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(coordinadores)

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DESIGN, MODELING AND SIMULATION OF AN EXPERIMENTAL RANKINE PTES SYSTEM

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ABSTRACT

Pumped Thermal Energy Storage (PTES) is a promising technology for the long-term storage of electrical energy by using thermal energy reservoirs. PTES can be mainly classified into system configuration employing Joule-Brayton cycles and systems using Rankine cycles. With respect to Rankine cycles, the coupling between a high-temperature heat pump with an ORC power system is attracting increasing interest. In this framework, an experimental Rankine PTES system based on the coupling between a kW-scale ORC system with a high temperature heat pump is going to be developed and commissioned. In this paper, the procedure followed for the design of such experimental facility is presented and discussed. Four main configurations are proposed and compared in terms of achievable system roundtrip efficiency and minimization of the exergy destruction rate. Results demonstrate that roundtrip efficiencies lower than 40% are reached with the use of out-of-shelf devices, while by a proper system configuration and the customized design of the heat engine and heat pump, roundtrip efficiencies higher than 60% can be achieved.

1 INTRODUCTION

Long-term energy storage is currently considered as a key technology to increase the penetration of renewables in the energy mix, and thus accelerate the transition towards a decarbonized economy. Among the possible technologies, Pumped Thermal Energy Storage (PTES) emerges as a promising solution (Tian & Xi, 2022). During charging phase, surplus electricity feeds a Heat Pump (HP), which transfers heat from a cold reservoir to a hot one. The electrical energy is therefore stored in the form of heat in a Thermal energy storage (TES) system. During the discharging phase, a heat engine uses the heat stored at the hot reservoir to convert it back to electrical energy. Main advantages of PTES systems include low costs, high scalability, flexible operation, long operating life, small footprint and low self-discharge rate (Benato & Stoppato, 2018). Usually heat engines employed within PTES architecture are based on two main thermodynamic cycles: Joule-Brayton and Rankine. Brayton PTES systems can operate at higher temperatures with consequent higher efficiencies. Nevertheless, PTES systems based on Rankine technology, especially Organic Rankine Cycles (ORC), are more compact, their components are technically mature and can convert heat available at lower temperatures, which facilitates thermal integration with the TES and heat pump system and in turn reduces costs (Dumont & Lemort, 2022). A typical Rankine PTES system is composed by a heat pump, two or more thermal reservoirs and an ORC unit. The heat pump, in its simplest configuration, embeds a compressor, a condenser, an expansion valve and an evaporator. More complex layouts include a recuperator or a second compressor to improve performance. The implementation of trans-critical thermodynamic cycles is also considered a promising way to improve heat pumps efficiency (Dumont & Lemort, 2022). On the other hand, an ORC engine in its more basic form presents an evaporator, a turbine, a condenser and a pump. Possible ways of improving performance also in this case include the introduction of a recuperator, the heat recovery stage occurring above the working fluid critical pressure, the use of flash or triangular cycles as well as the adoption of zeotropic mixtures (Dumont & Lemort, 2022). In the literature, some works explored these possible alternatives, for instance (Tian & Xi, 2022) compared four Organic Flash Cycles with a basic ORC and found that the latter presents better economic metrics, higher roundtrip efficiency and increased exergy efficiency. The analysis also revealed that improving the roundtrip efficiency and the economic viability of Rankine PTES systems is critical to accelerate the path towards commercialization. Concerning the working fluid, several studies identify the R1233zd(E) as an optimal solution (Eppinger et al., 2021,), with R1234ze(Z), R245fa and pentane showing similar performance (Frate et al., 2020). In general, the heat pump compressor represents the largest cost item in such systems and, while Rankine PTES are currently not cost-competitive compared to other storage solutions, they are expected to become bankable in the future due to forecasted cost reductions (Tillmanns et al., 2022).

Despite the research available on this technology, few works have been devoted to the experimental investigation of Rankine PTES systems. Among them, CHESTER project (*CHESTER Project*, 2018) employs a high-temperature HP and an ORC unit and stores both sensible and latent heat. Both units employ R1233zd(E) as working fluid (Hassan et al., 2022).This pilot plant has already been launched and first experimental tests are being conducted (Steger et al., 2021). However, the availability of experimental data is still scarce, and more research is required also at a theoretical level to assess the techno-economic feasibility of such technology (Dumont & Lemort, 2022).

