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# Start-up behaviour and first operation of a fully reversible Heat Pump–ORC system using fixed-speed scroll machines

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**Abstract**

Europe's energy transition requires an increase in both renewable electricity generation and renewable heating and cooling supply. Achieving these goals calls for flexible energy system capable of converting, storing, and dispatching energy across multiple vectors. Reversible Organic Rankine Cycle/Heat Pump (rev. ORC/HP) systems offer such versatility, as they can switch between power production and heating modes while sharing key components. Although recent research has examined the steady-state behaviour of rev. ORC/HP systems, their transient operation – particularly during start-up – remains insufficiently understood. This work experimentally investigates the start-up dynamics of a fully rev. ORC/HP system equipped with fixed-speed reversible volumetric machines. The paper examines different electrical grid-connection strategies for reversible volumetric machines, highlighting their respective operational constraints, benefits, and limitations. The start-up procedures for both HP and ORC modes are then analysed in detail, with a focus on how fixed-speed operation shapes transient behaviour. Finally, the study proposed practical measures to mitigate the challenges associated with fixed-speed reversible machines.


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*Keywords: Reversible Heat Pump/ORC; Experimental; Start-up dynamics; Carnot battery*

## 1. Introduction

Europe's ongoing energy transition toward clean energy (driven by climate goals and the need for energy independence) demands a simultaneous increase in the share of renewable electricity and renewable heating and cooling supply. While the share of renewable electricity in Europe reached about 44% in 2023 [1], the share of renewables in heating and cooling remains lower, at 26.2% [2]. Increasing both shares of renewable energy is essential to achieve the goals sets by the European Union. Both electricity generation and thermal energy demand exhibit strong temporal variability as renewable power production fluctuates with weather conditions, while heating and cooling needs change according to daily and seasonal cycles. In this context, the ability to convert and store energy across multiple forms becomes crucial. The energy sector must move beyond standalone devices that operate in a single direction (e.g., electricity-to-heat or heat-to-electricity). Instead, versatile and hybrid machines are required (systems that can produce, store, and manage multiple energy vectors according to time-varying, sector-coupled demands). In this context, reversible Organic Rankine Cycle/Heat Pump (rev. ORC/HP) systems offer a promising solution. Rev. ORC/HP systems find potential applications across a wide range of sectors, including industrial waste heat recovery, district heating and cooling networks, geothermal systems, mobile and building-scale applications, and pumped thermal energy storage (PTES) [3]. Their ability to adapt operation modes according to energy availability and market conditions also opens opportunities for economically profitable deployment in future multi-vector energy systems.

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HPs and ORCs share many key components (heat exchangers and volumetric expanders/compressors) which makes their integration into a single reversible system technically feasible. Different levels of reversibility can be achieved, ranging from fully reversible systems, in which most components (heat exchangers, volumetric machines, and piping) are shared between the two operating modes, to partially reversible systems that only reuse selected components/piping. While sharing components reduces cost and system complexity, it often comes at the expense of less optimal performance or limited adaptability to specific applications and operating conditions [4]. Fully reversible systems can operate in two configurations: conventional cycles, where the working fluid maintains its flow direction and each heat exchanger keeps its function, and inverted cycles, where the fluid flow reverses and the heat exchangers exchange roles [4]. In the inverted reversible configuration, used in this work, the condenser in heat-pump mode functions as the evaporator in ORC mode, and vice versa, while their pressure levels remain constant. This configuration is particularly suited to applications such as pumped thermal energy storage, where the hot and cold loops remain physically fixed across operating modes.
Experimental studies on the steady-state performances of rev. ORC/HP systems are emerging [5] [6] [7]. However, the start-up and dynamic behaviour of such systems remain poorly understood.
In this work, the start-up behaviour of a rev. ORC/HP system employing fixed-speed reversible scroll machines is investigated. Although fixed-speed operation simplifies the electrical connection to the gird and reduces component cost, it also imposes several operational constraints. The implications of these constraints are examined, and different grid-connection strategies for reversible machines are discussed with their respective advantages and limitations.
The start-up procedure of both modes (HP and ORC) is described with its influence of the transient behaviour and overall performance during stat-up. Particular attention is given to the challenges linked to the fixed-speed operation. The start-up dynamics of both modes are compared. Finally, possible measures to mitigate these issues and enable smoother start-up are presented.

