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## Experimental results of an absorption heat transformer

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### Abstract

A single effect absorption heat transformer can be understood as a single effect absorption heat pump working in reverse, with absorber and evaporator operating at high pressure and the condenser and generator at low pressure. In an absorption heat transformer, the heat input to evaporator and generator (driving heat) is fed at an intermediate temperature level and the heat release takes place at a higher temperature level (revaluated heat) at the absorber and at a lower level at the condenser (rejected heat). One suitable application of absorption heat transformers is the recovery of heat from industrial processes. In the EU funded Indus3Es research project a heat transformer is developed for use in an oil refinery.

At the laboratories of TU Berlin a prototype of an absorption heat transformer has been measured with driving heat temperatures ranging between 60 and 100°C and revaluated heat temperatures between 80° and 130°C. The temperature for heat rejection to the ambient has been varied between 25 and 50°C. The obtained thermal COP (ratio between revaluated heat and heat input) is in the order of 0.45. The capacity of the plant is about 20-40 kW of revaluated heat. The test plant in the refinery will have a capacity of 200 kW.

The measurements have been compared with thermodynamic cycle simulations. The agreement between simulations and measurements has been found to be dependent on the operation conditions. The comparison revealed experimental pitfalls, such as malfunctions of flow meters, as well as non-realistic assumptions.

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*Keywords:* heat recovery; heat transformer; heat pump type II

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### 1. Introduction

An absorption heat transformer (AHT) is a device able to split a heat flow at an intermediate temperature level in two heat flows, one at a higher (revaluated) temperature level and another one at a lower temperature level (rejection heat) (see Figure 1, left). Such a device is also denominated type II absorption heat pump or booster heat pump. Absorption heat transformers are especially suitable for heat recovery from industrial processes, its main advantage being the capacity to upgrade to a usable level the temperature of waste heat streams using only negligible quantities of electrical energy and no additional primary energy. Within the Indus3Es project a prototype for a heat transformer system is being developed. In a first stage, a prototype developed by TU Berlin in the project "Efficiency and reliability of energy systems in urban districts with seasonal energy storage in aquifers" founded by the Federal Ministry of Economic Affairs and Energy of Germany able to work as a heat pump or heat transformer [1] has been investigated experimentally, and its results taken as a reference for new developments. In this paper the main obtained results are presented.

A recent review [2] summarizes a significant part of the research in the field of AHT in the last years, including single, double and triple effects devices. Mostly theoretical and simulative studies are found, experimental results of AHTs are seldom presented. Results of laboratory measurements of a small scale single effect device (2 kW nominal capacity, COPs around 0.2) can be found in [3]. A bigger prototype coupled with a distillation system have been measured achieving gross temperature lifts around 20K and COPs of up to 0.38 [4]. First results of an industry pilot plant in China (around 5MW revaluated heat) revaluating heat 20 to 30 K using 95°C hot water presented very promising COP values up to 0.47 [5]. The most comprehensive presentation of experimental results presents the performance of a single effect LiBr AHT with temperature variations around nominal conditions, with a heat output around 100 kW using 85°C hot water to produce 120°C steam [6].

The laboratory measurements at TU Berlin presented in this work investigate the effects of changing temperature levels. Variations of temperature lift and thrust are studied separately and presented as a function of the so called characteristic temperature difference. In addition, the effects of changing the solution have been also investigated. Overall heat transfer coefficients have been calculated for each heat exchanger and will be taken as a reference for the evaluation of the improvements achieved with the optimized heat exchangers being currently under development within the Indus3Es project. Finally, the measurements results are compared with simulations using a recently presented thermodynamic model [7].

## 2. Prototype and test rig

The prototype is a so-called absorption heat converter able to work either as a heat pump/absorption chiller or as a heat transformer [1] and uses H<sub>2</sub>O/LiBr as its working fluid. The prototype has been derived from absorption chillers designs from TU Berlin [8] with same tube bundles and heat transfer area at all four main heat exchangers, in order to investigate the differences in performance for different operation modes.

