Experimental Investigation of a Novel Hybrid Heat Pump

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Abstract

Heat pumps can employ renewable electrical energy to generate domestic heat without the use of fossil fuels. Hybrid heat pumps aim at utilizing different low-temperature heat sources. In this study, a hybrid heat pump is developed which can draw ambient energy from either a ground source heat exchanger or an air source heat exchanger depending on their corresponding temperature levels. To fully exploit the energetic potential, additional operation modes are enabled. These include efficient parallel operation of both ambient heat sources, novel defrosting interconnections as well as active regeneration of the soil. A novel two-compressor refrigerant cycle is developed. The hybrid heat pump is tested in a laboratory under realistic conditions and the refrigerant cycle behavior is analyzed. The single-source operation results promise comparable efficiencies with conventional ground source heat pumps. Switching between the single source operation mode requires a procedure to not loose refrigerant mass in the unused part-cycle. The PID controller of the air refrigerant part-cycle expansion valve is not optimized and leads to minor fluctuations on evaporation pressure, superheating temperature and heating power. Parallel operation shows high efficiency at low ambient temperatures while requiring low heating power from the heat sources. The evaporation pressures are properly separated and stable. Further operation modes like defrosting and active regeneration of the ground heat source still need to be tested.

Keywords: hybrid heat pump, novel refrigeration cycle, parallel operation, experimental investigation

1. Introduction

Heat pumps are heat generators for both space heating (SH) and domestic hot water (DHW). Most heat pumps operate with electrical energy and can therefore be operated renewably by employing renewable electrical energy sources like wind turbines and photovoltaics. Differently from direct electric heaters, heat pumps can deliver a larger amount of heat energy than electrical energy consumed. This is due to the utilization of low-grade thermal energy from ambient heat sources like air, ground water or the soil. A so-called refrigerant cycle is needed to utilize these low-temperature sources. An ideal refrigerant cycle is a reverse Carnot cycle. Its efficiency is dependent on the boundary conditions, namely the temperature levels of the heat source T_{so} and the heat sink T_{si}. The maximum theoretically possible efficiency is described by the reverse Carnot efficiency η_{carnot}:

$$\eta_{\mathrm{carnot}} = \frac{T_{\mathrm{si}}}{T_{\mathrm{si}}-T_{\mathrm{so}}} \qquad \text{(eq. 1)}$$

In practice, a heat pump's efficiency is calculated as the ratio between the heating power $\dot{Q}_{\mathrm{heat}}$ and the electrical power P_{el} and is called the coefficient of performance (COP):

$$\mathrm{COP} = \frac{\dot{Q}_{\mathrm{heat}}}{P_{\mathrm{el}}} \qquad \text{(eq. 2)}$$

Commonly, COPs vary between 2 and 6 depending on the temperature levels of the heat source and heat sink (SH or DHW) (John Cantor Heat Pumps, n.d.; Naldi, et al., 2014; Natural Resources Canada, 2022). The heating power $\dot{Q}_{\mathrm{heat}}$ is calculated via the mass flow $\dot{m}$, the specific heat capacitance c_p, and the temperatures of inlet and outlet flow, T_{in} and T_{out} respectively:

$$\dot{Q}_{\mathrm{heat}} = \dot{m} * c_p * (T_{\mathrm{out}} - T_{\mathrm{in}}) \qquad \text{(eq. 3)}$$

Air source heat pumps (ASHP) are heat pumps with an air source heat exchanger (ASHX) as the heat source, ground source heat pumps (GSHP) are connected to ground source heat exchangers (GSHX), usually via a brine cycle. While ASHPs usually have lower investment costs, the annual efficiency is generally lower than that of a GSHP, since the soil is a large heat storage and is usually on a higher temperature level during winter. An increased source temperature leads to an increased efficiency according to equation 1.

