



# Simulation-based investigation of active regeneration in a novel dual-source heat pump refrigerant cycle

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## Abstract

Dual-source heat pumps serve as a complex compromise to reduce individual disadvantages of traditional air- and ground-source heat pumps. One way to reduce net energy extraction and therefore space requirements compared to ground-source HPs is by adding refrigerant cycle-based active regeneration. In this paper, instead of using solar thermal regeneration, a refrigerant cycle-based procedure is introduced and investigated. Serially connecting two heat sinks on different temperature levels allows subcooling of the refrigerant before expansion. An experimental investigation is conducted, showing the condensation shifts and therefore more heating power being delivered to the low temperature heat sink. Only 23 % is delivered to the high temperature sink. Two ways of controlling the ratio between the heating power between these two heat sinks are investigated with a verified physical simulation: First, changing the refrigerant charge affects the ratio of heating power to the heat sinks. Increasing the refrigerant charge floods the condenser and shifts condensation and therefore 81 % of heating power happens in the high temperature heat exchanger. This yields high total heating powers and efficiencies but is inherently less flexible; Second, adding a mid-expansion valve between the two heat sinks allows for flexible control and a wide range of heating power ratios. While total heating power and efficiency are reduced by up to 10 %, the high temperature heat sink receives between 22 to 73 % of the total heating power, demonstrating the high flexibility. This allows integration of refrigerant cycle-based active regeneration with flexible power outputs into a dual-source heat pump to reduce ground-source heat exchanger sizes.

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## 1. Introduction

Heat pumps are one of the main technologies for the heat transition in the building sector, shifting from burning fossil fuels to sustainable heat generation. However, densely populated urban areas still pose several challenges for HPs as explained by the IEA [1]: While air-source heat pumps are easy to install and comparatively cheap, they suffer from low winter efficiencies and high sound emissions. On the other hand, ground-source HPs employ ground-source heat exchangers (GSHX) to benefit from more stable heat source temperatures and therefore higher efficiencies in winter and have lower noise emissions. However, the installation of these GSHX is both expensive and complex, with vertical boreholes requiring suitable geology and lengthy allowance procedures, horizontal collectors typically require large outdoor areas for the installation, not available in many urban areas [1, 2]. A suitable decentralized heat pump solution for densely populated urban areas (e.g., terraced buildings) is still required especially where air-source or ground-source heat pumps are not easily applicable and no centralized solution like district heating networks are economically feasible.

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Dual-source heat pumps (DSHP) allow heat extraction from two heat sources. Typically, air, ground and solar thermal heat sources are combined [3]. DSHPs with a GSHX often focus on increasing efficiency by using the more efficient heat sources and / or reducing the energetic load on the GSHX by utilizing air or solar thermal energy when these heat sources are (efficiently) available. For example, by switching to ambient air-based heat exchangers during relatively warm periods during spring, autumn and even winter, the total net energy extracted from the GSHX is reduced, allowing a smaller GSHX dimensioning compared to traditional ground-source HPs [2]. Studies found a reduction of about 50 % of extracted energy and therefore borehole length reduction is possible when combining ground and air heat sources in a DSHP without compromising annual efficiency [4–6]. However, there are disadvantages due to additional hydraulic or more complex refrigerant cycle interconnection if one wants to incorporate both heat sources in one DSHP system on the hydraulic or refrigerant cycle side, respectively.

Similarly, combining a GSHX with solar thermal collectors enables regeneration of the GSHX when solar energy is available. For example, these systems can use conventional ground-source HPs and couple the solar thermal collectors to directly regenerate the GSHX via the brine cycle [3, 7]. These systems are indirectly reducing the extracted energy from the GSHX by feeding energy into the GSHX, reducing the net energy load. These systems can also prevent thermal drift of a GSHX, which is a long-term temperature drift caused by extracting more energy from a GSHX than can be passively regenerated over the year. Thermal drift of a GSHX dimensioned too small can be prevented by actively regenerating the GSHX during summer. For example, Chhugani et al. [8] showed that this operating strategy allows reduction of a borehole GSHX size of up to 35 % compared to traditional design at no efficiency cost.

Few studies have investigated a combination of the two core principles of this idea: Utilizing the highly flexible and irradiation-independent air-source heat exchangers with regeneration of a small dimensioned GSHX. Allaerts et al. [9] have combined a dry cooler for an office building, reducing the GSHX size by about 50 %. However, this study was done on a cooling-dominated building and did not incorporate the increased effectiveness of the refrigerant cycle. Another study by Siren et al. [10] have found area reductions of 89 % for larger borehole fields for a DSHP with ASHX regeneration. However, their focus was on reducing borehole spacing for a large multi-family heating system.