For this reason, an experimental PTES plant composed by a high temperature HP, two thermal reservoirs and a kW-scale ORC is going to be developed and commissioned with the objective of advancing the Technology Readiness Level (TRL) of Rankine PTES systems. The experimental activity will allow to characterize the performance of the storage system during nominal, off-design and transient operating conditions, with the possibility of identifying the most critical parameters affecting the system performance during charge and discharge operations. In this paper, the methodology followed during the preliminary design phase of the experimental set-up is presented. Four different system configurations will be considered and compared. The basic one starts from considering commercially available units, while the remaining three explore more complex layouts tailored for the application. The improvement of the latter configurations compared to the basic one will be measured in terms of increasing the roundtrip efficiency of the overall energy storage system and minimizing the exergy destruction rates at a component level. The system performance has been also investigated for different operating conditions of the heat pump, mainly varying the inlet temperature at the heat pump evaporator. **646**

2 SYSTEM CONFIGURATION

As shown in Figure 1, the experimental facility includes a heat pump and an ORC system integrated with each other through a TES subsystem. The latter consists of two storage tanks at different temperature levels (hot tank and cold tank) and two pumps. These pumps enable the circulation of a heat transfer fluid (based on the expected working temperatures a thermal oil is chosen) to transfer the thermal power supplied by the heat pump to the ORC engine. During the charging phase, a high-temperature heat pump based on a reverse Rankine cycle (subsystem on the left) is used to raise the temperature of the heat transfer fluid, which is subsequently stored in the hot tank. The compressor (COMP.HP) and the thermal oil pump (PUMP2.TES) are coupled with inverter-driven electrical motors. An electric heater is included at the evaporator inlet side to simulate various inlet temperature conditions. This allows testing the performance of the heat pump under different operating conditions, representing real-life scenarios where the system is powered by a PV system as the electricity supplier and a solar thermal system as the heat source. During the discharge phase, the heat transfer fluid stored in the hot tank is used as a heat source for the ORC system (subsystem on the right). Similarly, the pumps are coupled to inverter-controlled electric motors. The thermal oil pump (PUMP1.TES) is controlled to meet the desired electrical power, while the organic fluid pump (PUMP.ORC) is operated to adjust the ORC evaporating pressure according to the controller logic.

Based on preliminary evaluations, the design temperature of the hot tank has been set at 140°C, and an ORC unit with a net power output of approximately 15 kW has been considered. Currently, in this preliminary phase of the study, four alternative configurations for the heat pump and the ORC system are being examined:

- *Configuration A*: both HP and ORC are based on products already available on the market.
- *Configuration B*: a commercial ORC system is considered while the design of the heat pump is optimized for the application considered.
- *Configuration C*: both the heat pump and the ORC system expressly conceived for this application are designed. Specifically, the introduction of an intermediate TES system and the introduction of subcooling sections downstream of the COND.HP and upstream of the EVAP.ORC is considered.
- *Configuration D:* same heat pump design as configuration B and the ORC system equipped with an additional heat exchanger and volumetric machine able to expand the working fluid under two-phase conditions to implement an additional triangular cycle.

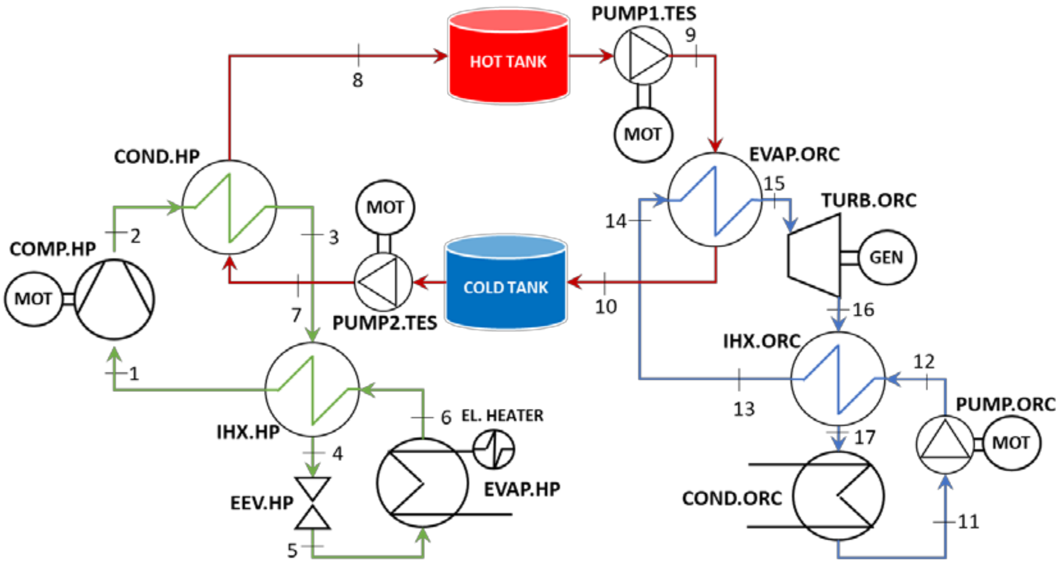

Figure 1: System configuration of the experimental Rankine PTES system.