## 2. Case study

The experimental test bench consists of a fully rev. ORC/HP system coupled with a thermocline storage tank, forming a PTES-like configuration. The installation is integrated with the District Heating Network (DHN) of the University of Liège and is designed to smooth both thermal and electrical peak demands. The prototype was developed based on the design methodology presented in [8].
Both operating modes are represented using different colours in the P&ID (Figure 1). In HP mode, the system extracts heat from the DHN, which serves as the heat source. In this configuration, HX1 functions as the condenser, while HX2 operates as the evaporator, and the thermal storage acts as the heat sink. The refrigerant pump and dry cooler are bypassed through a dedicated valve arrangement. In ORC mode, the system converts the stored thermal energy into electricity. The thermal storage then serves as the heat source, with HX1 acting as the evaporator and HX2 as the condenser. The condenser heat is rejected to the dry cooler, which acts as the heat sink.
Although the system is designed to use the DHN as the heat source in HP mode, the network was not in operation during the summer testing period. To enable preliminary experiments, an oil boiler was employed to simulate the DHN and provide a controllable heat source.
The test rig is designed to deliver nominal capacities of 80 $kW_{\mathrm{th}}$ in heat pump mode and 5.6 $kW_{\mathrm{e}}$ in ORC mode, operating with R1233zd(e) as the working fluid. These values are subject to experimental validation. The corresponding design round-trip efficiency, accounting for thermal integration within the system, is 56%. The system uses three scroll machines as volumetric machines, brazed-plate heat exchangers, and a peripheral pump for refrigerant circulation. Three scroll machines are operated in parallel, as no single commercially available machine with the required swept volume was available within the targeted cost range. This configuration enables capacity modulation by activating one or more machines and allows adaptation of the effective swept volume to the operating mode. In heat pump mode, a swept volume of approximately 242 $cm^3$ is required, which is achieved by operating two machines, while in ORC mode a swept volume of about 213 $cm^3$ is required under nominal conditions, corresponding to the operation of all three machines to ensure comparable refrigerant mass flow rates in both modes. The pump is installed at the lowest point of the test bench to ensure adequate net positive suction head (NPSH) and prevent cavitation. The main components of the system are summarized in Table 1.
The system is equipped with temperature, pressure, and mass flow sensors located at key positions in the cycle (see Table 2). As the installation was initially developed as a demonstration prototype and not a research prototype, the instrumentation is limited and does not allow for complete thermodynamic characterization. The test bench will therefore require further upgrades to meet academic research standards. Data acquisition and manual control are performed via a Siemens LOGO! platform, chosen for its simplicity and low cost.

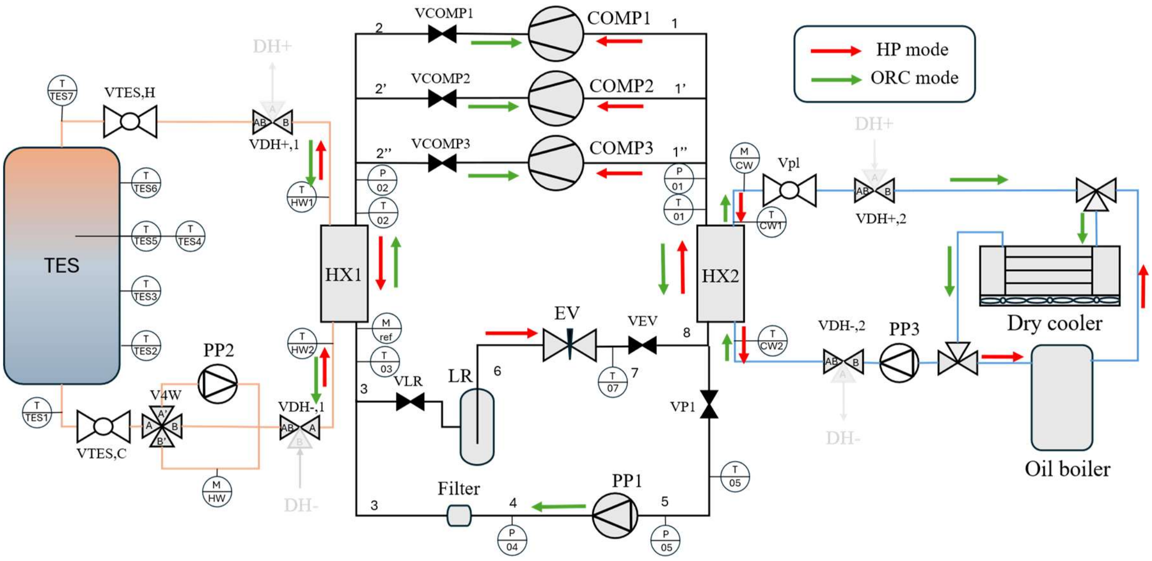



Figure 1. P&ID experimental set-up.

