David Tomás Sánchez Martínez
Lourdes García Rodríguez
(coordinadores)

Proceedings of the 7th International Seminar on ORC Power Systems

Editorial Universidad de Sevilla

Proceedings of the 7th International Seminar on

ORC
Power Systems

David Tomás Sánchez Martínez
Lourdes García Rodríguez
(coordinadores)

Proceedings of the 7th International Seminar on

ORC Power Systems

(ORC2023)

4th to 6th
SEPTEMBER
SEVILLE 2023

EDITORIAL UNIVERSIDAD DE SEVILLA

Sevilla, 2024

Colección Actas
Núm.: 91

Review chairs of the 7th International Seminar on ORC Power Systems

Prof. Christoph Wieland (University of Duisburg-Essen) - *Review Chair*
Prof. Sotirios Karellas (National Technical University of Athens) - *Review Co-Chair*
Prof. Teemu Turunen-Saaresti (LUT University) - *Review Co-Chair*
Dr. Diego-César Alarcón-Padilla (Plataforma Solar de Almería - CIEMAT) - *Review Co-Chair*
Dr. Francesco Crespi (Universidad de Sevilla) - *Review Co-Chair*

ISBN 978-84-472-2745-7
DOI: https://dx.doi.org/10.12795/9788447227457

Diseño de cubierta y maquetación: los autores

INDEX OF CONTRIBUTIONS

DOI: https://dx.doi.org/10.12795/9788447227457_72

REVERSIBLE HEAT PUMP-ORC PILOT PLANT – EXPERIMENTAL RESULTS AND FLUID CHARGE OPTIMIZATION

Maximilian Weitzer*, Sebastian Kolb, Dominik Müller, Jürgen Karl

Chair of Energy Process Engineering
Friedrich-Alexander-Universität Erlangen-Nürnberg (FAU)
Fürther Straße 244f, 90429 Nürnberg, Germany

*corresponding author: maximilian.weitzer@fau.de

ABSTRACT

The reversible operation of the heat pump process and the organic Rankine cycle as a Carnot battery has gained attraction of the scientific community in recent years. Despite numerous publications investigating the potential of such systems as a new energy storage technology, experimental results are scarce. The present work presents the outcomes on process level of a first experimental characterization of a reversible heat pump – organic Rankine cycle (HP-ORC) pilot plant designed for a maximum charging capacity of 15 kW_{el} in heat pump mode and discharging capacity of 9 kW_{el} in ORC mode. The experimental campaign comprised 150 stationary operating points. The plant yielded a maximum COP of 9.5 and a maximum ORC net efficiency of 4.4% for the defined boundary conditions of this experimental campaign. While the heat pump favors low temperature lifts for a high COP, high temperature gradients are favorable for ORC operation. Moreover, limitations on component and system level were determined. The challenging aspect of a proper fluid management of such reversible systems was identified as a crucial factor. Finally, the impact of fluid charge on ORC operation was experimentally evaluated and revealed significant optimization potential by means of an adequate fluid management.

1 INTRODUCTION

Balancing the fluctuating production of renewable energies with the demand is a key challenge in today's energy systems and requires large energy storage capacities. Carnot batteries are an emerging energy storage technology that potentially provides large capacities for several days up to few weeks. Surplus electrical energy charges a thermal energy storage, preferably by means of a heat pump process (HP) together with low-grade waste heat (Figure 1). For discharging, the thermal energy is transferred back to electrical energy by a power cycle, e.g. an organic Rankine cycle (ORC).

Carnot batteries in terms of heat pump – ORC systems rely on technically mature components (Vecchi et al., 2022). Moreover, such system allow a straightforward heat source integration and the storage capacity is scalable and low-cost as simple hot water storages can be used for typical operating temperatures up to 200°C (Frate et al., 2020; Weitzer et al., 2022b). Although the primary purpose of Carnot batteries is electrical energy storage (Dumont et al., 2020), several authors proposed heat pump – ORC systems as cogeneration units, e.g. in geothermal applications (Kaufmann et al., 2021; Weitzer et al., 2022a) or in building energy systems (Dumont et al., 2015; Palomba et al., 2021). The idea of reversible HP-ORC systems was initially proposed for building energy systems in combination with a solar roof and geothermal heat (Quoilin, Sylvain and Lemort, 2013). Although numerous simulative studies investigated different configurations of this concept, only few prototypes have been realized so far. Recently, a review paper comprehensively summarized current commercial developments in the field of Carnot batteries where most of the projects are in concept stage or pilot operation while few commercial installations exist (Novotny et al., 2022).

Combining a heat pump process and an ORC in a reversible configuration significantly reduces the machinery costs as both processes share major components. However, the design of a reversible configuration is not trivial and must pay particular attention to an appropriate choice of components, lubrication concept and fluid selection amongst others (Eppinger et al., 2021; Steger et al., 2020).