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Superheating temperature T_{sup} is a major criterium for a properly operating refrigerant cycle. Superheating of the refrigerant is necessary at the inlet of the compressor. Superheating temperature is an increase of the refrigerant temperature at compressor inlet $T_{\text{comp,in}}$ above the evaporation temperature T_{evap}. Fluid parts would damage the compressor and reduce the service life significantly. Therefore, the expansion valves control the mass flow through the evaporator to allow superheating of the refrigerant. Superheating temperature is then defined as:

$$T_{\text{sup}} = T_{\text{comp,in}} - T_{\text{evap}} \qquad \text{(eq. 4)}$$

The fluids within refrigerant cycles are called refrigerants. They change their phase between liquid and gaseous, absorbing or dissipating heat accordingly. They need to evaporate and condensate at usable temperature ranges and require high specific vaporization capacities for compact component design. There is a strong need for new refrigerants due to formerly employed fluids being either damaging to the ozone layer and current ones having high global warming potentials when released. A Europe-wide phase down of harmful refrigerants is pursued (EU, 2014). While natural refrigerants like propane (R290) are promising regarding the environmental impact as well as thermodynamically, their flammability still prevents broad usage. A large number of mixtures is currently on the market, weighing up environmental impact and safety of use.

Research is going on to combine several heat sources within one single heat pump. These are commonly referred to as hybrid heat pumps (HHP). One research group employed two separate GSHXs together with an air-based regenerator as the heat source for a HHP. The system was optimized for an office building and the cooling load was dominant. The heat pump purely operated with the GSHXs, while the air-based regenerator supported in keeping these at low-enough temperature levels for passive cooling. The two GSHXs were operated serially for heating, cooling or being allowed to regenerate (passively or actively via the ASHX). Still, they found a decrease of 47 % in borehole sizing when compared to a single ground-source heat pump system. (Allaerts, et al., 2014)

Another project analyzed a HHP with an air and a ground heat source. The heat pump could switch between either heat source and deliver heating power to an office building. They found comparable efficiencies to a GSHP while having half the area for a horizontal GSHX. The interconnection did not allow parallel operation of both heat sources or novel defrosting methods. Only passive regeneration of the GSHX was possible. (Corberan, et al., 2018)

Within the research project Hybrid Heat Pump+, funded by the Federal Ministry for Economics and Climate Action, Germany, parallel operation of two heat sources was identified as a major innovation. The basic scheme can be seen in Figure 1.

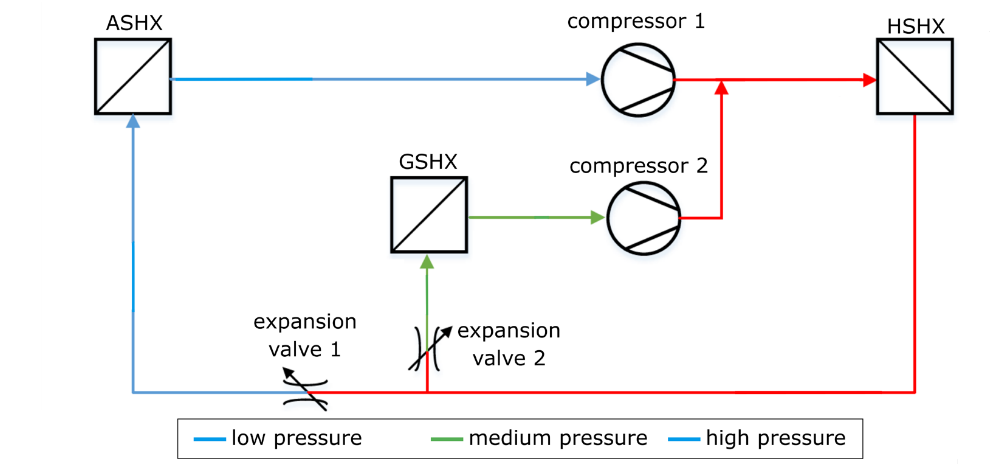

Figure 1: Scheme of the initial idea of a refrigerant cycle with two heat sources, two compressors and two expansion valves. This allows separated evaporation pressures (blue and green) to supply one condensation pressure (red).