Table 1. Details of the main components

| Component | Model | Parameter | Value |
|---|---|---|---|
| Scroll machine | Sanden TRSA12 | Max shaft speed | 6000 RPM |
| | | Swept Volume (Compressor) | 121 $cm^3$ |
| | | Swept Volume (Expander) | 71,2 $cm^3$ |
| | | Asynchronous motor | 4 kW |
| Refrigerant pump | SAWA MP73-RKFE | Nominal shaft speed | 2900 RPM |
| | | NPSHr | 5 m |
| | | Asynchronous motor | 1,5 kW |
| High pressure heat exchanger (HX1) | SWEP P200THx140 | Type | Brazed plate |
| | | Nb of plates | 140 |
| | | Area | 17,8 $m^2$ |
| Low pressure heat exchanger (HX2) | SWEP P200THx140 | Type | Brazed plate |
| | | Nb of plates | 140 |
| | | Area | 17,8 $m^2$ |
| Expansion valve | Danfoss ICM 20-B | Type | Motorised control valve |
| Liquid receiver | OCSCOLD RLV | Volume | 10 L |

Table 2. Sensors information

| Property | Sensor Type | Range | Accuracy |
|---|---|---|---|
| Temperature | PT100 | - | ±0.3 °C + 0.5 % of MV |
| Pressure | Relative | 0-10 bar rel. | ±1.0% FS |
| Working mass flow rate | Coriolis | 10-550 g/s | ±0.5 % of MV (≥ 100 g/s) |
| Water mass flow rates | Volumetric counter DN40<br>Volumetric counter DN50 | 0.6-15 (max 30) $m^3/h$<br>1-25 (max 50) $m^3/h$ | - |

## 3. Connection of reversible volumetric machines to electrical grid

One of the main challenges in operating the rev. ORC/HP system lies in the fixed-speed configuration of the scroll machines. This choice was initially made for economic reasons, as it offered a simple and cost-effective solution for connecting the reversible volumetric machines to the electrical grid.
However, the grid connection of reversible volumetric machines requires careful consideration of the motor-generator type and the electrical interface employer. Several possible configurations exist, each presenting specific advantages, limitations, and implications for system dynamics and control.
In the following section, the different connection strategies are presented and discussed to justify the current fixed-speed configuration and to motivate the transition toward variable-speed operation in future work.

### *3.1. Motor/Generator configurations*

Two primary approaches exists for coupling the volumetric machines to electrical machines:

1. **Separate motor and generator:** One electrical machine drives the compressor during the compressor mode, while a second generator extracts work during the expansion mode. This configuration allows each machine to be optimized for its specific function, providing better efficiency and fewer issues with voltage or speed matching. This configuration is opted in [4] and [9].
2. **Single bidirectional motor:** A single electric machine operates in both directions, functioning as a motor in compression mode and as a generator in expansion mode. While this approach simplifies the electrical connection and reduces cost, the machine must be designed to handle both operating modes, which can limit efficiency compared to the dual-machine solution. This configuration is opted in [6] and [7] as well as in this work.

### *3.2. Choice of motor type*

Once the number of machines is determined, the selection between synchronous and asynchronous motors must be considered:

1. **Synchronous motors** offer precise speed control and can operate efficiently over a range of loads, but typically require more complex control electronics and can be more expensive. This is used in [7].
2. **Asynchronous (induction)** motors are robust, widely available, and inexpensive. They can directly connected to the grid, but their operating speed varies slightly under load due to slip, which may limit fine control of the scroll machine's mass flow rate and pressure ratio. Most of the rev. HP/ORC use this configuration.

In this work, an asynchronous motor is used.

### *3.3. Electrical interface*

Frequency inverters (VFDs) are typically installed between the grid and the electric machine to provide precise speed and torque control allowing better compressor/expander speed matching under varying operating conditions. Although variable-speed operation can improve part-load and off-design performance, the resulting gains in volumetric machines are often modest compared to the influence of external boundary conditions [10] [11]. Nevertheless, speed control remains particularly valuable during start-up, as discussed in Section 4.
To reduce prototype costs for potential future commercialization, the demonstrator omits the frequency inverter and instead employs a direct fixed-speed configuration, in which the 50 Hz electrical grid (Belgium) drives a bidirectional asynchronous motor connected via a pulley ratio to the scroll machine.
In this setup, the scroll machines operate at a nominal speed of 6000 RPM, achieved by coupling a standard 3000 RPM asynchronous motor to a 2:1 pulley transmission (the actual speed varies slightly due to slip inherent to induction motors). High-speed motors capable of directly achieving 6000 RPM were excluded due to their high cost and limited availability. The high rotational speed was selected primarily due to the small swept volume of the scroll machines. Operating at this speed allows the system to achieve a refrigerant mass flow rate close to the design point.
Some losses arise from the pulley transmission, but some losses would also occur if a frequence inverter were used, as inverters introduce their own electrical conversion losses.