The present work presents the findings of a first experimental characterization of a reversible heat pump – ORC pilot plant at FAU Erlangen-Nürnberg. Additionally, the challenging aspect of fluid management in reversible systems is presented in terms of experimentally evaluating the impact of fluid charge on system operation. As experimental results of such systems are still scarce in literature, the outcomes of this work bridge the gap between simulative modelling and experimental validation of reversible heat pump – ORC systems.

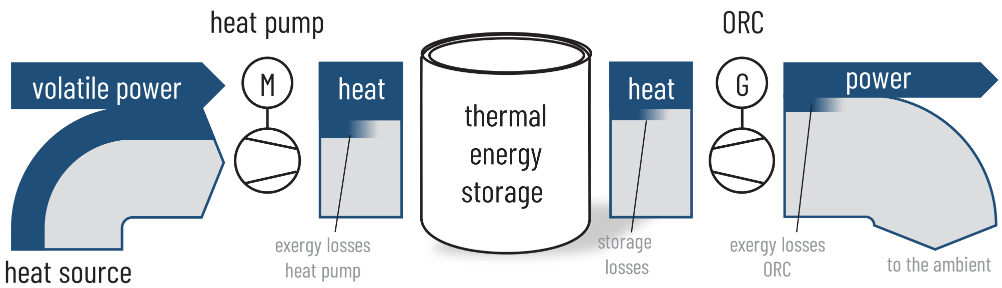

Figure 1: Working principle of a Carnot battery system based on heat pump and ORC.

2 EXPERIMENTAL SETUP AND METHODOLOGY

2.1 Carnot battery pilot plant

The detailed design process as well as the proof of concept and first launch of the reversible heat pump – ORC pilot plant were presented in previous studies (Steger et al., 2020, 2021). The prototype is designed for up to 15 kW_{el} in HP mode and up to 9 kW_{el} in ORC mode with two hot water storages with a storage capacity of 270 kWh_{th}. The pilot plant includes a screw machine that can be operated reversibly and an additional piston compressor that is not investigated in the present study. As shown in the simplified P&ID in Figure 2, the pilot plant is a fully reversible configuration, thus heat pump and ORC share all major components and pipes, beside pump and throttling valve. The brazed plate heat exchangers HEX 1 (state points 3↔4) and HEX 3 (1↔6) serve as condenser and evaporator in heat pump mode and vice versa in ORC mode. Another heat exchanger HEX 2 recuperates heat in both modes (2↔3 and 5↔6). A reversible screw machine is utilized as compressor and expander (4↔5), respectively. In both modes, a reversible oil separator at the high pressure vapor side extracts lubricant oil from the cycle and injects it into the screw machine. Moreover, a liquid receiver is included before the throttling valve (pneumatic needle valve) (2→1). A centrifugal pump circulates the working fluid in the ORC (1→2). Pneumatic valves allow switching between heat pump and ORC mode. The refrigerant R1233zd(E) is used in the experiments. Auxiliary water cycles allow heating or cooling through the heat exchangers, depending on the operating mode. Basic PID control loops for the auxiliary cycles are set up in the programmable logic controller (PLC) to ensure comparable heating and cooling boundary conditions. Furthermore, the PLC logs the data of temperature, pressure and flow sensors in an interval of 2 seconds.