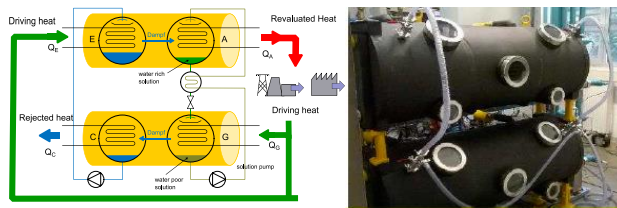


Fig. 1. Scheme and view of the measured AHT prototype

### 3. Methodology

Experimental measurements at the laboratory of TU Berlin have been made using the test rig presented at the left side of figure 2. The test rig feeds the absorption heat transformer with constant external heat carrier (pressurized water) flow rates and temperatures at the inlet of the four main heat exchangers.

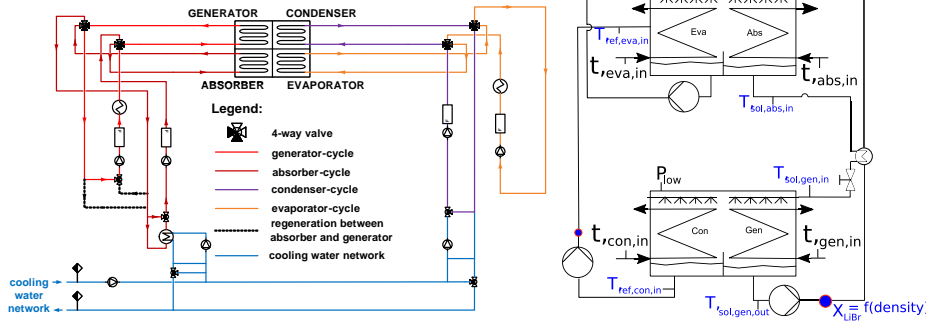


Fig. 2. Test rig (left) and relevant measured values at AHT (right)

For the evaluation of the AHT performance the temperature of the working solution ( $\text{H}_2\text{O}/\text{LiBr}$ ) and refrigerant entering and leaving each heat exchanger has been measured. In addition, pressure sensors are used at both higher and lower pressure vessels. The mass flow rate and density of the refrigerant and solution after the corresponding pumps is measured using Coriolis meters. The LiBr mass fraction of the concentrated solution can be determined experimentally from density and temperature.

All heat flows are calculated using the external fluid temperature sensors  $t_{x,in}$  and  $t_{x,out}$  and the external volumetric flow rate  $v_x$ . Density ( $\rho_{fluid}$ ) and  $c_p$  are calculated using the CoolProp library [9]. For each heat exchanger, we calculate the heat flow rate with:

$$Q_x = v_x \cdot \rho_{fluid} \cdot c_{p,fluid} \cdot (t_{x,in} - t_{x,out}) \quad (1)$$

(x equals either *abs*, *gen*, *cond* or *eva*)

For the determination of the thermal performance the thermal coefficient of performance or  $\text{COP}_{th}$  is used. It is the ratio between useful or revaluated heat at the absorber  $Q_{abs}$  and used driving heat at the generator  $Q_{gen}$  and evaporator  $Q_{eva}$ .

$$\text{COP}_{th} = \frac{Q_{abs}}{Q_{gen} + Q_{eva}} \quad (2)$$

The measurement uncertainties have been calculated using the methods described in the Guide to the expression of the Uncertainty in Measurement, GUM [10].

For all tests the external fluid flow rates have been kept constant at a nominal value. Either one or more of the external fluid inlet temperatures or the solution flow rate have been varied, keeping all other values constant at its nominal value. The nominal values and the variations performed are included in table 1.