### *3.4. Challenges of fixed-speed operation*

Operating scroll machines at a fixed rotational speed, introduces several challenges:

- Limited dynamic control: Without an inverter, the mass flow rate and pressure ratio cannot be adjusted in real-time.
- Parallel operation: To adjust system capacity without speed variation, multiple scroll units are connected in parallel. Units can be activated or deactivated depending on thermal or electrical load.

As explained in section 4, the biggest issue related to fixed speed operation is the start-up for the HP mode and the fact of working off-design in ORC operation.

## 4. Start-up dynamics

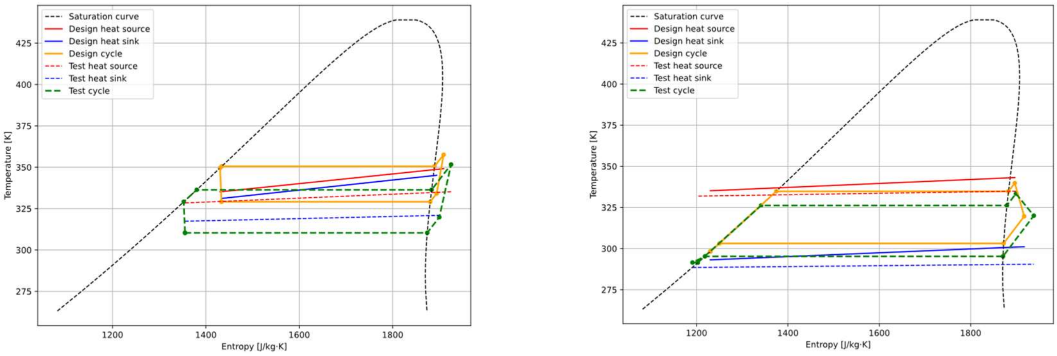
This section presents the start-up dynamics and procedure, as well as the issues associated with high fixed-speed operation.

### *4.1. Boundary conditions*

Table 3 summarizes both the design conditions and the experimental test conditions, which are notably off-design, particularly in terms of the refrigerant mass flow rate. The low mass flow rate observed during testing is primarily attributed to the lower temperature levels compared to the design point. As a result, the heat transfer rates in the heat exchangers were significantly reduced, leading to oversized heat exchangers and thus very small pinch-point temperature differences (between 0,3 and 1,3 K). The deviation from design conditions can be explained by two main factors: (1) The DHN was not operational during the experiments and the oil boiler used as substitute could not achieve the same supply temperatures. (2) The thermal storage tank was not insulated, making it difficult to maintain high temperatures for the ORC operation.
Operating with fixed-speed volumetric machines under such off-design conditions is challenging, especially during start-up, as will be shown in the following sections.
Table 3 compares the design conditions of each mode to the test boundary conditions and Figure 2, shows the T-s diagrams of the design and test conditions of both modes, highlighting the difference.

Table 3. Design/Test boundary conditions

| Variable | HP mode Design | HP mode Test | ORC mode Design | ORC mode Test |
|---|---|---|---|---|
| **External variable** | | | | |
| Hot water temperature | 70 °C | 55°C | 76 °C | 62°C |
| Cold water temperature | 62°C | 48°C | 15 °C | 16°C |
| **Internal variable** | | | | |
| Refrigerant mass flow rate | 0.422 kg/s | 0.213 kg/s | 0.499 kg/s | 0.075 kg/s |
| Cold water mass flow rate | 3.2 kg/s | 2.25 kg/s | 2.6 kg/s | 1.8 kg/s |
| Hot water mass flow rate | 1.4 kg/s | 1.4 kg/s | 3 kg/s | 1.4 kg/s |
| Number of compressor/expander | 2 | 2 | 3 | 1 |
| **Powers** | | | | |
| Electrical power | 10 kW | 7.2 kW | 5.6 kW | - |
| Condenser power | 82.5 kW | 47.2 kW | 94.6 kW | 17.1 kW |
| Evaporator power | 67.3 kW | 41.6 kW | 100 kW | 16.8 kW |




a) HP mode

b) ORC mode

Figure 2. Ts diagram showing the design point and the test point for both HP and ORC modes.

### *4.2. Start-up / operation difficulties*

<u>Start-up HP:</u>

Because the system uses fixed-speed scroll compressors at high speed, the heat-pump start-up is particularly challenging.

During stand-by, liquid refrigerant accumulate inside the scrolls. Starting directly at full speed in the presence of liquid can (1) cause mechanical damage due to liquid slugging and (2) can lead to a large electrical consumption as the compressors try to expel the liquid.