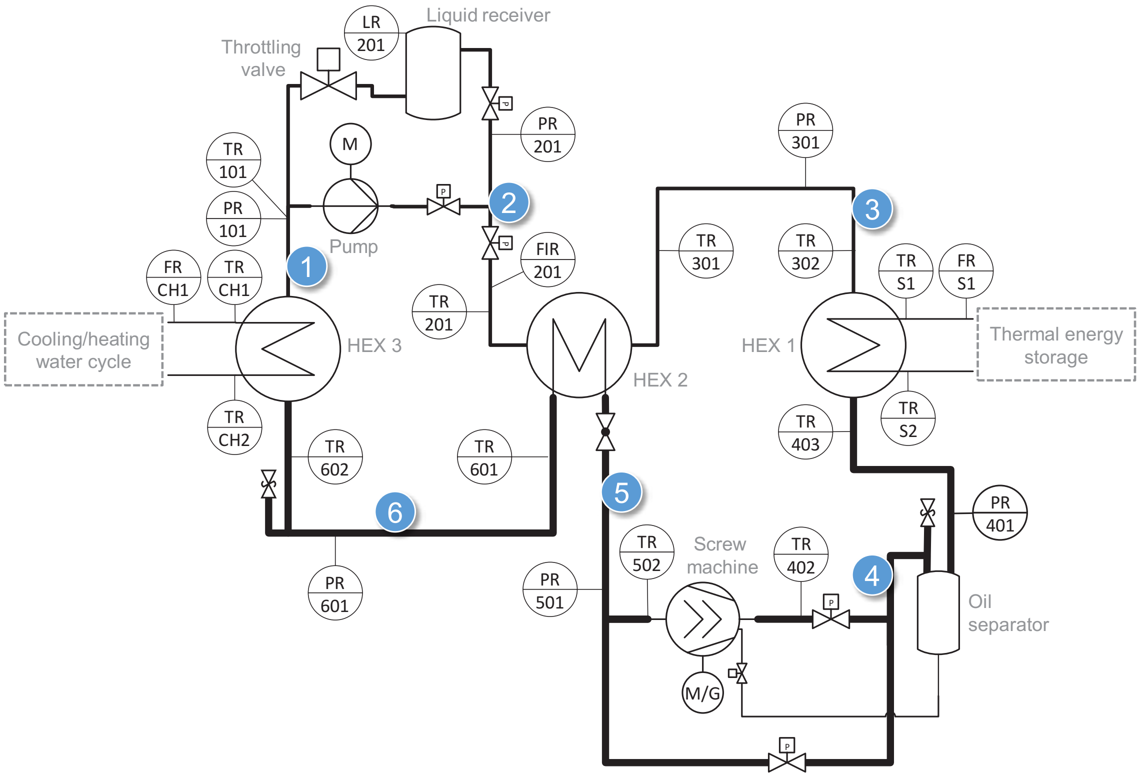

Figure 2: Simplified P&ID of the reversible HP-ORC pilot plant.

2.2 Methodology 432

In this experimental campaign, 150 stationary operating points were recorded in total, 92 in ORC mode and 58 in heat pump mode. Defined stationary criteria that compare the current 30-second mean values with those from 15 minutes ago ensure equal stationary conditions. The maximum deviation of these mean values is defined at 0.5 K for temperatures and 2% for pressures and flows. 15-minute mean values of the measured data were calculated at stationary conditions for all operating points. An auxiliary water cycle substituted the thermal energy storage.

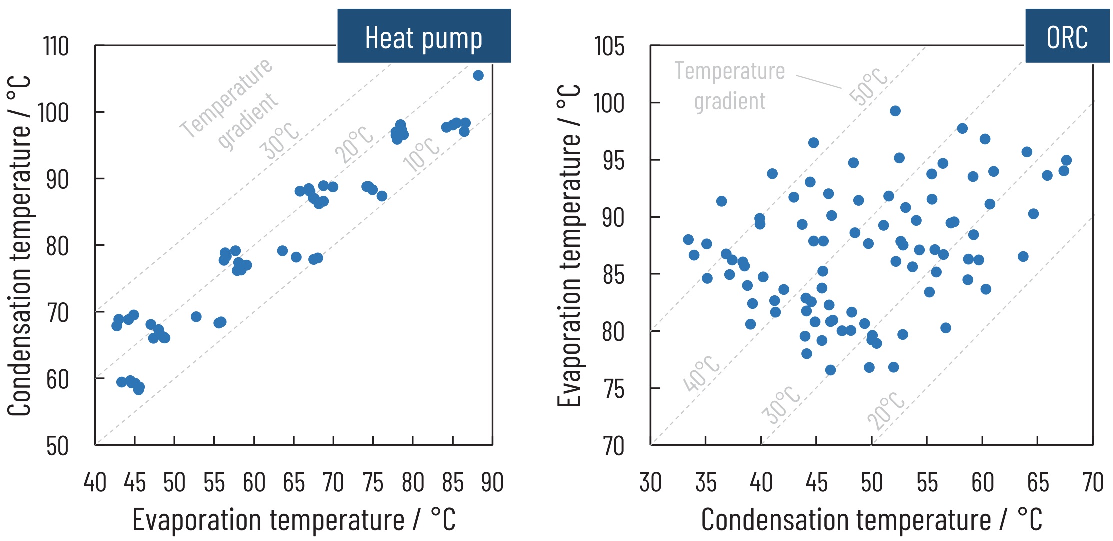

Figure 3: Phase change temperatures of all operating points for heat pump and ORC mode.

In heat pump mode, heat source temperatures from 50°C to 90°C were investigated in steps of 10 K, with upper storage temperatures between 60°C and 110°C and storage temperature glide of 10 K or 20 K. As the motor/generator unit was originally designed for ORC mode, the maximum motor torque limited the temperature lift of the heat pump process which barely exceeds 25 K (Figure 3). Thus, in case of a required high temperature lift, several consecutive heat pump runs might be necessary to reach a certain storage temperature. Moreover, the maximum temperature of the water cooling cycle did not allow exceeding a storage temperature of 110°C. For each heat source temperature, the compressor speed was increased stepwise until the maximum torque was reached. The opening of the throttling valve was adjusted to ensure slightly superheated conditions at the compressor inlet.