Table 1 Experimental conditions

Measurement uncertainty	nominal	minimum	maximum
External mass flow rate at each heat exchanger, $\text{m}^3/\text{h}$	5	5	5
Solution mass flow rate, $\text{l/h}$	400	200	600
External fluid absorber inlet temperature, $t_{2,in}$ , $^{\circ}\text{C}$ (Revaluated heat)	130	105	135

External fluid generator & evaporator inlet temperature, $t_{1,in}$ , °C (Driving heat)	85	60	100
External fluid condenser inlet temperature, $t_{0,in}$ , °C (Rejection heat)	25	20	50

## 4. Measurement results and analysis

### 4.1. External temperature variations

Figure 3 shows the result of one of these series with constant driving temperatures (generator and evaporator inlet temperatures at 85, 90 or 95°C) and constant cooling water temperature (condenser inlet temperature at 25°C). The solution and external flow rates have been kept at their nominal values. Measurements with driving heat inlet temperatures of 90°C and 95°C have been made for only one point with an absorber inlet temperature of 130°C.

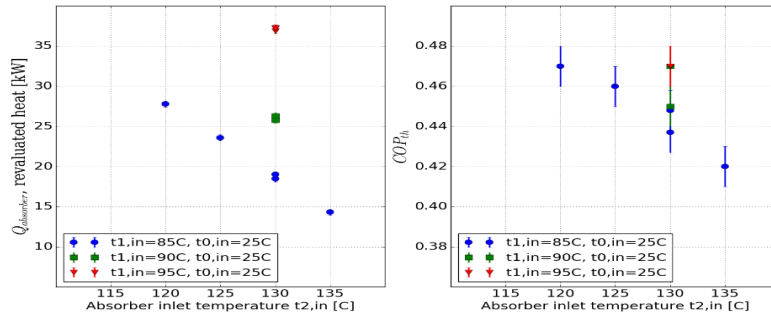


Fig. 3. Variations of absorber inlet temperatures (revaluated heat temperatures) for constant evaporator, generator and condenser external heat carrier inlet temperatures

An almost linear relation can be observed between revaluated heat temperature and obtained revaluated (absorber) heat flow rate if all other temperature levels are kept constant for a driving inlet temperature of 85°C. The absorber heat flow measured with an absorber inlet temperature of 120°C is 27 kW. It is reduced linearly down to 14 kW if this inlet temperature is increased up to 135°C (increasing the temperature lift from 35 to 50 K). The thermal COP of the heat transformer with absorber inlet temperature of 120°C is around 0.47 and becomes smaller (also in an almost linear way) if the absorber inlet temperature increases, being around 0.42 for an absorber inlet temperature of 135°C. Both revaluated heat flow and COP are reduced if the temperature lift (difference between absorber and evaporator/generator temperature) increases and the temperature thrust (difference between evaporator/generator and condenser temperature) are kept constant.

The only measurement performed with a driving temperature of 95°C and a cooling water temperature of 25°C is also shown in figure 2, for a revaluated heat temperature over 130°C. For this measurement a revaluated heat flow rate of 37 kW has been measured at the absorber with a thermal COP around 0.47.

For constant driving heat inlet temperatures of 85 and 90°C and a revaluated heat flow inlet temperature of 130°C the cooling water temperatures has been changed between 20 and 35°C. The results of these variations are presented in figure 4. For both variations we find again a linear relation between the cooling water temperature and the absorber heat flow, the latter increasing its value from 14 to 30 kW with decreasing cooling water temperature. The COP values increase as well with decreasing cooling temperature, and reach a maximum around 0.47 for a temperature lift of 40 K and temperature thrust of 70 K, and a minimum of 0.38 for a temperature lift of 45 K and a temperature thrust of 50 K.