An ASHX as well as a GSHX was chosen. Both heat sources can be designed at lower power, since peak loads can be supplied by both heat sources in parallel. Also, flexible variation of the load on the sources helps managing the energy content of the GHSX while preventing freezing. To allow efficient parallel operation, an interconnection using two compressors needs to be developed. Further operation modes were to be realized: First, defrosting of the ASHX using the GSHX is more efficient, since reverse cycle defrosting always draws energy from the heating system heat exchanger (HSHX). Second, active regeneration of the GSHX during times of comparatively high ambient air temperatures can reduce the necessary area of the GSHX even further compared to previous research results. These required additional valves and control algorithms.

For an evaluation of these operation modes, a prototype needs to be built and experimentally investigated. The novel refrigerant cycle operation is analyzed. Of special interest are superheating temperatures of the refrigerant within the heat sources, stability of the expansion valve controls and refrigerant amount within the operating part-cycle. Based on the efficiency findings during single-source operation, a comparison of the COPs with a conventional heat pump line is drawn. This allows evaluation of the impact of the increased complexity of the refrigerant cycle on single source operation. Parallel operation can be analyzed first regarding the refrigerant cycle stability concerning superheating, refrigerant amount and overall stability of the operation. Also, the parallel operation COP as well as the power needed from the GSHX can be compared to the single source operation.

2. Methodology

An interconnection was designed based on a refrigerant cycle with two heat sources and two compressors connected to a single HSHX. The implications on the refrigeration cycle had not been analyzed before and were a major evaluation point for the experimental investigation. Additional valves and components can influence the single-source operation. In parallel operation, it was unknown how the superheating of the refrigerant after the evaporators and the expansion valves will behave. Also, the refrigerant mass needed to be supervised closely to avoid a lack during certain operation modes or when switching from one operation mode to another.

Apart from the single source and parallel operation of both sources, the interconnection was required to allow several further operation modes:

- Defrosting using reverse cycle mode, where the ASHX is defrosted using the HSHX as the heat source.
- Defrosting of the ASHX using the GSHX as the heat source.
- Active cooling using either the ASHX or the GSHX as the heat sink for the HSHX.
- Active regeneration of the GSHX, both serially to heating as well as parallel to ASHP operation.

The HHP was manufactured by the industry partner ratiotherm GmbH & Co. KG and implemented into a thermal test bench, where the boundary conditions could be set. Testing for the single source operation modes was done based on the standard DIN EN 14511 (DIN e.V., 2019). This is the commonly used standard for heat pump efficiency testing in Europe and allows for comparable results. Market-available heat pumps from the same manufacturer were used for a comparison of the single-source operation. This allows a determination of the impact of the additional components within the refrigerant cycle on the efficiency.

The parallel operation mode of both heat sources was tested at low ambient temperatures. Since this operation mode is supposed to cover primarily peak loads, both compressors are supposed to run at full speed at design ambient temperatures according to (DIN e.V., 2020). For the location of the test bench, Ingolstadt yields a design ambient temperature of $T_{\mathrm{a,design}} = -13.3$ °C. At highest possible load, the total efficiency was measured and compared to the extrapolated single source operation efficiencies of the HHP. Internal sensors of the HHP allowed a supervision of the superheating and the control of the expansion valves. For this control, a temperature sensor was located at the inlet of the compressor. The evaporation temperature is commonly determined by the evaporation pressure along with a lookup table of the specific refrigerant.

For all experiments, the refrigerant amount was supervised. A large refrigerant collector was included in the interconnection to buffer the necessary refrigerant amount. However, if refrigerant is locked out of the operated part-cycle into a non-operated part-cycle by valves, a lack of refrigerant mass can impact the efficiency significantly. Two methods were used to detect this behavior:

- A flow sight glass within the refrigerant cycle, commonly placed in front of the expansion valve, allowed a determination of the refrigerant’s state at this point. Usually, the refrigerant is supposed to be fully liquid at this point. Gas bubbles are an indicator for too low refrigerant amount within the currently operated part-cycle of the HHP.
- The evaporation pressure got reduced when the amount of refrigerant was lowered. This could help when comparing different experiments with each other for example after an operation mode switch (e.g., from ASHP operation to GSHP operation and back). If the evaporation pressure is reduced, this is an indicator for a loss of refrigerant mass to the non-operated part-cycle.