A similar methodology was applied for the ORC mode. As illustrated in Figure 3, the ORC reached a broader range of operation compared to the heat pump. Hot storage water with a temperature between 100°C and 120°C and a storage temperature glide of 10 K or 20 K supplied heat to the evaporator. For each heat source temperature, pump speed and expander speed were varied within the component and system limits. Table 1 summarizes the main set and controlled parameters of the experiments.

Table 1: Ranges of set and controlled parameters of the experiments.

HP parameter	Operating range	ORC parameter	Operating range
Heat source temperature	50…90°C	Upper storage temperature	100…120°C
Storage temperature glide	10…20 K	Storage temperature glide	10…20 K
Heat source volume flow	73 l/min	Storage volume flow	20…82 l/min
Upper storage temperature	60…110°C	Cooling water temperature	15…25°C
Storage volume flow	6…42 l/min	Cooling water volume flow	44…51 l/min
Mass flow working fluid	0.05…0.36 kg/s	Mass flow working fluid	0.14…0.36 kg/s
Throttling valve opening	59…99 %	Pump speed	2000…2800 rpm
Compressor speed	600…1800 rpm	Expander speed	1000…3100 rpm

3 EXPERIMENTAL RESULTS

3.1 Heat pump mode

The main performance indicator to assess the cycle efficiency of a heat pump process is the coefficient of performance (COP) defined in equation 1.

$$COP = \frac{\dot{Q}_{storage}}{P_{el,compressor}} = \frac{\dot{m}_{S1} \cdot (h_{S1} - h_{S2})}{P_{el,compressor}} \tag{1}$$

Figure 4 shows the electrical power consumed by the compressor and the COP plotted against the temperature gradient. The temperature gradient describes the difference between condensation and evaporation temperature (see Figure 3). Generally, an increasing power consumption and decreasing COP were observed for an increasing rotational speed of the compressor and a corresponding higher temperature gradient. As the motor/generator was originally selected for ORC operation, the maximum motor torque limited the rotational speed and power of the compressor. Due to that limitation, the temperature gradient did not exceed 25 K. Two sets of experiments were performed. For the upper left group of operating points, the storage water temperature was 10°C higher than the heat source temperature, thus the water temperature lift (i.e. external temperature lift) was around 10 K. These conditions resulted in a COP between 4.9 and 9.5. For the group of operating points at the lower right side, the storage temperature was 20°C above the heat source temperature, leading to a COP between 1.4 and 6.4. Thus, an increasing water temperature lift generally yields lower COP values. However, the COP also depends on the absolute temperature level and the rotational speed leading to scattered values within the two groups of operating points.

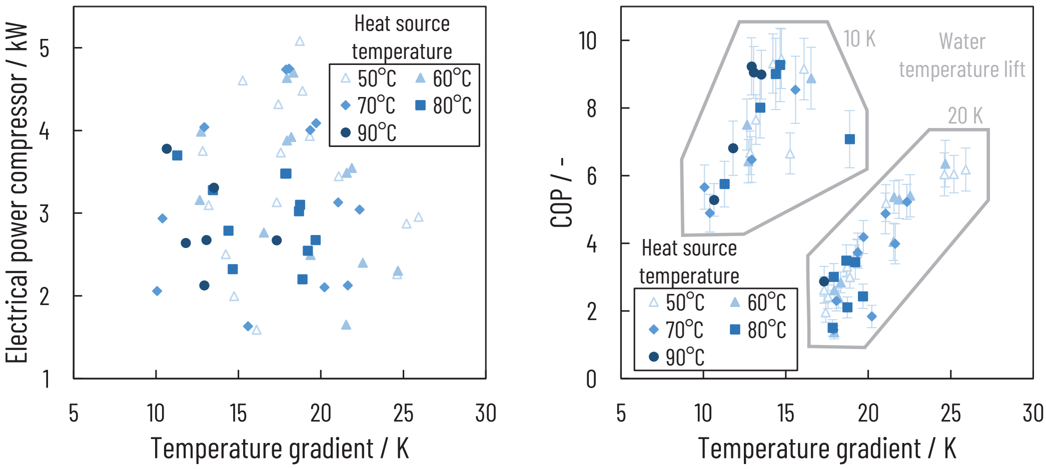

Figure 4: Electrical power consumption of the compressor and COP plotted against the temperature gradient for all heat pump operating points.

Table 2 summarizes the resulting process conditions for heat pump mode. An exemplary T-s diagram with a heat source temperature of 80°C is illustrated in Figure 5. It shows that phase change in condenser and evaporator were often not finished at the heat exchanger outlet. Thus, phase change partly occurred in the recuperator. Experimental tests with another compressor and motor are envisaged to overcome the limited temperature gradient of the current setup.