Refrigerant cycles can enable regeneration even across negative temperature gradients, e.g., when the air temperature is slightly lower than the GSHX temperatures. In these cases, when the HP uses the air-source heat exchanger as the source to deliver heat to the GSHX, the small temperature gradient allows a highly efficient so-called active regeneration (AR). Reum et al. [11] have investigated the potential of such a system in annual simulations. Only serial AR was investigated when no heating power was required. They found significant reductions in the GSHX size were possible by utilizing the GSHX only as a peak-load heat source and regenerating the GSHX during relatively warm winter periods. The DSHP system did, however, suffer from low efficiencies, resulting from inefficient AR (which was only approximated with several assumptions) and relatively inefficient ground-source operation. They proposed additional research to investigate AR in DSHP system to improve the operation mode as well as optimization of the control strategy.

### *1.1. Research question*

This paper aims to investigate the operation of AR, especially with regards to the interconnection and optimal control strategies of the refrigerant cycle. Delivering heat on two different temperature levels in one single refrigerant cycle promises to offer increased flexibility in various use cases, e.g., in GSHX utilization in a DSHP system. Without impacting the heating power provided. However, the integration of this AR is not trivial, both in the refrigerant cycle as well as the control strategy.

Therefore, a refrigerant cycle interconnection with two serially linked heat exchangers acting as heat sources is introduced and investigated. Based on an experimental investigation, optimized operational strategies are proposed and assessed simulatively. The first solution does not change the refrigerant cycle interconnection but only adjusts the refrigerant charge. The second solution adds a mid-expansion valve (MExV) to increase flexibility of the operation. The results are investigated for effectiveness, efficiency, and flexibility in the proposed use case of AR.

## **2. Methodology**

In the following sections, the methodology of the paper is described. First, the experimental investigation of two serially heat sink heat exchangers is explained. Condensation mostly occurs in the heat exchanger with lower temperature and shifts most heating power from the high-temperature heat exchanger (HTHX) to the

low-temperature heat exchanger (LTHX). This makes practical solutions for this condensation shift necessary. Second, a simplified simulation model in Simscape<sup>1</sup> is introduced and used to assess two possible solutions to the practical problem of condensation shifting. The overall methodology schematic is displayed in Fig. 1.

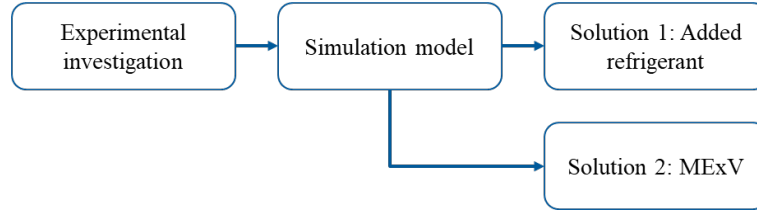


Figure 1: Schematic of the methodology.

For the evaluation, heating, regenerative and electrical powers are calculated. Heating powers – for both the HTHX and the LTHX powers – are calculated following:

$$\dot{Q} = \dot{V} \times \rho \times c_p \times \Delta T. \quad (1)$$

In the HTHX, constant values for water are used for the density  $\rho_w$  and the specific heat capacity  $c_{p,w}$  to calculate the heating power  $\dot{Q}_{\text{heat}}$  in the HTHX. In the LTHX, a glycol brine is used – since the exact composition remains unknown, average values of  $\rho_b = 1,065 \text{ kg/m}^3$  and  $c_{p,b} = 3.1 \text{ kJ/kg}$  are used to calculate the regenerative power  $\dot{Q}_{\text{reg}}$  in the LTHX. The ratio between of the heating power to the HTHX to the total heating power – which is the sum of both heating powers – is calculated using:

$$r = \frac{\dot{Q}_{\text{heat}}}{\dot{Q}_{\text{heat}} + \dot{Q}_{\text{reg}}}. \quad (2)$$

Electrical power is calculated using the product of voltage  $U$  and current  $I$ :

$$P_{el} = U \times I. \quad (3)$$

Additionally, the common efficiency ratio for heat pumps – the coefficient of performance (COP) – is used:

$$COP = \frac{\dot{Q}_{\text{heat}}}{P_{el}}, \quad (4)$$

Only the electrical power of the compressor is used in this work. A specific case is used in this paper to account for the total heating power delivered by the refrigerant cycle, which adds the regeneration power to the usable heating power to calculate the total COP:

$$COP_{\text{tot}} = \frac{\dot{Q}_{\text{heat}} + \dot{Q}_{\text{reg}}}{P_{el}}, \quad (5)$$

### 2.1. Experimental investigation

The experiment is conducted on a simple serial interconnection of two heat sinks as displayed in Fig. 2. During AR, the ASHX is the evaporator followed by the compressor (Comp) compressing the gaseous refrigerant. The refrigerant R454B then flows through the HTHX, the accumulator (Acc) and the LTHX in series and the expansion valve (ExV) reduces the pressure and controls the superheat at compressor inlet. The accumulator is placed between HTHX and LTHX because the DSHP was originally supposed to operate only with HTHX for heat supply. The LTHX was included as the second heat source.

