermodynamic performance as well as component design. In addition to this, and as a new track in 2023, works on heat pump technology were also accepted in order to raise awareness of the strong ties between both technologies, specifically in energy storage applications.

This book provides an excellent overview of the current maturity of power systems based on Organic Rankine Cycle technology for applications as diverse as geothermal and waste heat recovery in industry or downstream of other prime movers (e.g., marine applications). It is also an excellent source of information to understand the current challenges faced by the technology, stemming from a very competitive market and increasingly stringent environmental regulations.

The organizers of ORC 2023 hope that the reader finds this work as exciting as the attendees to the conference and, maybe, make the decision to join the 8th edition to the conference in 2025.

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