Table 2: Range of process conditions during the experimental campaign.

434

HP parameter	Operating range	ORC parameter	Operating range
Storage thermal power	4…31 kW	Storage thermal power	24…67 kW
El. power compressor	1.6…5.1 kW	El. power expander	0.3…3.7 kW
Condensation pressure	58…106°C	Evaporation pressure	6.0…10.3 bar
Evaporation pressure	43…88°C	Condensation pressure	1.7…4.8 bar
Max. COP	9.5	Max. net efficiency	4.4 %

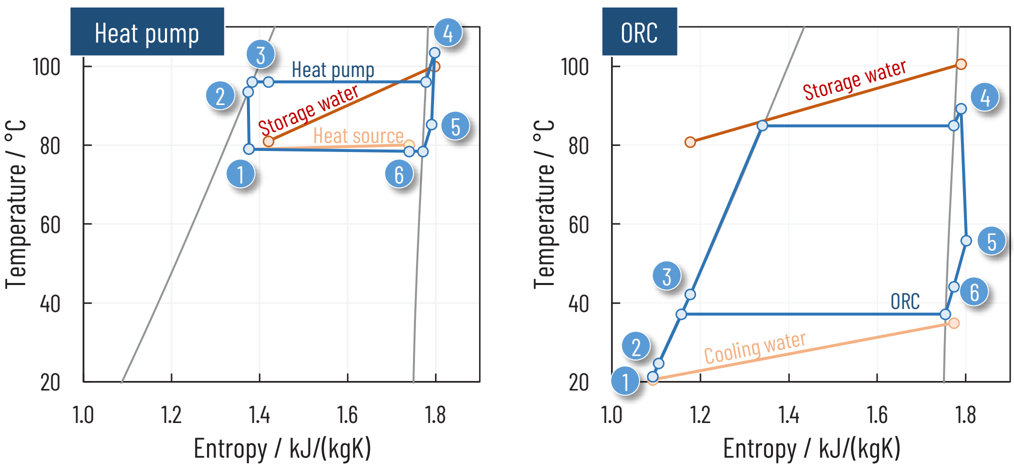

Figure 5: T-s diagram of heat pump and ORC process for upper and lower storage temperature of 100°C and 80°C respectively (state point numbers refer to the P&ID in Figure 2).

3.2 ORC mode

The global efficiency of the ORC is evaluated in terms of the net electrical efficiency as described in equation 2.

$$\eta_{el,net,orc} = \frac{P_{el,net}}{\dot{Q}_{storage}} = \frac{P_{el,expander} - P_{el,pump}}{\dot{m}_{S1} \cdot (h_{S2} - h_{S1})} \tag{2}$$

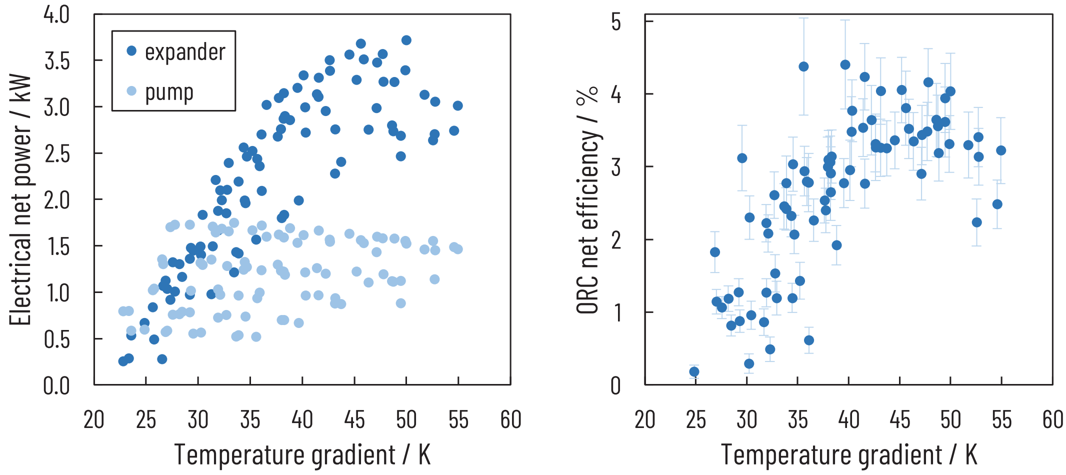

Figure 6: Electrical power of expander and pump and ORC net efficiency plotted against the temperature gradient for all ORC operating points.