To avoid these issues, the following start-up procedure is applied to ensure that only vapor enters the compressor at start-up:

1. **Preheat the evaporator:** Activate the cold-side water pump (PP3) to preheat the evaporator.
2. **Remove residual liquid from the compressors:** Run the refrigerant pump (PP1) in reverse to circulate refrigerant through the evaporator. This causes any remaining liquid in the compressors to evacuate and be replaced with vapor. Once the compressors rotate freely (visually confirmed), stop PP1 and close valve VP1 (see Figure 3).
3. **Ensure superheat:** Partially close valve EV (approximately 40% open) to create a pressure drop that ensures sufficient refrigerant superheating.
4. **Activate the hot-side water pump:** Turn on the hot-side water pump (PP2) **only after** VP1 and EV are closed. Starting PP2 prematurely can raise the condenser temperature relative to the evaporator, causing refrigerant to condense in the evaporator and refill the compressors with liquid.
5. **Start the compressors:** Once the above steps are completed, power the compressors.

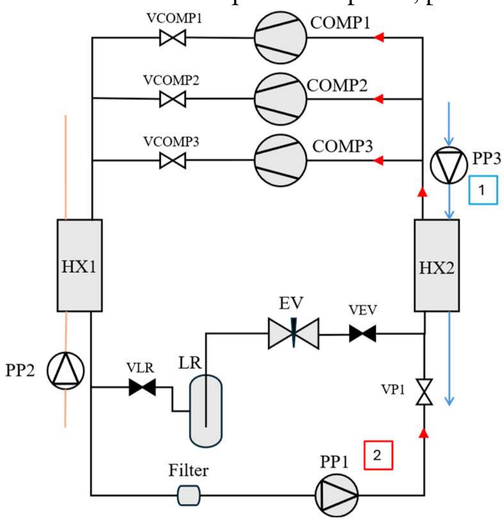



Figure 3. Two first steps of the start-up procedure in HP mode.

Figure 4 compares two start-up procedures. In the left plot, the start-up is performed only after using PP1 to remove the liquid from the compressors. This approach results in only a small transient current peak. In contrast, the right plot shows a start-up performed without using PP1 beforehand. In this case, a pronounced current peak appears at the beginning of the sequence. This behaviour must be considered when selecting the circuit breaker. Early in the experimental campaign, the circuit breaker repeatedly tripped due to the high current inrush caused by liquid remaining inside the compressor.

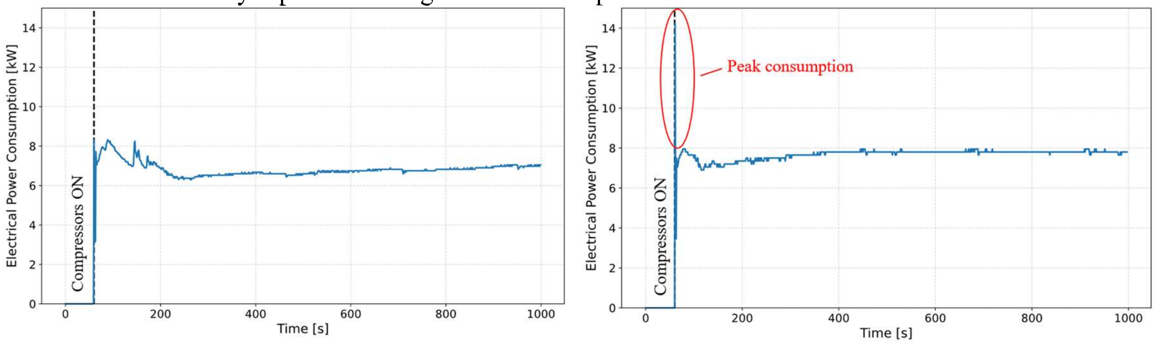



Figure 4. Comparison of start-up electrical consumption of compressors between start-up without residual liquid trapped inside compressors (left) or with residual liquid right).

Start-up ORC:

The start-up procedure of the ORC is generally smoother and more stable. In this configuration, both pumps of the hot source and cold sink can be started first, as the resulting pressure difference favours the intended flow direction. The expander begins rotating spontaneously due to the pressure difference, thereby removing any residual liquid from the scroll.

The start-up of the ORC follows the sequence below:

1. Activate the cold water sink and hot water source pumps (PP2 and PP3)
2. Start the refrigerant pump (PP1) (the refrigerant flow direction is the one shown in Figure 1)
3. Once the pressure ratio across the expander is considered sufficient (approximately 2), connect the expander to the grid.