<sup>1</sup> MathWorks website on Simscape, <https://www.mathworks.com/products/simscape.html>

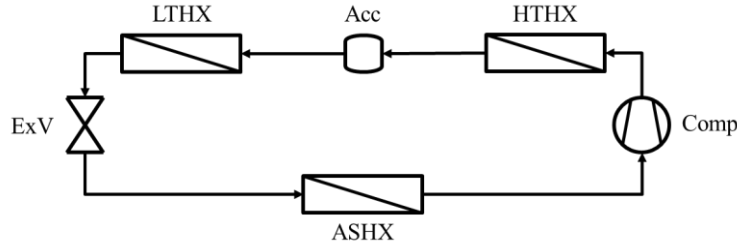


Figure 2: Schematic of the serial connection of two heat sinks for AR used in the experimental test.

A hydraulic test bench with a climate chamber is used to conduct these tests. A detailed description including sensors and measurement uncertainties can be found in Reum et al. [12]. The ASHX is placed in a climate chamber with controlled 7 °C air temperature. The HTHX is operated with water with 40 °C inlet temperature and 800 l/h volume flow. The LTHX is connected to an ice storage: A water tank of approximately 200 l is activated by the brine cycle with 300 l/h. The AR interconnection of the DSHP into the hydraulic test bench is shown in Fig. 3. In the procedure, first, the ice storage is frozen until most of the water in the ice storage has formed ice on the heat exchanger surface. This is done in regular HP operation with the GSHX as the source. Second, the ice storage can then serve as the heat sink connected to the LTHX during AR. The ice allows almost stable operation of AR for a brief period.

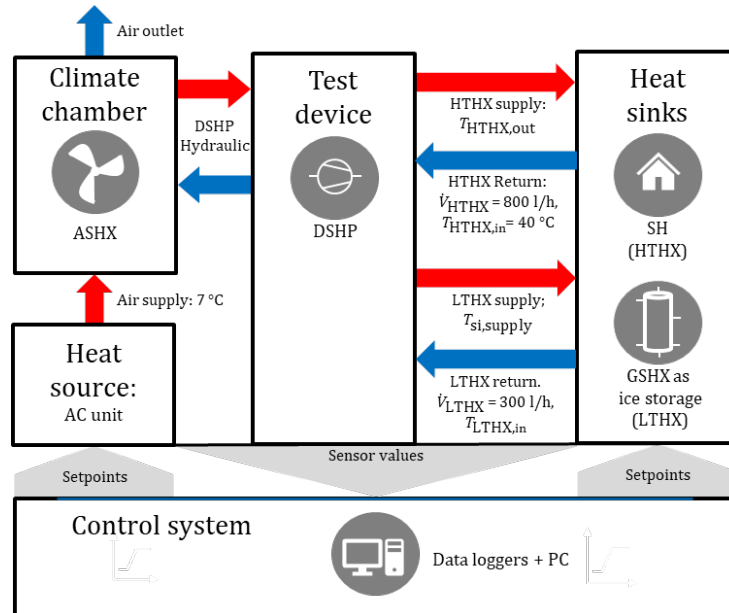


Figure 3: Schematic of the hydraulic test bench. The indicated interconnection and sensor values apply to AR.

Pressure and temperature between the main cycle components are measured. Mean pressure and temperature measurements allow calculation of the enthalpies using the REFPROP<sup>2</sup> software. For the investigation of the refrigerant cycle effects, these values are shown in pressure enthalpy diagrams.

## 2.2. Simulation model

For the investigation of possible solutions to change the heating power ratio between HTHX and LTHX, a simplified simulation model is created. The simulation model is set up in MATLAB Simscape, which is a MATLAB-based programming language integrated in Simulink to allow physical modelling. It uses refrigerant data for R454B extracted from REFPROP. It consists of a refrigerant mass flow source, system-level heat exchangers, and an electronic expansion valve.

The ideal refrigerant mass flow source assumes isentropic compression. The mass flow is calculated using the 10 coefficient method of the compressor datasheet [13], dependent on the evaporation and condensation temperatures.

<sup>2</sup> National Institute of Standards and Technology website on REFPROP, <https://www.nist.gov/srd/refprop>

The heat exchangers are designed with nominal conditions. The nominal refrigerant mass flows are taken from the aforementioned maximum refrigerant mass flow of the compressor. The ASHX is parametrized to fully evaporate the nominal refrigerant mass flow at 7 °C ambient air as the boundary condition. The LTHX is initially intended to serve as a brine-based evaporator, so it is parametrized to fully evaporate the nominal refrigerant mass flow at 5 °C brine inlet temperature. The HTHX is parametrized to fully condense the overheated refrigerant gas at 35 °C water outlet temperature. These are approximations of the real heat exchangers.

The electronic expansion valve controls the refrigerant mass flow to achieve superheating at compressor inlet of 5.3 K. A PI controller is employed.

Additionally, a small accumulator with a volume of 1 l is added between the HTHX and the LTHX. This is similar to the physical DSHP setup. An internal heat exchanger – transferring heat from before the expansion valve to after the ASHX outlet – is neglected too for simplicity. During AR, the effect of this heat exchanger is considered negligible, as the subcooling is reducing the high-pressure side temperature and therefore the heat transfer rate of the internal heat exchanger.