Figure 6 shows the electrical power generated by the expander and consumed by the pump as well as the ORC net efficiency against the temperature gradient. For some operating points with low temperature gradients, the pump consumption exceeds the generated power of the expander. This results from a comparably low efficiency of the axial centrifugal pump and could be solved by replacing it with a volumetric pump. The generated power increases with increasing temperature lift. In contrast to the heat pump, a high temperature gradient is beneficial for the ORC efficiency. 435

Altogether, the ORC yielded a maximum net efficiency of 4.4%. Again, Table 2 summarizes the process conditions and Figure 5 exemplarily illustrates a T-s plot at upper and lower storage temperatures of 100°C and 80°C. A comparatively high degree of subcooling of around 20 K across all operating points indicates that the fluid charge of the ORC was too high. Further potential is thus expected from a fluid charge optimization. The generated power did not reach design conditions due to a limited pump performance, limited cooling capacity of the cooling water grid and refrigerant excess in the condenser. The impact of the latter aspect is investigated in the following section.

4 FLUID CHARGE OPTIMIZATION

4.1 Concepts of reversible fluid management

A challenging aspect in the design of reversible HP-ORC systems is an adequate fluid management in order to supply both processes with the required amount of working fluid at all operating conditions. Excess fluid typically accumulates in the condenser, increases the condensation pressure and reduces the power output in ORCs while too low fluid charges lead to pump cavitation (Dickes et al., 2020). A recent experimental study varied the fluid charge of a 1 kW_{el} reversible HP-ORC test rig and concluded that the fluid charge has a significant impact on overall efficiency, reversible machine performance, system stability and obtainable operation points (Tassenoy et al., 2022). A common measure is the integration of a liquid receiver before the throttling valve in heat pumps and before the pump in ORCs. However, for reversible systems there are several concepts for the integration of liquid receivers (Figure 7). In the first option (a), the liquid receiver is bypassed in both processes. Thus, the fluid charge can only be adapted between operating times or the liquid valve can permanently remain open to the process. The fluid charge and lower system pressure can be actively controlled by adding a membrane charged with inert gas or a piston in the receiver vessel. In option (b) at least one receiver is directly integrated in the process. Receiver A corresponds to the current setup of the pilot plant. If only one receiver is integrated, the receiver is bypassed in the other operating mode. A reversible utilization of the receiver (c) allows integrating the same receiver in both modes.

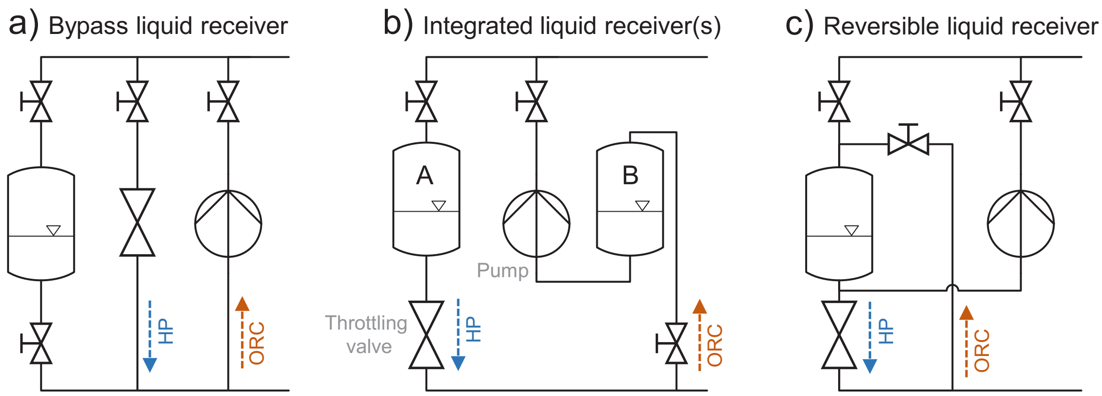

Figure 7: Different configurations of liquid receiver integration in reversible HP- ORC systems.

4.2 Experimental fluid charge evaluation

In the current experimental setup (Figure 2), a liquid receiver is integrated in the heat pump process before the throttling valve. During operation, the integrated liquid receiver stores surplus working fluid and ensures that the process is supplied with the suitable amount of fluid. This principle works as long as the vessel is neither flooded nor emptied. Therefore, a careful selection of proper liquid receiver volume is crucial. Figure 8 shows the balancing role of the liquid receiver during an experimental day with different compressor speeds and throttling valve openings.