3 METHODOLOGY

A model of the system has been developed in MATLAB software to evaluate the performance of the heat pump, TES section and ORC unit under design conditions. Thermophysical properties for water and organic fluids have been obtained through CoolProp (Bell et al., 2014) while the thermal oil has been characterized by data provided by the manufacturer (Eastman, n.d.). For both HP and ORC models, temperature, pressure, entropy and enthalpy of each point have been calculated by establishing flow mass and enthalpy balances and taking into account compressor and turbine isentropic efficiencies and pump efficiency (the assumed values are reported in Table 1), while pressure drops within components have been neglected. Based on recommendations provided by (Hassan et al., 2020), the maximum temperature reached at the outlet side of COMP.HP has been limited to 160°C.
The exergy of each subsystem has been also computed in order to analyze the exergy efficiency and the exergy losses at component level. The classical exergy rate balance equation at steady state has been implemented for each component, as follows:

$$\sum \dot{m}_i e_i + \dot{Q}\left(1 - \frac{T_a}{T_c}\right) = \sum \dot{m}_o e_o + \dot{W} + \dot{E}_D \quad (1)$$

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where $\dot{m}$ is mass flow rate of the stream substance, $\dot{Q}$ is heat flow through component boundary, T_a is the ambient temperature, T_c is the temperature at component boundary, at which heat is exchanged with the environment, e is the specific exergy of the stream (based on the system configuration only the physical exergy defines the total specific exergy of such streams), $\dot{W}$ is work rate of the component, and $\dot{E}_D$ is the exergy destruction in the component due to irreversibility. Subscripts i and o represent inlet and outlet to and from the component, respectively.

Table 1: Main assumed efficiencies in the model.

TURB.ORC isentropic efficiency	Pump efficiency	ORC Organic efficiency	COMP.HP isentropic efficiency	COMP.HP electro-mechanical efficiency	TES efficiency
0.7	0.6	0.9	0.8	0.95	0.98

The roundtrip efficiencies expressed in energy and exergy terms have been used as main performance indexes of the PTES system. Specifically, the roundtrip efficiency expressed in energy terms (η_{RT}) is defined as the ratio between the electrical energy produced by the ORC (W_{ORC}) during a complete discharge phase and that absorbed by the heat pump (W_{HP}) during a fully charging phase. By assuming that the heat pump and ORC always operates at nominal conditions, η_{RT} can be also computed as the multiplication of the Coefficient of Performance of the Heat Pump (COP_{HP}) by the efficiency of the ORC (η_{ORC}) and by the efficiency of the TES system (η_{TES}):

$$\eta_{RT} = \frac{W_{ORC}}{W_{HP}} = \frac{\dot{W}_{ORC} \cdot t_{DIS}}{\dot{Q}_{EVAP.ORC} \cdot t_{DIS}} \cdot \frac{\dot{Q}_{EVAP.ORC} \cdot t_{DIS}}{\dot{Q}_{COND.HP} \cdot t_{CH}} \cdot \frac{\dot{Q}_{COND.HP} \cdot t_{CH}}{\dot{W}_{COMP.HP} \cdot t_{CH}} = \eta_{ORC} \cdot \eta_{TES} \cdot COP_{HP} \quad (2)$$

where t_{DIS} and t_{CH} are the discharging and charging times, respectively. Instead, the exergy roundtrip efficiency is defined as the ratio of the overall exergy produced by ORC (which corresponds to electrical energy) and the overall exergy introduced in the HP, that is, the sum of the electrical energy absorbed by the compressor and the exergy introduced in the evaporator of the heat pump ($\dot{E}_{EVAP.HP}$): 648

$$\eta_{EX,RT} = \frac{\dot{W}_{ORC}}{\dot{E}_{EVAP.ORC}} \cdot \frac{\dot{E}_{EVAP.ORC} \cdot t_{DIS}}{\dot{E}_{COND.HP} \cdot t_{CH}} \cdot \frac{\dot{E}_{COND.HP}}{\dot{W}_{COMP.HP} + \dot{E}_{EVAP.HP}} \quad (3)$$