3. Thermal Test bench

A thermal test bench was used to emulate the ambient conditions during the testing of the HHP. It consists of a hydraulic test bench, sensors and actuators as well as a control program written in LabVIEW on a connected PC. The hydraulic part has a heat source and a heat sink. The heat source is supplied by electric heaters and can deliver the heat via a brine cycle to the heat pump. This allows source temperatures below water freezing point. The heat sink is a water-based cycle connected to a cooling machine.

Both hydraulic cycles can control the volume flow and the temperature return flow. The volume flow is controlled by a flow valve. The current volume flow is measured continuously and a PID controller withing the LabVIEW control program changes the opening of the flow control valve accordingly. The temperature return flow to the heat pump is controlled by a mixing valve, comparing the currently measured return temperature to the setpoint. Similar to the volume flow control, a PID controller is implemented in the LabVIEW control system. Pipe lengths between the controlled components and the measuring sensors lead to delays especially when operating with low volume flows.

Insulated tubes connect the test bench to the HHP. They lead to an additional delay when changes to the setpoints occur. To avoid major impacts of these delays, static tests are conducted.

To allow ASHP tests, there is also a climate chamber where the ASHX was placed. The climate chamber is supplied by an air conditioning system and an exhaust in flow-through connection. Sensors allow control of the temperature and the relative humidity in the climate chamber. Since the ASHX extracted additional heat from the climate chamber, the setpoints of the air conditioning system needed to be adjusted. Again, static tests are beneficial to avoid complex control challenges.

The LabVIEW program as well as the monitoring system of the HHP store the measurement data continuously. This allows evaluation of the experiments using MATLAB.

4. Results and Discussion

This chapter is divided into three parts: First, the designed interconnection is shown and explained. Second, the experimental results for single source operation are discussed. Third, parallel operation is analyzed.

4.1. Designed interconnection of the HHP

The HHP was designed as a split heat pump, which means the ASHX is within a different casing than the compressors and connected via refrigerant-filled pipes. This allowed usage of existing parts from the industry partner as well as simple connection to the test bench's climate chamber. To tackle the challenge of the refrigerant phase-down, a novel blend was used. R-454B has a significantly lower GWP than common refrigerants for heat pumps, but has an ASHRAE classification of A2L (Chemours, 2021). This means it is flammable. Refrigerant sensors had to be added to the test bench facilities to reduce risks.

To allow the additional operation modes, several components had to be added. These included magnetic valves, one-way valves as well as several manual ball valves as two-directional flow valves. For a full automation of the heat pump, the manual valves need to be substituted. A large refrigerant gatherer was included to buffer the refrigerant amount needed for different operation modes. Additional heat exchangers were added: a high-temperature heat exchanger for a second heating system on an increased temperature level was included into the GSHP branch. An economizer was added into the ASHP branch, which is increasing the efficiency by transferring heat from the condensated high pressure refrigerant flow into the superheated refrigerant flow before the compressor (Jin, et al., 2016). These heat exchangers were constructive adjustments from the industry partner to be able to use existing parts. Further operation modes are possible, but within this paper, the focus lies on these aforementioned ones. The full interconnection is shown in Figure 2.