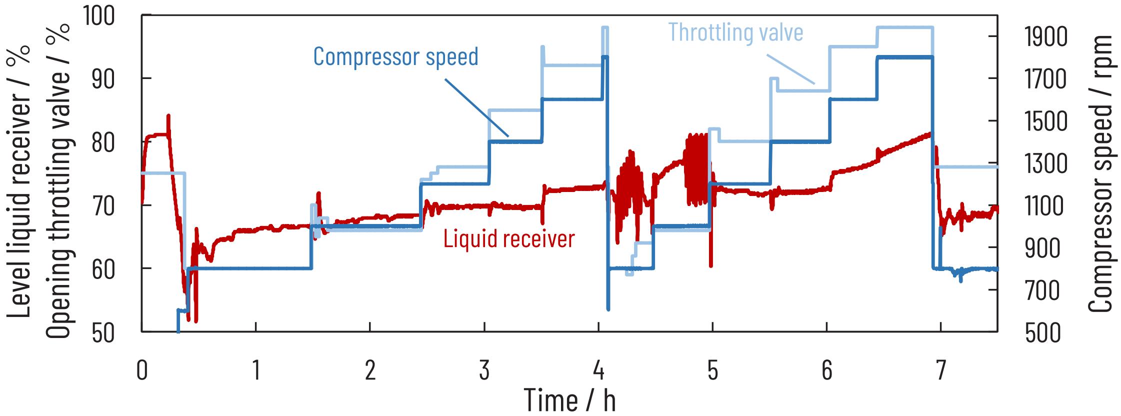

Figure 8: Self-regulating fluid charge due to the liquid receiver in heat pump mode.

In ORC mode, the liquid receiver is bypassed and thus cannot balance the fluid inventory in the cycle. In order to determine the impact of the fluid charge on ORC operation, a separate experimental campaign was started. An excerpt of the results is shown in Figure 9. On the left side, the T-s diagram for eight different ORC fluid charges between 16.3 kg and 22.7 kg at constant operating conditions (mass flow 0.25 kg/s, evaporation temperature 90°C) is given. The results show that the fluid charge mainly affects the condensation temperature. Surplus liquid working fluid accumulates in the condenser and occupies the heat exchange area. This results in high condensation temperatures and a high degree of subcooling. Due to the lower pressure gradient, less power can be generated in the expander (Figure 9 right). The lowest condensation temperature and highest power generation was observed for the lowest fluid charge. However, even lower fluid charges were not possible due to pump cavitation that prevented a proper ORC operation.

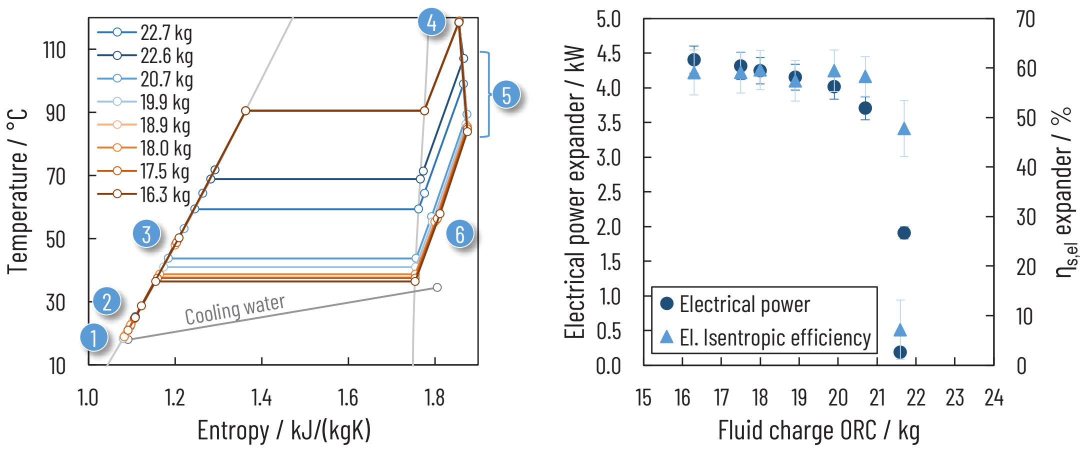

Figure 9: T-s diagram (left) and performance indicators (right) for various ORC fluid charges.

5 CONCLUSION AND OUTLOOK

The first experimental characterization of a reversible heat pump – ORC pilot plant comprised 150 stationary operating points. The pilot plant reached a maximum COP of 9.5 and a maximum ORC net efficiency of 4.4% for the defined boundary conditions of this experimental campaign. While the heat pump favors low temperature lifts for a high COP, high temperature gradients are favorable for ORC operation. The maximal motor torque mainly restricted the operating range of the heat pump. Therefore, experiments with another compressor and motor are envisaged. In ORC operation, the comparatively high consumption and maximum capacity of the centrifugal pump limited the net efficiency. Despite those limitations, the reversible operation of the components worked well and disclosed optimization potential on component and system level. The fluid management of reversible systems was identified as a crucial optimization lever to improve efficiency. Different configurations to optimize the fluid charge for both operating modes were presented. In this context, the liquid receiver in the heat pump process proved its function to balance the working fluid charge. As no liquid receiver is included in the ORC, the ORC efficiency is highly sensitive to the fluid charge in this setup. 437

Upcoming work will further analyze the experimental results on component and system level in order to identify further optimization potential. Moreover, the optimal fluid charge and optimal fluid properties will be studied in detail and taken into account for upcoming experiments, including different working fluids and fluid mixtures. The results will feed a simulation model of the plant that will also consider fluid charge sensitivity.