4 RESULTS

4.1 Configuration A

The first configuration analyzed concerns the use of an off-the-shelf high temperature HP and a commercial ORC unit. To evaluate the expected performance of this configuration, the technical data reported in the datasheets of the commercial units produced by Rank (Rank, n.d.) have been used as a reference. The declared nominal efficiency of the ORC unit is about 10% while the heat pump COP achieved with a temperature lift of 40°C (which means an inlet water temperature at EVAP.HP of 100°C) is about 4. Consequently, the roundtrip efficiency of this PTES system should be lower than 40%. With the aim of identifying the main exergy losses and introducing some improvements on the system configuration, an exergy analysis has been carried out. As a first step, the thermodynamic cycles of the two subsystems have been defined. The refrigerant R1233zd(E) has been assumed as working fluids for both the ORC system and the HP, since it is a fluid with a very low greenhouse effect, non-flammable and with no impact on ozone depletion. Other assumptions are a minimum temperature difference in the heat exchangers of 10°C and a subcooling of the working fluid at COND.HP of 5°C.
Figure 2 shows the T-s diagram of the ORC cycle and of the heat pump expected at design conditions with a water inlet temperature at the heat pump evaporator of 100°C. The calculated performance resembles the data declared by the manufacturer. In this configuration, the electrical power absorbed by the HP is about 30 kW while the thermal power required by the ORC unit is 15% greater than the amount provided by the HP, meaning that the system shows a reduced duration in the discharge phase.

The exergy analysis of this PTES system (Figure 3(a)) highlights that the main irreversibility occurs in the machines (TURB.ORC and COMP.HP). However, reducing these endogenous exergy destructions is hindered by the current technological state-of-art. Significant irreversibility occurs also in the heat exchangers, especially in the EVAP.ORC. These losses can be reduced by decreasing the minimum allowed temperature (with a consequent increase in the heat transfer area) but since they are mainly exogenous, its minimization can be reached only reducing the mean temperature difference. By assuming a water inlet temperature at the EVAP.HP of 100°C, an energy and exergy roundtrip efficiency of about 41% and 27%, respectively, are expected. Decreasing the water inlet temperature at EVAP.HP results in an increase in the temperature lift provided by the heat pump. Consequently, as shown in Figure 3(b), this leads to increased exergy losses and degradation of the overall PTES performance, with a drop in the roundtrip efficiencies up to 20% and a decrease in the thermal power produced by heat pump.

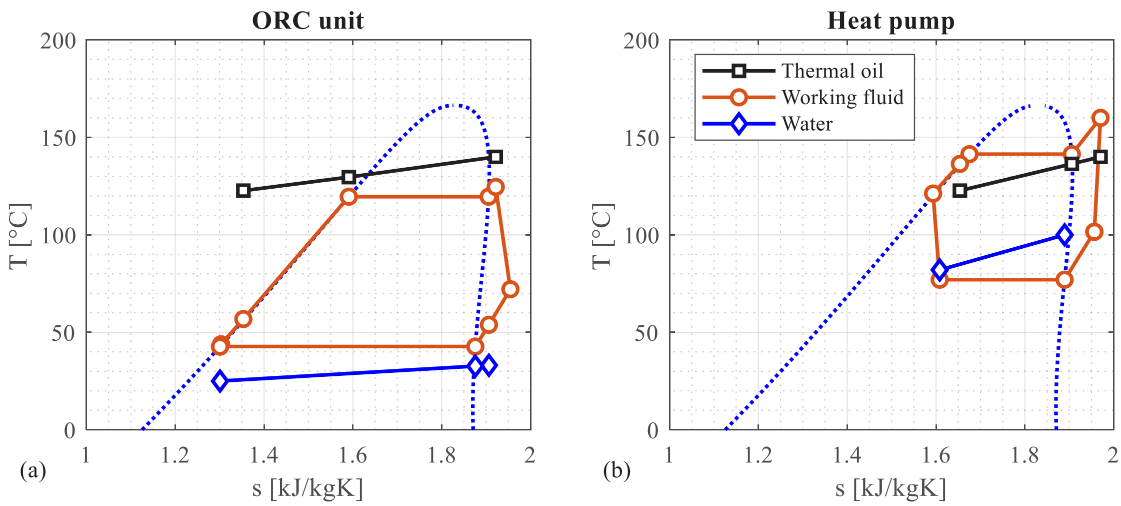

Figure 2: Thermodynamic cycle of (a) ORC unit and (b) heat pump for the configuration A.