The parametrization is clearly simplified and not validated at this stage, but the simulation model is initially run at similar boundary conditions as the experiment. Simulations are analyzed at steady state conditions: The results are taken after running for 1800 s. This qualitative similarity simulation needs to show similar behavior as the experiment and can be demonstrated again in the pressure-enthalpy diagram. Validation needs to be done in future work.

### 2.3. Simulative investigation of possible solutions to condensation shifting

Finally, two possible solutions are tested in the simplified simulation model. First, the refrigerant mass in the refrigerant cycle is increased. This can be done by changing the initial conditions of the refrigerants in the two-phase fluid volumes of the heat exchangers and the accumulator. A higher refrigerant content is expected to flood the LTHX with liquid refrigerant, thus increasing the condensation pressure and shifting more of the condensation into the HTHX.

Second, a second expansion valve, the MExV, is added between the HTHX (more specifically: behind the accumulator) and the LTHX. The interconnection schematic during AR is shown in Fig. 4. The opening of the MExV is varied and the effects on the performance parameters are investigated. Three openings are tested, multiples of the expansion valve opening during the similarity simulation: 1) two times the opening, 2) five times the opening, and 3) twenty times the opening. The MExV must not be opened less than the ExV, otherwise the expansion of the refrigerant to evaporation pressure would completely happen in the MExV and the LTHX can start acting as a heat source.

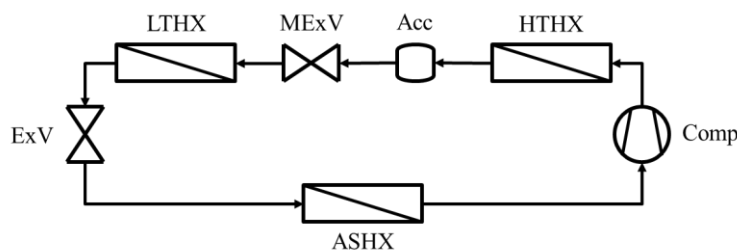


Figure 4: Schematic of the serial connection of two heat sinks for AR including the MExV.

## 3. Results & Discussion

In this section, the results of the investigations are demonstrated. First, the exemplary experimental results are shown, followed by the simulative study of possible solutions. In total, six different evaluations are conducted: (1) the experimental investigation, (2) the simplified simulation model to show similarity with the experiment, the solutions to (3) add more refrigerant and (4-6) the three MExV openings. The results are discussed in the last section.

### 3.1. Experimental investigation

The experimental investigation (investigation number 1) of AR is shown in Fig. 5. AR is activated at 13:30. After an initial stabilization phase of both refrigerant and hydraulic loops, a stable period establishes between 14:15 and 15:00. This is due to the ice in the ice storage melting, allowing stable conditions in the LTHX with

inlet temperatures of around 14 °C. After this stable operation, the water is heated, and the heating powers start changing again. The DSHP is deactivated just before 15:15. A half hour period ( $t_{exp}$ ) during the stable operation period is selected for the analysis.

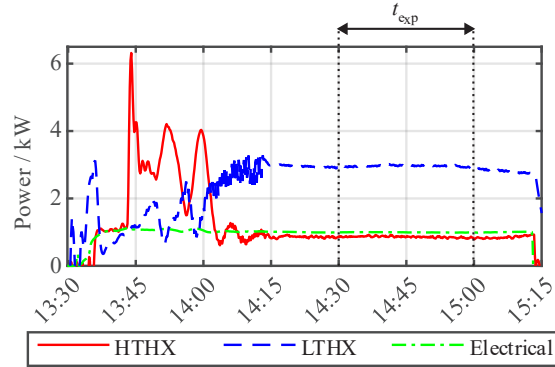


Figure 5: Experimental analysis of AR;  $t_{exp}$  shows a period of constant AR during the ice-melting phase.

Most of the heating power is transferred to the LTHX connected to the ice storage (2.96 kW). The heating power to the HTHX is calculated across the water loop (0.86 kW) is even lower than the consumed electrical energy at 0.99 kW, yielding a COP of about 0.86 and a HTHX heating power ratio of only 23 %. While  $COP_{tot}$  is high at 3.84, this can be considered problematic if more heating power is required by the HTHX, e.g., for space heating purposes.

As the next step in the investigation, the experimental AR process is shown in the pressure-enthalpy diagram of the used refrigerant R454B shown in Fig. 6. The six numbers indicate the relevant process points: 1: ASHX outlet / compressor inlet, 2: compressor outlet / HTHX inlet, 3: HTHX outlet, 4: LTHX inlet, 5: LTHX outlet, 6: ASHX inlet. Constant refrigerant mass flow is assumed, neglecting fluctuations. Points 3 and 4 are approximated, as temperature measurements are not meaningful in or close to the two-phase region and were just placed on the saturation line. The ratio of enthalpy changes in the HTHX (2-3) and LTHX (4-5) roughly correspond to the ratio in heating power in these two heat exchangers. It is still obvious that all / most of the phase change happens in the LTHX and hotgas cooling occurs in the HTHX. This is due to the significant lower temperatures in the LTHX, due to which the refrigerant condensation moves to the LTHX.