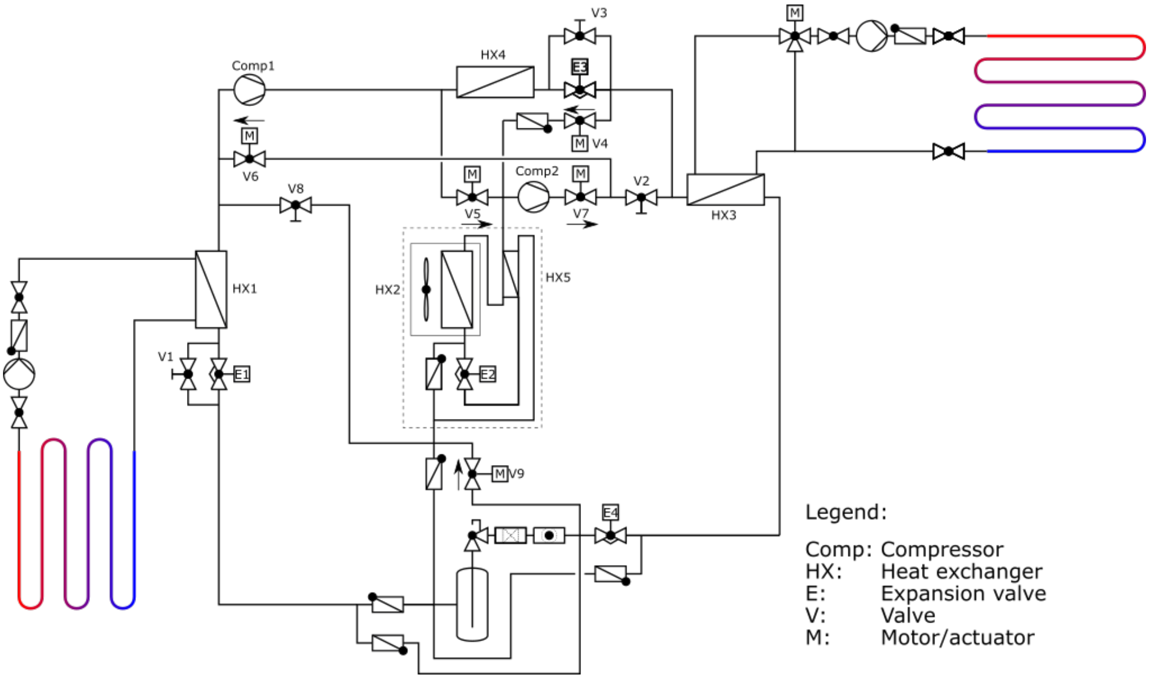

Figure 2: Interconnection of the novel HHP including all components. Sensors are not shown. Both ground source as well as heating system are shown schematically.

During GSHP operation, HX1 is the GSHX and the refrigerant is compressed by Comp1. HX4 is the high temperature heat exchanger and currently not used. Through the ball valve V3, the refrigerant reaches the HSHX HX3 and condenses. Through one-way valves and the refrigerant gatherer along with filters and the sight glass, the liquid refrigerant reaches the expansion valve E1. After the pressure reduction, the refrigerant is fed into HX1, closing the cycle.

ASHP operation uses Comp2, HX2 and E2 respectively. Different valves need to be opened to allow the refrigerant cycle, namely the manual ball valve V2 and the magnetic valves V7 and V9. The economizer HX5 is used here.

Parallel operation is the combination of both previous operation modes, ASHP and GSHP. Both compressors are run as well as both expansion valves and the aforementioned valves have to be opened accordingly.

Defrosting can be done using a reverse cycle principle. By operating Comp1 and opening the ball valve V2 and the magnetic valves V5 and V6, heat can be taken from HX3 on a low pressure and the refrigerant condenses within the ASHX HX2. In this case, E4 acts as the controlling expansion valve.

The second defrosting mode is using the GSHX HX1 as the heat source and Comp1 as the compressor. Opening the magnetic valve V5 and controlling with the expansion valve E1 closes the cycle.

Active regeneration of the GSHX HX1 can be done by using the Comp2 to compress the evaporated refrigerant from the ASHX HX2. Through the magnetic valves V6 and V7, the refrigerant reaches HX1. The ball valve V1 leads the refrigerant to the refrigerant gatherer and the magnetic valve V9. E2 acts as the expansion valve.

4.2. Single source operation

The single source operations were measured according to (DIN e.V., 2019). The standard defines source and sink temperatures, tolerances and test durations. The test bench cannot fulfill all the requirements (e.g., temperature sensors and electricity meter exceeded tolerances). But the measurement points regarding source and sink temperatures were followed, so the measurements for both the air source as well as the ground source operation were based on this standard.