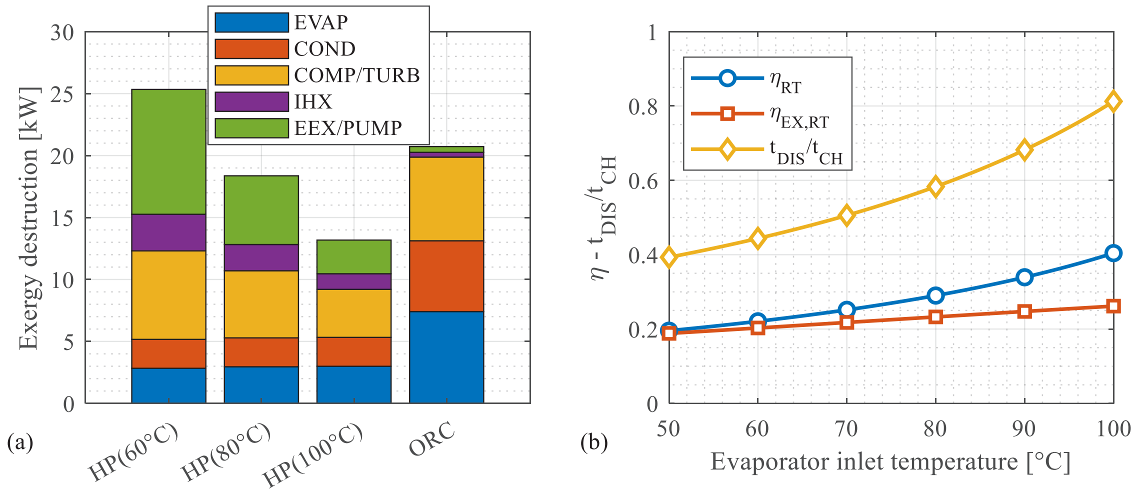

Figure 3: (a) Exergy destruction in the heat pump and ORC and (b) expected performance indexes as a function of the evaporator inlet temperature for configuration A.

4.2 Configuration B

An alternative configuration features a commercial ORC engine and a custom HP through the addition of a sub-cooler downstream of the condenser. Moreover, different types of working fluid of the heat pump are analyzed to identify the solution that allows to maximize the HP performance. In a preliminary screening step, the selection space has been reduced to a group of organic fluids considering criteria as critical temperatures falling within the range of 150-245 ºC, low flammability, as well as Ozone Depletion Potential (ODP) and Global Warming Potential (GWP) lower than 5. Afterwards, a parametric analysis aimed at maximizing the COP by varying the inlet water temperature at the EVAP.HP has allowed to further reduce the list of suitable candidates. Figure 4(a) shows the ones with the highest potential to maximize the roundtrip efficiency of the system. In general, the fluids selected show better performance compared to the fluid used in the commercial heat pump (R1233zd(E)), with an increase of 10-15% in the energy roundtrip efficiency. At higher evaporating temperature (above 60°C), n-hexane leads to the highest energy roundtrip efficiency (53%), while slightly lower performance is achieved by acetone and cyclopentane. On the other hand, acetone should be preferred below these temperatures although the performance improvement over that obtained by using n-hexane is marginal (less than 1 percentage point). To better understand the effect of the chosen working fluid on PTES performance, Figure 4(b) compares the exergy destruction between acetone and n-hexane for temperatures at the HP evaporator inlet of 60ºC and 100ºC. Compared to the previous configuration, the main advantages come from the reduction of the exergy losses in the EEX and IHX. This is mainly due to the introduction of a sub-cooler downstream of the condenser, which decreases both the average temperature difference in the IHX and the fluid cooling occurring in the EEX. At 60°C no Internal Heat Exchanger (IHX) is considered for acetone since it is a wet fluid. A reduction in exergy destruction can be observed with higher evaporating temperatures, which allow to reduce the exergy losses in the compressor, internal heat exchanger and electronic expansion valve. At an evaporator temperature of 100ºC, the exergy destruction for the n-hexane case is about 8% lower than that of acetone thanks to the lower irreversibility introduced in the EEX.

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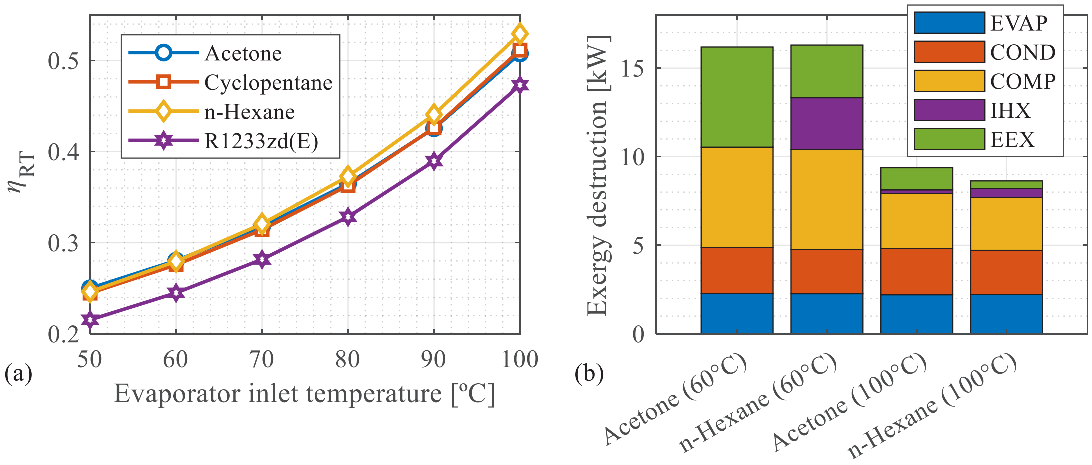

Figure 4: (a) Roundtrip efficiency and (b) exergy destruction in the heat pump at two evaporator inlet temperatures for configuration B.