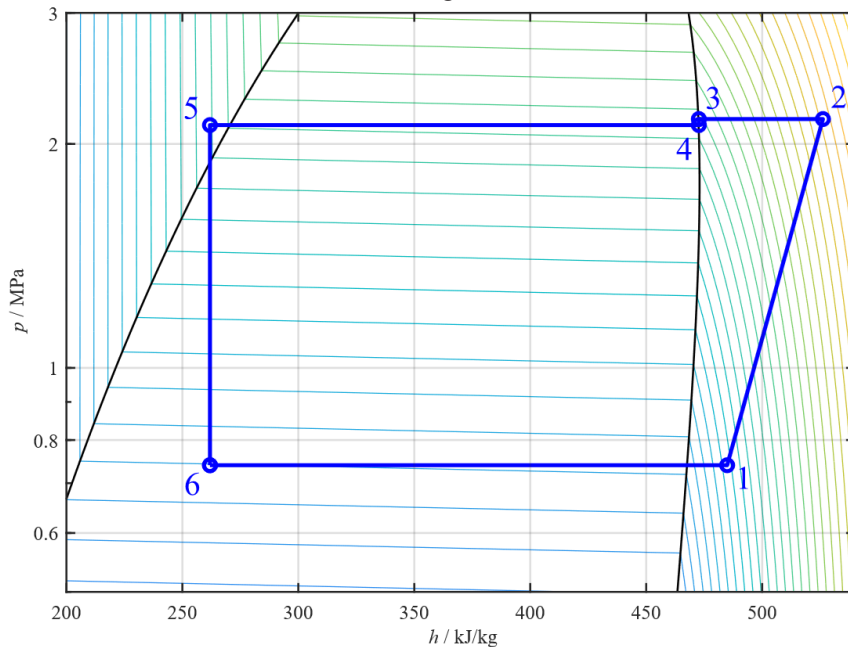


Figure 6: Pressure-enthalpy diagram of AR experimental test.

### 3.2. Similarity of the simulation model

The investigation number 2 tries to reproduce the experiment in the physical Simscape simulation. Boundary conditions are taken from the experiment: The inlet air temperature is 7 °C, the water inlet

temperature 40 °C and the brine inlet temperature 14 °C. Initial conditions of the refrigerant in the simulation are adjusted to represent the experimental results. These results are displayed in the pressure enthalpy diagram in Fig. 7, indicating similarity without detailed validation of the cycle. The general shape of the experimental investigation is reproduced with a few limitations of the simulation:

- The evaporation pressure is significantly lower than in the experiment. This implies a too small ASHX or can be the effect of the internal heat exchanger;
- The superheating at evaporator outlet (at point 1) in the simulation is significantly closer to the desired setpoint of 5.3 K compared to the experiment, where fluctuations in the expansion valve affect the mean superheating (as described already in Reum et al. [12]);
- The compression is assumed isentropic;
- Pressure losses in and between HTHX and LTHX are neglected;
- The HTHX outlet is further in the two-phase region, indicating part-condensation in the HTHX;
- The LTHX outlet is not subcooled, possibly due to a too small LTHX or not exactly fitting refrigerant charge.

Still, the simulation demonstrates a general similarity in the qualitative shape as well as in the powers transferred, e.g., the electrical power in the experiment is 0.99kW and 0.79 kW in the simulation. The difference is explainable by the neglected isentropic, mechanical and inverter efficiencies. The simulation is therefore deemed appropriate for investigating the possible solutions to change the power ratios between HTHX and LTHX as described above.

In this simulation – investigation number 2 –, the heating power in the HTHX is 1.05 kW, to the LTHX 3.86 kW and the electrical power to the compressor is 0.79 kW yielding a heating power ratio of 21 %, like the experiment. COP is 1.33 and COP<sub>tot</sub> is 6.21, both increased in comparison to the experiment due to the simplifications mentioned above.