For the evaluation, the COP was calculated using equation 2. Over the duration defined in the aforementioned standard, the heating power using equation 3 as well as the electrical power of the compressor were measured. The division lead to the COP.

Figure 3 shows the results for the single source operations and compares these with a market-available heat pump of the industry partner ratiotherm. As expected, lower sink temperatures T_{si} lead to higher efficiencies COP. With increasing source temperatures (ambient air temperature and brine temperature respectively), the COP increases as well. The comparison is done with heat pumps from the Max-line of ratiotherm (ratiotherm GmbH & Co. KG, n.d.). They contain several similar components like the heat exchangers, but different compressors and most notably a different refrigerant. Another factor is, that the HHP COPs do not include peripheral power for pumps, ventilators and electrical components. While these generally play a minor role compared to the compressor, this lowers the total electrical consumption and therefore increases the COP. These are the major reasons for the improved COP of the HHP compared to the market-available Max-line heat pumps. It can, however, be said, that the HHP does not suffer from the additional complexity of the hydraulic refrigerant cycle to a large extent.

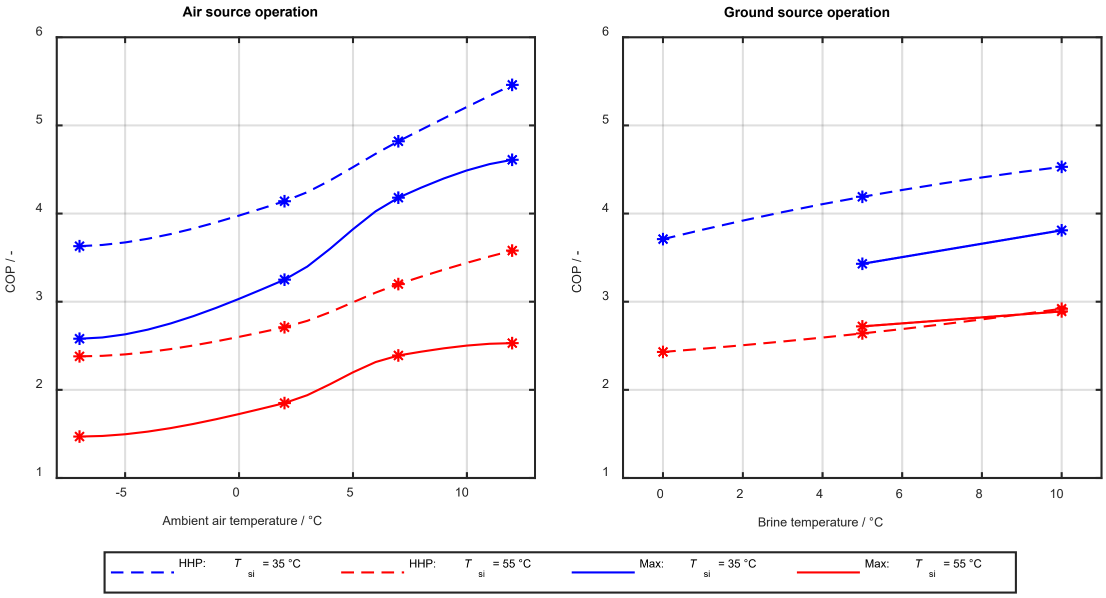

Figure 3: Experimentally derived COPs of the HHP. Different heating temperatures T_{si} are shown in red and blue. Dashed lines show the HHP measurement results, the full line the comparison with heat pumps from ratiotherm's Max production line. The measurement points are interpolated using monotone piecewise cubic interpolation (Fritsch and Carlson, 1980).

During the measurements of the air source operation, a significant fluctuation of the COP was detected. This was a result of significant fluctuations of the evaporation pressure. This in turn is a result of fluctuating superheating temperature T_{sup}, which is again a result of fluctuating expansion valve opening. This behavior is shown in figure 4. The expansion valve was not controlled appropriately, fluctuating between 21 and 37 % PWM opening degree. The most likely reason is improper PID parameters within the HHP software. This needs to be considered when analyzing further operation modes.