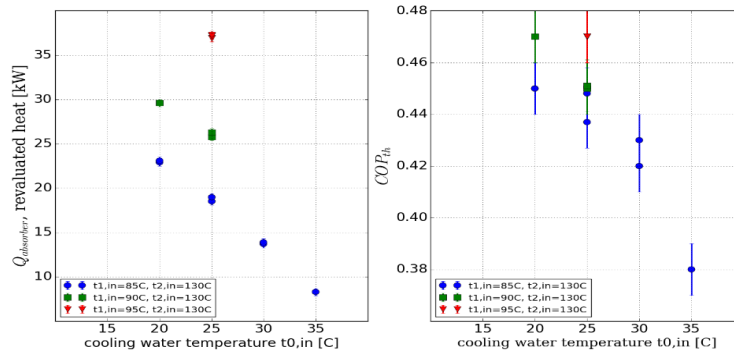


Fig. 4. Variations of cooling water inlet temperatures for constant absorber, generator and evaporator external fluid inlet temperatures

The results presented in figures 3 and 4 show how the revaluated heat and the COP differs depending on the lower, intermediate and higher temperature levels. For the representation of all these different results the use characteristic temperature difference  $\Delta\Delta t$  is convenient. The characteristic temperature difference relates all mean temperatures of the external heat carriers and is defined as:

$$\Delta\Delta t = (t_G - t_A) - R \cdot (t_C - t_E) = R \cdot (t_E - t_C) - (t_A - t_{EG}) \quad (3)$$

with  $t_G$ ,  $t_A$ ,  $t_C$  and  $t_E$  being the mean values of the external heat carrier temperatures at generator, absorber, condenser and evaporator respectively.

$R$  is the so-called Dühring factor, a property of the given fluid working couple, that can be considered as a constant for the operation range of the heat pumps (circa 1.15 for Libr/H<sub>2</sub>O).

The characteristic equation method has been extensively used in the past [11][12] for characterization of absorption chillers, and its use for the characterization of heat transformer has been recently presented in detail [1].

Figure 5 presents all temperature variations presented in figures 3 and 4 and six more points (3 of them with cooling water temperatures of 50°C made prior to a test rig modification). The (green) squares are used for a problematic measurement point with  $t_{2,in}=115^{\circ}\text{C}$ ,  $t_{1,in}=100^{\circ}\text{C}$  and  $t_{0,in}=50^{\circ}\text{C}$  with unexpected values

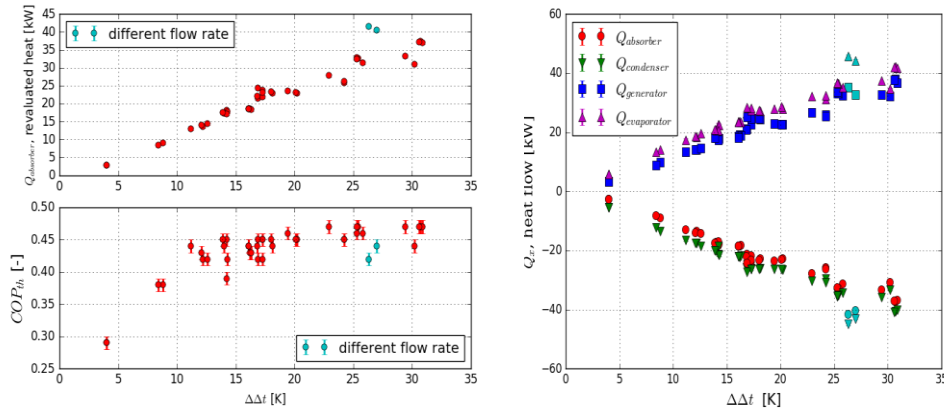


Fig. 5. Heat flow rate and  $\text{COP}_{\text{th}}$  as a function of the characteristic temperature difference

As figure 5 shows, the relationship between  $\Delta\Delta t$  and heat flows of absorber, generator, condenser and evaporator are almost linear. The heat transformer is able to work revaluating heat between 4 and 40 kW, and showing COPs over 0.38 for revaluated heat loads over 10 kW.