4.3 Configuration C

In this third configuration, in addition to the design of a custom heat pump, the design of the ORC unit is also optimized with the aim of maximizing the roundtrip efficiency of the PTES system. The exergy analysis of the first configuration highlighted the important exergy destruction occurred in the ORC heat exchangers. To reduce it, the minimum temperature difference at the EVAP.ORC and COND.ORC is lowered to 5°C by increasing their heat transfer area. Furthermore, since the main irreversibility in the EVAP.ORC occurs in the liquid-to-liquid heat transfer stage, a preheater is introduced into the ORC layout together with an intermediate TES tank. The new arrangement allows to lower the thermal oil mass flow rate circulating in the preheater with respect to that circulating in the evaporator. The better match between the flow streams allows then to improve the heat transfer and decrease the exergy destroyed in the heat exchangers. The T-s diagram of the thermodynamic cycle realized by such a configuration is shown in Figure 5(a). Obviously, a proper modification of the HP layout with the introduction of a sub-cooler downstream of the condenser is also required. This change allows to keep the same temperature level at the intermediate TES and to operate with a ratio between mass flow rate circulating in the HP sub-cooler ($\dot{m}_{SC.HP}$) and that circulating in the condenser ($\dot{m}_{COND.HP}$) equal to the ratio between the flow rate circulating in the ORC preheater ($\dot{m}_{PHX.ORC}$) and in the subsequent evaporator ($\dot{m}_{EVAP.ORC}$). The improvements on the performance of the PTES system consequent to the layout proposed in configuration C are shown in Figure 6. A further increase in the energy roundtrip efficiency of about 8-10 percentage points is achieved compared to configuration B reaching values higher than 60% for inlet water temperature at EVAP.HP of 100°C and minimum allowed ratio between thermal oil circulating in ORC preheater and evaporator. In fact, as can be observed in Figure 6(a), the lower the ratio of the mass flow rates is, the greater the increase in the PTES roundtrip efficiency (at the same time, this also requires the introduction of a larger intermediate TES). This is mainly due to the increasingly reduction of the average temperature difference between cold and hot flow during the preheating phase of the ORC working fluid, with a consequent decrease in the exergy losses in this component. This is confirmed by Figure 6(b), where an important reduction in the exergy destroyed in the EVAP.ORC can be observed compared to previous configurations (about 2 kW instead of 7 kW obtained in the configuration A). It should be noted that there is minimum value of the ratio $\dot{m}_{PHX.ORC}/\dot{m}_{EVAP.ORC}$, which occurs when the temperature difference at the ORC preheater between the incoming organic fluid and the outgoing heat transfer fluid reaches the minimum value (5°C).

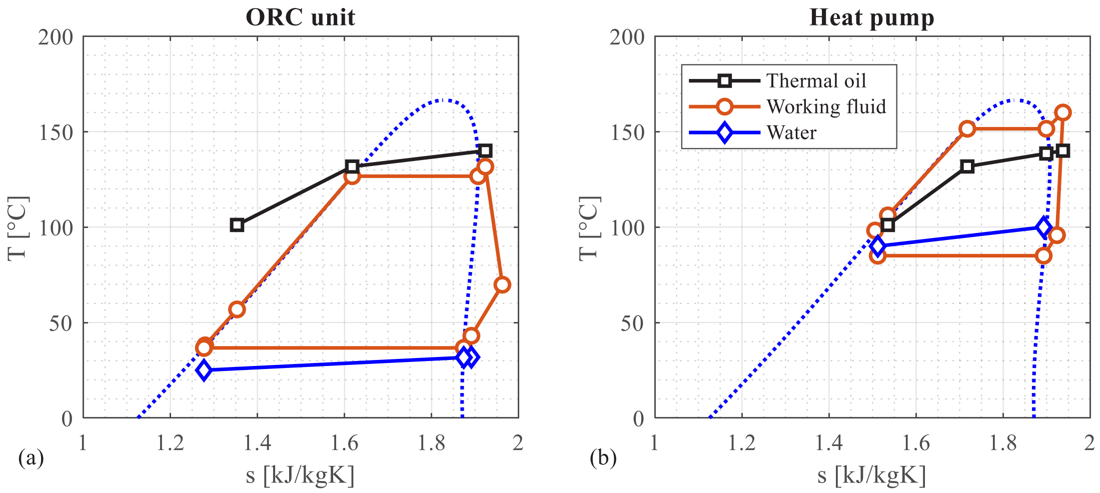

Figure 5: Thermodynamic cycle of (a) ORC unit and (b) heat pump for the configuration C.