A shift of the refrigerant mass was detected as well. When switching between the single source operations, a significant lack of refrigerant occurred. This could be seen in both the sight glass before the expansion valve as well as in decreasing evaporation pressures compared to previous experiments in the same operation mode. To avoid this, all valves had to be opened when switching between the operation modes, allowing the refrigerant to spread in the whole refrigerant cycle instead of gathering in the evaporators. Since this procedure took time and even had to be repeated occasionally, a more careful analysis and a proper procedure still needs to be developed to tackle this difficulty.

4.3. Parallel operation

As discussed, parallel operation was tested in low ambient air temperature conditions to be able to cover peak loads. Due to a mechanical issue with the air source compressor (Comp2 in figure 2), full loads were not possible anymore without overloading the inverter and an error stopping the operation. Therefore, the test was conducted at lower inverter frequency $f = 50\ \%$ and increased ambient air temperature $T_{\mathrm{a}} = -10\ °\mathrm{C}$ instead of at full load at the design ambient temperature $T_{\mathrm{a,design}} = -13.3\ °\mathrm{C}$. Theoretically, parallel operation therefore could deliver more heating power, but the general behavior regarding refrigerant amount and superheating temperature should remain the same. If possible, this should be examined in future tests.

Figure 5 shows the heating and electrical power as well as the COP during parallel operation. Over the shown duration, the average COP was 3.43. To compare this with the single source operation, the air source operation needed to be extrapolated and resulted in a COP of 2.69. The brine source operation yielded a COP of 3.71 at similar boundary conditions. While the latter is better than the parallel operation, parallel operation is more efficient than the air source operation. The main benefit is the omission of any other peak load heat generators: In practice, electrical heating rods are often used to cover peak loads, adding an electrical efficiency of 1. The parallel operation of the HHP is significantly more efficient in these low temperature ambient air conditions. The second benefit is a significantly lower load on the GSHX. While the HHP extracted about 3.7 kW from the brine cycle, the single source operation would need 5.8 kW to cover the same heating power of about 8 kW, which the HHP delivers during this experiment. This is a reduction to about 64 %.

Figure 4: Graph of the air source operation focusing on evaporation temperature and superheating. The fluctuations derive from an improper PID controller for the expansion valve.

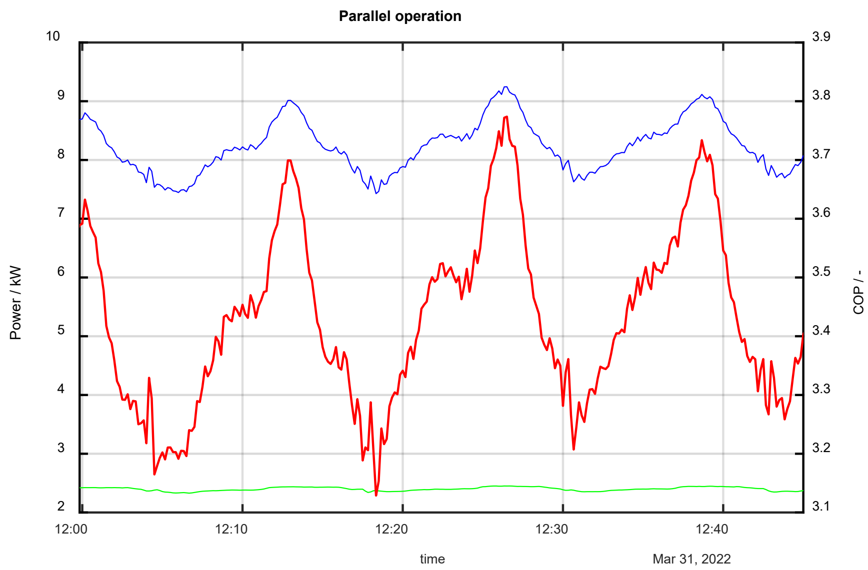

Figure 5: Overview of the parallel operation at ambient temperature $T_a = -10$ °C, brine temperature $T_{brine} = 0$ °C and sink temperature $T_{si} = 35$ °C.