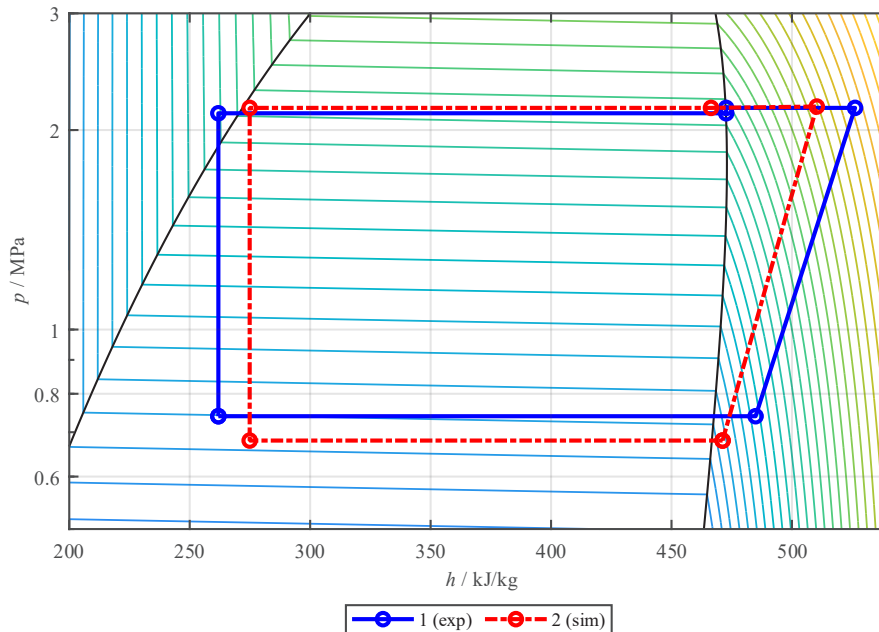


Figure 7: Pressure-enthalpy diagram of AR experiment compared to the similarity simulation. Investigations 1: experimental analysis (blue) and 2: reproduction with a simplified simulation model (red) are shown.

### 3.3. Simulative investigation of the solutions to the condensation shift

Two possible solutions to shift the ratio between the heating power split between HTHX and LTHX are investigated: First, flooding the LTHX with refrigerant is investigated by increasing the refrigerant mass in the cycle. Second, the MEXV is introduced (while not increasing the refrigerant mass in the cycle) and three different openings are tested.

#### 3.3.1. Adding more refrigerant

The first possible solution is adding refrigerant into the cycle to overcharge the cycle. This generally leads to increased condensation pressures and subcooling – in the current setup with the HTHX and LTHX in series, flooding of the LTHX is expected: Since the refrigerant amount in the evaporator is controlled by the expansion

valve, additional refrigerant has to end up in the high-pressure branch of the cycle. This leads to additional condensation in the HTHX and therefore more heating power [14].

This effect is investigated by increasing the initial liquid ratio of refrigerant inside the high-pressure side of the cycle. In this case, initial vapor quality is changed: (a) in the LTHX from 0.25 to 0, (b) in the accumulator from 0.2 to 0.75, and (c) in the HTHX from 1 to 0.2. The values are adjusted incrementally until condensation mostly happens in the HTX. The effect is shown in the pressure enthalpy diagram in Fig. 8. If successfully shifts more heating power from the LTHX to the HTHX, but simultaneously increases the compressor power: Here, the HTHX receives 4.32 kW, the LTHX 1.00 kW of heating power. The electrical power to the compressor is increased to 0.86 kW. As expected, the HTHX heating power is significantly increased over the LTHX at the cost of increased electrical consumption due to increased condensation pressure. While  $COP_{tot}$  at 6.16 remains similarly high as in the similarity simulation, the traditional COP increases to 5.00.

### 3.3.2. Adding the MExV

The second solution to change the heating power ratios between HTHX and LTHX is by adding the MExV and varying its opening. No additional refrigerant is needed and the MExV is expected to add a significant pressure drop between the HTHX and the LTHX, therefore changing the temperature gradient between LTHX and the brine cycle and thus changing the heating power. This accordingly changes the heating power available to the HTHX. The same expansion valve as used for the ExV to control the evaporator superheating is added between the HTHX (more specifically: the accumulator) and the LTHX. Three different openings of the MExV are tested: The ExV in investigation 2 showed an opening of about 0.02 mm. 2-, 5- and 20-times multiples of this value are tested for the MExV opening to allow the ExV to control the superheating of the evaporator still effectively.

The results show that high opening percentages of the MExV yield comparable results to the simulation with no MExV at all, which decreasing the opening of the MExV incrementally increases the HTHX heating power at the cost of lower LTHX power and higher electrical consumption. The results in the pressure enthalpy diagram are shown in Fig. 8. For example, investigation 4 (MExV 20 times the initial ExV opening) yields a HTHX heating power of 1.08 kW and 3.81 kW at the LTHX at 0.79 kW of electrical power. Investigation 6 (MExV 2 times the initial ExV opening) yields a HTHX heating power of 3.47 kW and 1.31 kW at the LTHX at 0.85 kW of electrical power. Investigation 5 (MExV 5 times the initial ExV opening) yields values in between. While the effects on the evaporation pressure are low, the condensation pressure increases systematically with lower MExV openings. While  $COP_{tot}$  generally decreases with closing the MExV due to the increasing condensation pressure (to transfer heat to the HTHX), the COP increases due to the shift of the condensation into the HTHX.