The operational challenges associated with the expander differ from those encountered in HP mode. The main limitation lies in achieving the necessary rotational speed. Under the tested conditions (Table 3), the temperature difference - although close to the design point – was insufficient to establish a mass flow rate and pressure ratio capable of producing positive power output. Consequently, the expander operated in a motoring regime, consuming electrical power rather than generating it. This behaviour is illustrated in Figure 5, where the measured power exceeds the calculated isentropic power (given by $\dot{W}_{is} = \dot{m} \cdot \Delta h_{is}$).

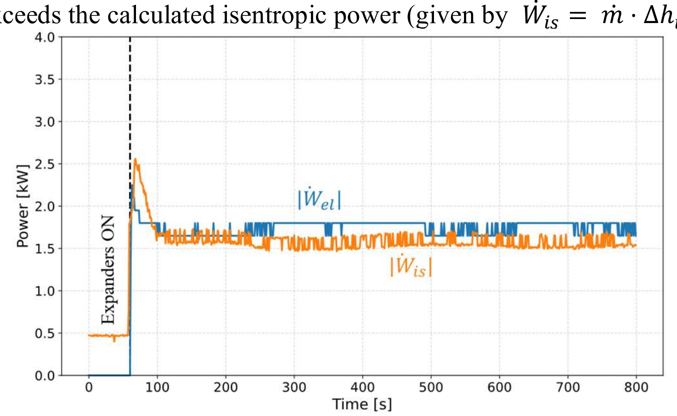



Figure 5. Comparison of start-up electrical production/consumption of expander vs isentropic work.

In Figure 5, the isentropic power fluctuates. This instability primarily results from the low mass flow rate, which is significantly below the design-point vale. Under these conditions, the Coriolis flow meter operates outside its calibrated measurement range, leading to unstable readings.

### *4.3. Start-up dynamics*

This section presents a comparison of the start-up dynamics for both modes. Steady-state operation is considered to be achieved when the measured variable reaches and stays in 95% of its steady-state value.

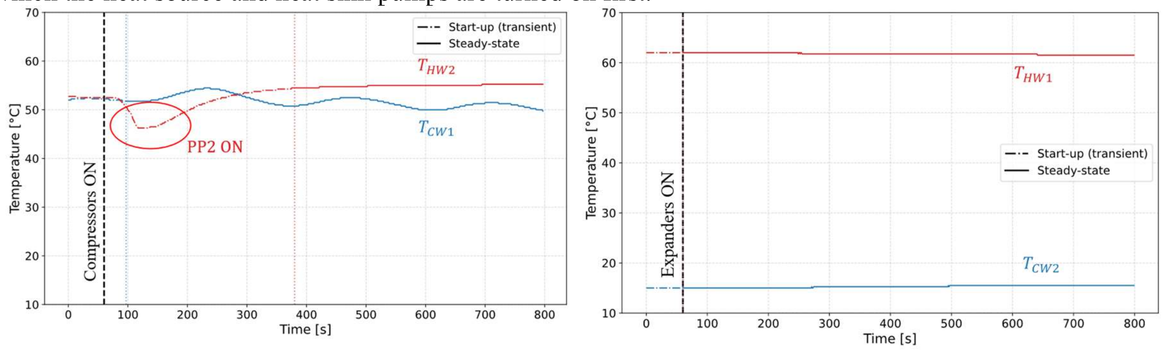
Temperature profiles:

As can be seen in Figure 6, the trends of the secondary fluid temperature differ significantly between HP and

ORC modes.

In HP mode, $T_{HW2}$ is the inlet water temperature at the condenser while $T_{CW1}$ is the inlet water temperature at the evaporator. For $T_{HW}$ , the temperature initially drops before stabilizing. This behaviour is due to the start-up sequence: the pump PP2 is switched on after the compressors. Consequently, during the first steps of start-up, the heat exchanger is heated primarily by conduction. Once water from the thermal storage begins circulating, the temperature decreases due to the water contained in the pipes between the tank and the heat exchanger before stabilizing at the thermal storage outlet temperature. In contrast, $T_{CW}$ is already stable at the start because the water pumps have been running for some time. Any oscillations in $T_{CW}$ are caused by the ON/OFF operation of the boiler.