The fluctuations of the COP and the heating power during parallel operation were significant. As seen in the single source operation, the issue lies within the control of the expansion valve within the ASHX part-cycle. Figure 6 shows the behavior of evaporation temperature and superheating of both part-cycles. While the operation was stable for the GSHX part-cycle, the values fluctuated for the ASHX part-cycle. This is a fundamental issue with the expansion valve control and not of the parallel operation.

The refrigerant amount was found to be no issue, since both part-cycles are actively taking part in the operation. No lack of refrigerant was detected, even during switching to and from parallel operation.

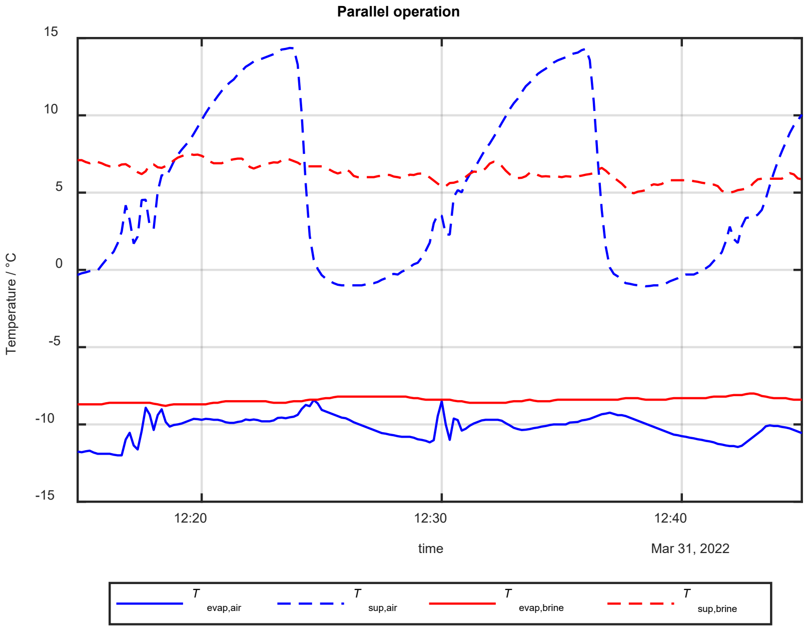

Figure 6: Graph of the parallel operation focusing on evaporation temperatures and superheating. The fluctuations of the air source part-cycle are based on an improper PID controller of the expansion valve.

5. Conclusions and Outlook

The developed interconnection for the HHP is significantly more complex than traditional heat pumps. Not only two compressors and expansion valves need to be controlled, the number of valves is a challenge not only economically but also for the control strategy. Whether the full list of operation modes is necessary and ecologically as well as economically feasible needs to be determined further.

First tests with the HHP in single source operation showed promising results. Even though the complexity of the refrigerant cycle is increased and additional components are placed in the refrigerant flow, the efficiency was comparable with market-available heat pumps. Detailed analysis including the electric power consumption of the periphery as well as optimization of the PID control of the air source operation expansion valve need to follow. One more challenge was detected: The refrigerant amount was not easily available to the other single source operation when changing the heat source. Further investigation of this behavior and development of a proper control strategy might further increase the complexity of the HHP control.

Parallel operation of both heat sources proved to be possible. The evaporation pressures were separated during operation and the expansion valves controlled independently. The efficiency was promising: It was an improvement compared to air source operation, while – expectedly – not reaching efficiencies of brine source operation. However, the heating power was significantly increased over the single source operations. Also, the power extracted from the GSHX could be reduced to almost 64 % by parallel operation at this specific measurement point. The issue with the air source part-cycle expansion valve fluctuating could be seen in this operation mode as well, leading to an overall fluctuating heating power and COP. No issue with the shifting refrigerant amount was observed.

The remaining operation modes need to be analyzed in detail. The refrigerant amount issue needs to be evaluated and possible control strategies to be developed. For an energetic evaluation, an annual simulation using models derived from the experimental data and an optimal control strategy needs to be conducted. These are further steps to tackle the challenges of the novel HHP.

6. Acknowledgments

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