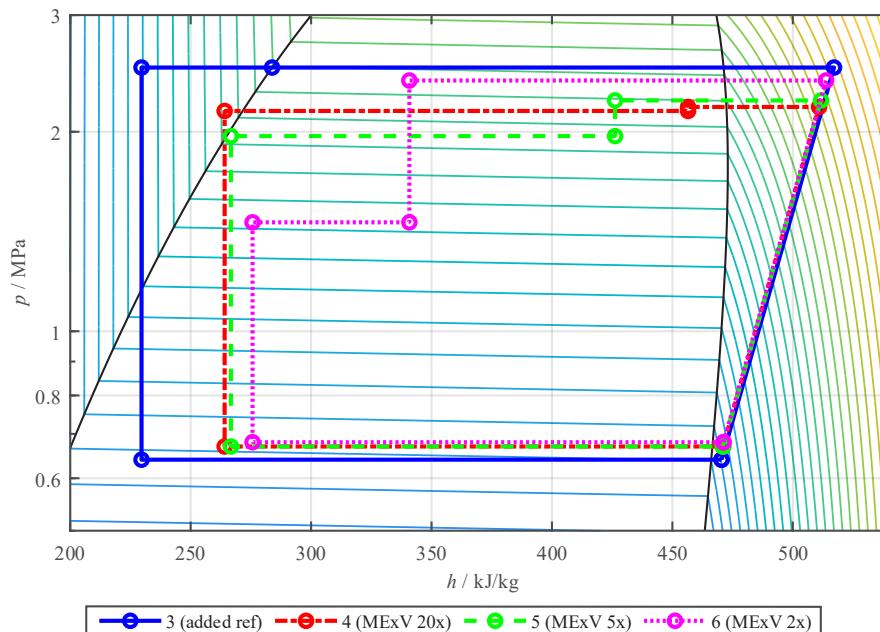


Figure 8: Pressure-enthalpy diagram of AR simulations demonstrating the different solutions: (3) Adding more refrigerant (blue) and (4-6) three different openings of the MExV (red, green, purple).

### 3.4. Discussion

The results of all six investigations are collected in Table 1, calculated with the equations (1) to (5). The experiment falls short of in terms of the COPs due to the simplified simulation models, neglecting heat, pressure, and compressor losses. Absolute values and higher precision reproduction of physical behavior require validation of the refrigerant cycle simulation and needs to be conducted in future studies for more in-detail investigations of this process.

The COP of below 1 is reasonable for the experimental reference case, since most of the heat in this reverse Carnot cycle is delivered to the LTHX and should not be confused with  $COP_{tot}$ , which includes all heat delivered. While the highest COPs for heating in the HTHX are achieved with adding more refrigerant, the highest total COP including heating power to the HTHX and the LTHX is achieved in the original simulation. However, most of the heating power was originally transferred to the LTHX, indicated by a low ratio value of 21 % compared to 81 % in the simulation with added refrigerant. The simulations with the added MExV show a wide range of heating power ratios to be realized, between 22 to 73 %. Wider ranges of MExV openings should allow an even more extended range for this split ratio.

Table 1: Results table of experiment and simulations

| No. | Name                                           | $\dot{Q}_{heat}$<br>kW | $\dot{Q}_{reg}$<br>kW | r<br>% | $P_{el}$<br>kW | COP<br>- | $COP_{tot}$<br>- |
|-----|------------------------------------------------|------------------------|-----------------------|--------|----------------|----------|------------------|
| 1   | Experiment                                     | 0.86                   | 2.96                  | 23     | 0.99           | 0.86     | 3.84             |
| 2   | Simulation of original experiment (simplified) | 1.05                   | 3.86                  | 21     | 0.79           | 1.33     | 6.21             |
| 3   | Simulation: More refrigerant                   | 4.32                   | 1.00                  | 81     | 0.86           | 5.00     | 6.16             |
| 4   | Simulation: MExV $\times 20$                   | 1.08                   | 3.81                  | 22     | 0.79           | 1.36     | 6.18             |
| 5   | Simulation: MExV $\times 5$                    | 1.70                   | 3.17                  | 35     | 0.81           | 2.10     | 6.03             |
| 6   | Simulation: MExV $\times 2$                    | 3.47                   | 1.31                  | 73     | 0.85           | 4.06     | 5.59             |

Further investigations of the COPs in the different simulations are shown in Fig 9. The figure implies the necessary prioritization: Adding more refrigerant to the refrigerant cycle shows the highest COP for heating purposes in the HTHX and only slightly lowered  $COP_{tot}$  compared to the original simulation. Adding more refrigerant also shows the highest total heating power to HTHX and LTHX. However, these benefits come at a loss of flexibility: It is not easily possible to shift more heating power to the LTHX anymore. Varying the temperatures of the boundary conditions – ambient temperature, HTHX and LTHX inlet temperature levels – might change these ratios as well, but further studies would need to be conducted for this purpose.