NOMENCLATURE

COP	Coefficient of performance	-	
h	Specific enthalpy	kJ/kg	
HP	Heat pump		
m	mass	kg	
ORC	Organic Rankine Cycle		
P	Electrial power	kW	
Q	Thermal energy	kJ	
s	Specific entropy	kJ/(kg K)	
t	Temperature	°C	
η	Efficiency	%	

Subscript

el	electric
th	thermal

REFERENCES

Dickes, R., Dumont, O., and Lemort, V. 2020. Experimental assessment of the fluid charge distribution in an organic Rankine cycle (ORC) power system, *Applied Thermal Engineering*, vol. 179, 115689

Dumont, O., Frate, G. F., Pillai, A., Lecompte, S., De paepe, M., and Lemort, V. 2020. Carnot battery technology: A state-of-the-art review, *Journal of Energy Storage*, vol. 32, no. July

Dumont, O., Quoilin, S., and Lemort, V. 2015. Experimental investigation of a reversible heat pump/organic Rankine cycle unit designed to be coupled with a passive house to get a Net Zero Energy Building, *International Journal of Refrigeration*, vol. 54, 190–203

Eppinger, B., Steger, D., Regensburger, C., Karl, J., Schlücker, E., and Will, S. 2021. Carnot battery : Simulation and design of a reversible heat pump-organic Rankine cycle pilot plant, *Applied Energy*, vol. 288, no. February, 116650

Frate, G. F., Ferrari, L., and Desideri, U. 2020. Rankine carnot batteries with the integration of thermal energy sources: A review, *Energies*, vol. 13, no. 18

Kaufmann, F., Schifflechner, C., Dawo, F., Wieland, C., and Spliethoff, H. 2021. Evaluation of Reversible Screw Compressors with an Adaptable Built-In Volume Ratio in Reversible Heat Pumps, *6th International Seminar on ORC power systems*, 1–10

Novotny, V., Basta, V., Smola, P., and Spale, J. 2022. Review of Carnot Battery Technology Commercial Development, *Energies*, vol. 15, no. 2

Palomba, V., Borri, E., Charalampidis, A., Frazzica, A., Karellas, S., and Cabeza, L. F. 2021. An Innovative Solar-Biomass Energy System to Increase the Share of Renewables in Office Buildings, *Energies*, vol. 14, no. 4, 914

Quoilin, Sylvain, O. D. and Lemort, V. 2013. Design, Modeling and Performance Optimisation of a Reversible Hp-Orc Prototype, in *2nd International Seminar on ORC Power Systems*

Steger, D., Karl, J., and Schlücker, E. 2021. Launch and First Experimental Results of a Reversible Heat Pump-Orc Pilot Plant As Carnot Battery, *6th International Seminar on ORC Power Systems*, no. 2018, 1–9

Steger, D., Regensburger, C., Eppinger, B., Will, S., Karl, J., and Schlücker, E. 2020. Design aspects of a reversible heat pump - Organic rankine cycle pilot plant for energy storage, *Energy*, vol. 208, 118216

Tassenoy, R., Dumont, O., Lemort, V., De Paepe, M., and Lecompte, S. 2022. Experimental investigation of a thermally integrated carnot battery using a reversible heat pump/ organic rankine cycle: Influence of system charge on performance of the reversible scroll compressor/expander and global performance, *19th International Refrigeration and Air Conditioning Conference Proceedings*, Advance Access published 2022

Vecchi, A., Knobloch, K., Liang, T., Kildahl, H., Sciacovelli, A., Engelbrecht, K., Li, Y., and Ding, Y. 2022. Carnot Battery development: A review on system performance, applications and commercial state-of-the-art, *Journal of Energy Storage*, vol. 55, no. PD, 105782

Weitzer, M., Müller, D., and Karl, J. 2022a. Two-phase expansion processes in heat pump – ORC systems (Carnot batteries) with volumetric machines for enhanced off-design efficiency, *Renewable Energy*, vol. 199, no. February, 720–32

Weitzer, M., Müller, D., Steger, D., Charalampidis, A., Karellas, S., and Karl, J. 2022b. Organic flash cycles in Rankine-based Carnot batteries with large storage temperature spreads, *Energy Conversion and Management*, vol. 255, no. December 2021, 115323

438

ACKNOWLEDGEMENT

This project has received funding from the European Union's Horizon 2020 research and innovation program under grant agreement no. 814945 – SolBio-Rev.

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.

KCORC
KNOWLEDGE CENTER
ORGANIC RANKINE CYCLE

eus EDITORIAL
UNIVERSIDAD DE SEVILLA