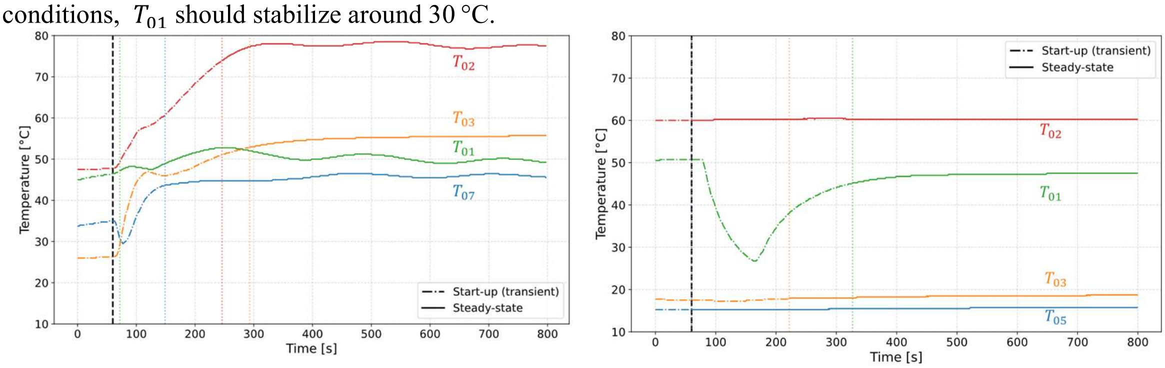
In ORC mode, temperature profiles are much more stable. This stability arises from the start-up procedure, in which the heat source and heat sink pumps are turned on first.




a) HP mode b) ORC mode

Figure 6. Comparison of secondary fluid temperature evolution in HP/ORC mode.

Figure 7 shows the refrigerant temperature profiles. In HP mode, the temperature that takes the longest to stabilize is at the condenser outlet ($T_{03}$), whereas in ORC mode, the slowest to stabilize is at the expander outlet ($T_{01}$). When the expander is turned on, $T_{01}$ initially decreases before reaching a steady value. This temperature increase occurs because the expander is driven by the motor to reach the high operating speed of 6000 RPM, rather than being braked, which affects the refrigerant temperature. Under normal operating conditions, $T_{01}$ should stabilize around 30 °C.



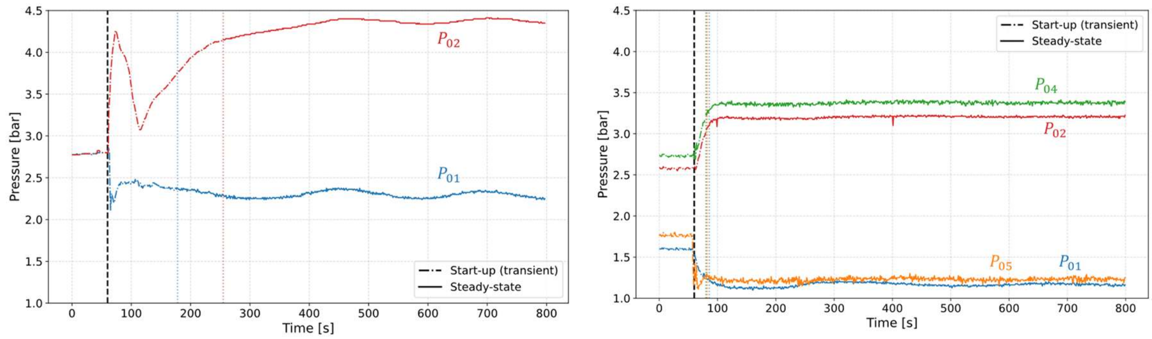

a) HP mode b) ORC mode

Figure 7. Comparison of refrigerant temperature evolution in HP /ORC mode.

Pressure profiles:

In HP mode, the pressure profiles follow the behavior of the secondary fluid temperatures (Figure 8). This correlation is likely caused by the very small pinch points in the heat exchangers (lower than 1K).

In ORC mode, four distinct pressure levels can be observed, thanks to the installation of two additional pressure sensors at the supply and exhaust of the refrigerant pump. The differences between $P_{01}$and $P_{05}$, as well as between $P_{02}$ and $P_{04}$, are due to two factors: the pressure drops across the heat exchangers and piping, and the hydrostatic pressure difference caused by the pump being located 2 m below the rest of the test bench. This installation ensures a sufficient NPSH, as discussed in Section 2.