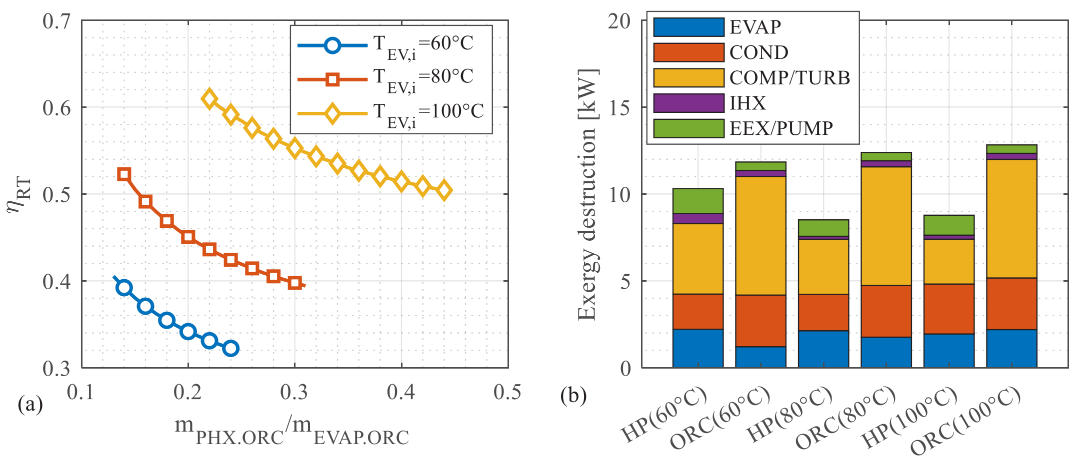

Figure 6: (a) Performance indexes as a function of the evaporator inlet temperature and (b) exergy destruction in the heat pump and ORC for configuration C.

4.4 Configuration D

In the last configuration the layout of the TES system is kept in its simplest form, with just two thermal storage tanks, but with the inclusion of a volumetric expander and preheater (PHX.ORC) in the ORC subsystem. This allows to realize an additional Trilateral Flash Cycle (TFC) by processing more mass flow rate in the circuit. In particular, the refrigerant mass flow rate is heated up in the ORC preheater and after split in two streams, one undergoing a two-phase expansion in a volumetric expander and the other one vaporizing and then expanding in the turbine. The T-s diagram in Figure 7(a) shows the thermodynamic cycle realized by this configuration. The advantage of such ORC layout relies in its operational flexibility. The addition of a TFC leads indeed to increase the utilization rate of the ORC, which from one side can exploit heat streams at lower temperatures and thus can be operated in off-design conditions, and from the other side can eventually provide larger quantities of electrical power by utilizing more heat from the TES. Figure 7(b) shows the exergy destruction for the different components of configuration D as well as the irreversibility occurring in the heat pump for different water temperature at the EVAP.HP. It can be seen that the total exergy destruction in the ORC unit is about 15 kW, which is still lower than the 21 kW occurring in the commercial configuration A (Figure **3**(a)), but higher than configuration C (13 kW, Figure 6(b)). This is due to the introduction of the trilateral flash cycle, which being able to generate a lower power per unit of thermal energy provided, reduces the overall efficiency of the power generation. At a component level an advantage of this configuration is the lower exergy destruction rate at the COND.ORC, which however is balanced by a higher exergy loss in the evaporator and the preheater, due to the higher amount of fluid processed.