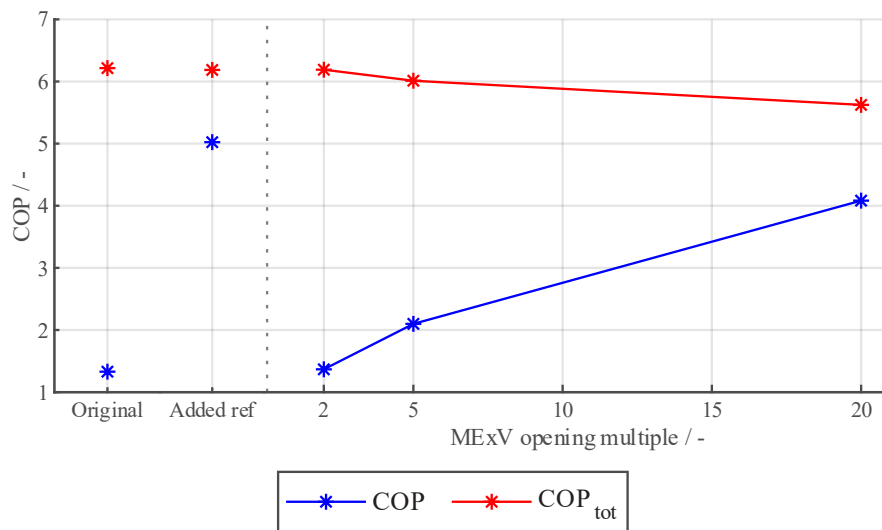


Figure 9: COP and  $COP_{tot}$  at different simulations. The original measurements show investigation 2, Added ref shows investigation 3 and the MExV opening multiples show investigations 4 to 6.

Adding the MExV increases this flexibility in the heating power ratio to HTHX and LTHX. Varying the opening of the MExV affects this heating power ratio and allows highly flexible control depending on the

requirements of the heat delivery – whether higher heating power is required in the HTHX or the LTHX. However, this flexibility comes at the cost of efficiency: Both COP and  $COP_{tot}$  decrease compared to the simulation with added refrigerant and  $COP_{tot}$  also to the original simulation. This correlates with the expectation since the pressure drop, due to the MExV, lowers the refrigerant temperature in the LTHX and therefore the heat transfer. This reduces possible subcooling power in the LTHX as seen in Fig. 8. This effect reduced the total heating power transferred in the cycle even when the heating power (to the HTHX) increased and needs to be considered in practice.

This interconnection including the MExV promises highest flexibility and adaptability to allow AR in a DSHP or similar use cases, as displayed by the high range of heating power ratios in Table 1. In the original case, the HTHX is supposed to supply heating power to the building, while the LTHX is supposed to be used to regenerate a GSHX. In different scenarios, the heating power ratio needs to be flexibly adjusted: In winter, high HTHX heating powers are still required to heat the building and only the remaining heating power can be used in the LTHX for regeneration. However, in autumn and spring, higher shares of LTHX heating power can be used and only low HTHX heating power is required by the building. In total, this opens the flexible option to reduce GSHX sizes in DSHPs by including variable regeneration power during AR into the control strategy. These proper control strategies need to be developed to automatize AR operation with optimal efficiency and reasonable limits and safety requirements.

#### 4. Conclusion

AR of a GSHX with parallel heating operation can be done utilizing two separate heat exchangers in series on the high-pressure side of a refrigerant cycle. However, with a high temperature gradient between these two heat exchangers – HTHX and LTHX – and a refrigerant mass in the cycle suitable for DSHP operation, most of the heat is delivered to the LTHX due to the beneficial temperature gradient to the low brine temperature cycle. Only about 23 % of the total heating power is delivered to the HTHX. While this can be suitable in some periods (e.g., when high AR heating power is required in the LTHX and only low heating power to the heating system in HTHX), this limits flexibility and does not allow increase in HTHX heating power. Accordingly, two possible solutions are successfully investigated with an unvalidated refrigerant cycle simulation, which does, however, show similar behavior as the experiment at the same boundary conditions.

First, adding more refrigerant increases effectiveness by increasing total heating power delivered to the HTHX and LTHX as well as the efficiency in both the COP and  $COP_{tot}$ , which includes the total heating power to both HTHX and LTHX. In this operation mode, more than 80 % of the total heating power is delivered to the HTHX rather than the LTHX. However, this comes at the cost of a) having to carefully adjust the refrigerant mass of the cycle for this specific operation mode and b) no improved flexibility in AR. In a full refrigerant cycle, this might even impact the regular DSHP operation for heating purposes. This possible effect and solutions still need to be investigated.

Second, the addition of MExV increases the flexibility of the operation mode AR significantly, allowing the ratio between HTHX and LTHX heating powers to be controlled more freely. The ratios between heating power to the HTHX and the LTHX can be varied from 22 to 73 %, possibly higher with wider ranges of MExV openings. The simulations also show a reduction in total heating power and  $COP_{tot}$  compared to less flexible systems (like with low and high refrigerant charge). The flexibility therefore comes at a cost of effectiveness and efficiency. Still, for most real use cases like AR in a DSHP, this flexibility might be beneficial in terms of controllability of GSHX temperatures, system-wide efficiency and improving durability by ensuring more consistent operation e.g., with fewer startups. Verification with an experimental setup needs to be conducted. Also, to investigate a use case in densely populated areas, a proper control logic needs to be developed and system simulations to be conducted to further investigate the benefits and disadvantages of this interconnection in terms of efficiency and GSHX sizing.

The interconnection can also be applied to other use cases: Whenever heat rejection at two different temperature levels is required, this system can be integrated to incorporate this operation into one single heat pump with high total efficiencies. These possibilities at more temperature levels need to be investigated further.

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