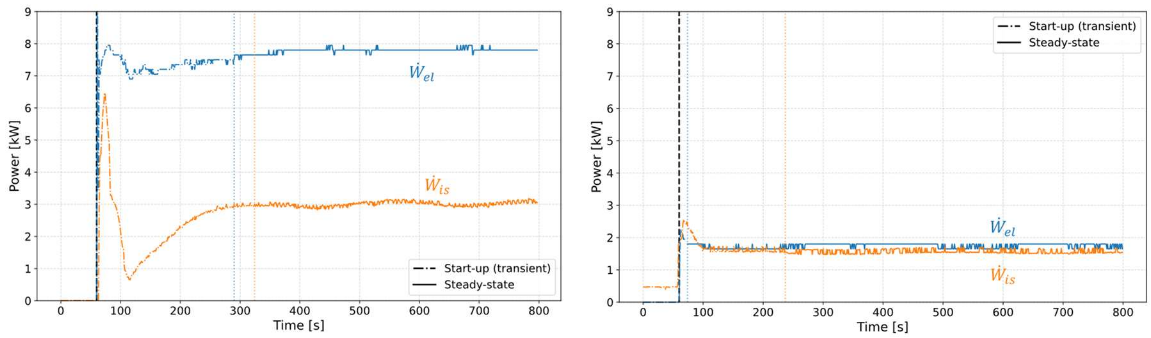



a) HP mode   b) ORC mode

Figure 8. Comparison of pressure evolution in HP/ORC mode

<u>Electricity/isentropic power consumption/generation :</u>

To make a meaningful comparison between the two operating modes, both the isentropic work and the electrical power consumption or generation are evaluated (Figure 9). In HP mode, the electrical power requires approximately 300 seconds to reach steady-state conditions, whereas in ORC mode, stabilization occurs much more rapidly. From the perspective of the electrical grid, energy must be supplied to the expander prior to its power generation phase. The duration of this transition depends on parameters such as expander rotational speed and external operating conditions. In the present case, this transition phase does not occur, leading to a relatively stable power output. The isentropic work also stabilizes faster in ORC mode than in HP mode, mainly as a result of the smoother and more controlled start-up procedure in the ORC configuration.

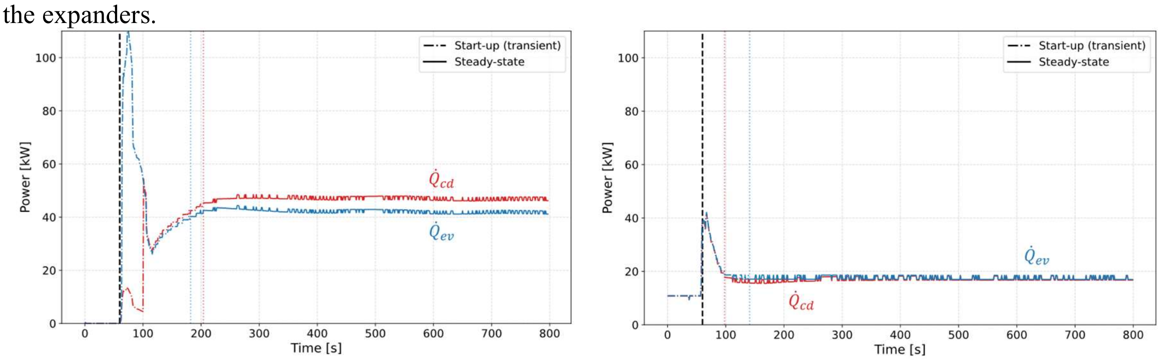



a) HP mode   b) ORC mode

Figure 9. Comparison of electrical power/isentropic power evolution in HP/ORC mode

<u>Thermal power profiles:</u>

Figure 10, shows the thermal inertia of the heat exchangers. The thermal inertia is higher in HP mode. This is due to the start-up procedure, while in ORC mode, both heat exchangers are preheated/cooled prior starting the expanders.




a) HP mode   b) ORC mode

Figure 10. Comparison of thermal power evolution in HP/ORC mode

## 5. Conclusion

As demonstrated in this study, the start-up of a reversible system employing high fixed-speed scroll machines presents significant challenges. In HP mode, operating the scrolls at 6000 RPM results in liquid blockage during start-up. Conversely, in ORC mode, the high rotational speed needed prevent efficient electricity generation under off-design conditions.
To mitigate these issues, several strategies were evaluated:

- Modification of the pulley ratio to reduce the fixed rotational speed. This approach, however, is unsuitable for the current test bench configuration. Due to the small swept volumes of the scrolls, achieving the design refrigerant mass flow rate – and consequently, the design power levels – requires operation at maximum speed.
- Integration of a dedicated inverter for each scroll machine. This option provides the most effective solution, enabling smoother start-up in HP mode and enhanced flexibility under off-design ORC conditions. Despite its higher cost, this solution was selected, as the project objective has evolved beyond simple demonstration toward optimized system operation.

While the fixed-speed configuration shows potential, its successful implementation would require lower operational speeds, which could be achieved only by selecting scroll machines with larger swept volumes. This adjustment would allow smoother start-up in HP mode. However, in ORC mode, a fixed-speed system would need to operate close to nominal conditions; otherwise, it may be unable to generate electricity.
This work also shows that the start-up dynamics is highly dependent on the start-up procedure and that finding an optimized start-up process can decrease risk of failure due to transient behaviours (current, pressures,…). The next steps of this work will be to optimize this procedure using variable speed scroll machine, to decrease these transient effects.

## Acknowledgements

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