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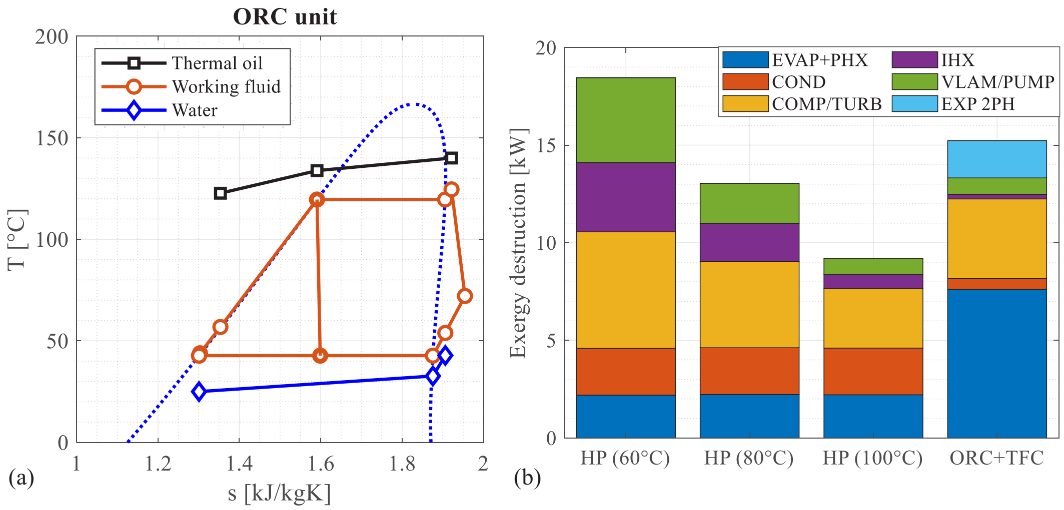

Figure 7: (a) Thermodynamic cycle of ORC unit equipped with a bi-phase expander and (b) exergy destruction in the heat pump and ORC for configuration D.

5 CONCLUSIONS

This paper discussed an experimental Rankine PTES system that is going to be developed and commissioned at University of Cagliari. The design of the facility has been investigated and multiple configurations have been proposed. Three different layouts have been compared to a more conventional solution composed by a commercial ORC and heat pump units integrated with a TES system. It has been demonstrated that by adopting a heat pump and an ORC unit available on the market, important exogenous exergy destructions should occur and expected roundtrip efficiencies lower than 40% should be achieved also with high temperatures at the evaporator inlet side of the heat pump. The three modified configurations allow to reduce the exergy destruction compared to the fully commercial one and so, to improve roundtrip efficiency. When optimizing the heat pump, n-hexane is found to be the optimal working fluid that allows to reach both the highest energy and exergy roundtrip efficiencies (up to 53%) when the inlet water temperature at the evaporator exceeds 60°C, while for lower temperature a wet working fluid as acetone should be preferred. A roundtrip efficiency higher than 60% could be achieved by optimizing the heat transfer between the TES and the ORC system. This has been done by adding a TES tank at intermediate temperature and a dedicated preheater introduced in the ORC layout. The last configuration analyzed, obtained by adding a two-phase expander and a pre-heater upstream the evaporator, leads to slightly lower roundtrip efficiency, 55%, but allows to have a higher operational flexibility in off-design conditions as well as during the power discharge phase. All the obtained results are based on data declared by manufacturer that will be validated during the experimental activities. Moreover, the obtained energy and exergy roundtrip efficiencies are based on the assumption that both the heat pump and the ORC unit always operate under nominal conditions. Obviously, as energy storage, the PTES system will frequently operate under off-design conditions. Therefore, the analysis of the performance under these operating conditions will be numerically and experimentally investigated in the future works.

NOMENCLATURE

Regular symbols

$\dot{E}$	exergy rate	(kW)
$\dot{E}_D$	exergy destruction rate	(kW)
$\dot{m}$	mass flow rate	(kg/s)
$\dot{Q}$	heat flow	(kW)
T	temperature	(ºC)
W	electrical energy	(kJ)
$\dot{W}$	electrical power	(kW)
e	specific exergy	(kJ/kg)
s	specific entropy	(kJ/kgK)
t	time	(h)

Subscripts

CH	charge
DIS	discharge
EX	exergy
RT	roundtrip
a	ambient
c	component boundary
i	inlet
o	outlet

Abbreviations

COMP	Compressor
COND	Condenser
COP	Coefficient of Performance
EEX	Electronic Expansion Valve
EVAP	Evaporator
GWP	Global Warming Potential
HP	Heat Pump
IHX	Internal Heat Exchanger
ODP	Ozone Depletion Potential
ORC	Organic Rankine Cycle
PHX	Pre-heater
PTES	Pumped Thermal Energy Storage
PV	Photovoltaic
SC	Sub-cooler
TES	Thermal Energy Storage
TFC	Triangular Flash Cycle
TURB	Turbine

Greek symbols

η	efficiency	(-)

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This book contains the compilation of works contributed to the *7th International Seminar on Organic Rankine Cycle Power Systems* (ORC 2023), held in Seville between the 4th and 6th of September 2023. The event was hosted by Universidad de Sevilla on behalf of the Knowledge Centre on Organic Rankine Cycle Technology (KCORC), incorporated in The Netherlands.

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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