The evaporator heat flows are for all measurements around 1.5 to 2 kW higher than the generator ones, and the condenser heat flows around 0.6 kW higher than the absorber ones, so the overall heat losses are constantly around 2 kW. The resulting COPs are higher than 0.42 for  $\Delta\Delta t$  values higher than 10, which corresponds to a revaluated heat load larger than 12 kW. Within a variation from 12 to about 40kW the  $\text{COP}_{\text{th}}$  almost always was larger than 0.4. For the smallest measurable absorber heat flow rate of 2kW, with a  $\Delta\Delta t$  value of 3.7, a COP of 0.3 has been measured. More measurements are needed for the range between  $\Delta\Delta t = 4$  and  $\Delta\Delta t = 12$ , but it seems that in this region the COP increase almost linearly with  $\Delta\Delta t$ . For  $\Delta\Delta t$  values over 12 the improvement of COP reduces with increasing  $\Delta\Delta t$  values, and seems to converge asymptotically to 0.47-0.48 as expected from fundamental theory, while the absorber heat flow rate further increases with increasing  $\Delta\Delta t$ , of course. of the problematic point mentioned before (green squares with  $\Delta\Delta t$  around 26) show a bigger deviation from the trend line than the other ones.

Looking at the right side of figure 4 it can be seen that the difference between evaporator and generator heat flows is much higher for the two measurement with different flow rates than for all others. After detailed investigations it was shown that the flow rates measurements for these points was not right due to a malfunction of the flow meters caused by dirt accumulated during the test rig adaptation. These unfeasible points have been detected with the help of the characteristic equation method. For the simulation these points are not used.

#### 4.2. Solution flow rate variations

In order to find the most appropriate solution flow rate for the operation of the AHT a measurement series have been made with constant volumetric flow rates (5 m<sup>3</sup>/h) and inlet temperatures (130°C at absorber, 85°C at evaporator and desorber and 25°C at condenser) of the external heat carriers entering the absorption heat transformer. Around the nominal value of 400 l/h the solution flow rate has been varied in a range between 200 and 600 l/h. The corresponding variations of absorber heat flow and COP are presented in figure 6.

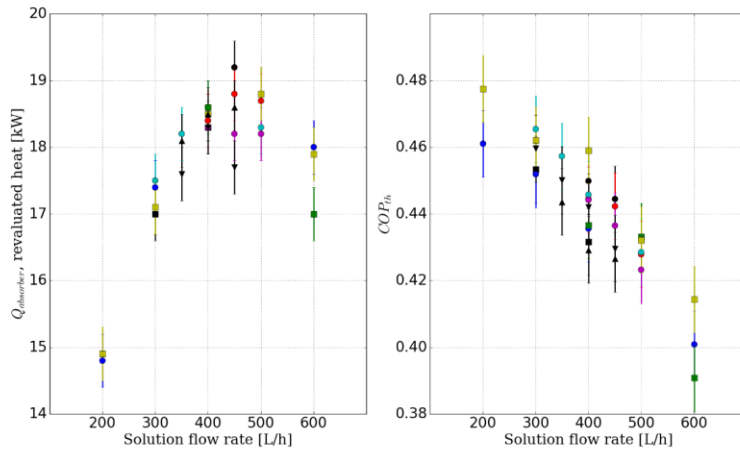


Fig. 6: Revaluated heat flow and COP for solution flow rate variations

The different colors and symbols employed in figure 6 represent different measurement days.

The results of figure 6 show that the absorber heat flow increases if the solution flow rate increases from 200 to 400 l/h. For solution flow rates between 400 and 500 l/h no common trend can be identified for the different measurements, and there is no clear maximum. The differences between the measured absorber heat flows for all these measurements are very small, and can be also been caused by different random effects. For all measurements with a solution flow rate of 600 l/h the absorber heat flow is smaller than those measured with 500 l/h. The COP monotonously decreases by increasing the solution mass flow rate.

A similar behavior has been observed experimentally in the past for variations of the evaporator heat flow and COP with changing solution flow rate in an absorption chiller [13].

#### 4.3. Heat transfer coefficients determination

The so-called heat transmissivity or UA value of each main heat exchanger can be calculated for each experiment using measured heat flows, solution or refrigerant temperatures entering and leaving the heat exchanger and temperatures of the heat carrier. The values obtained from the external temperature variation tests are presented in figure 7, using same symbols as those used in figure 5.

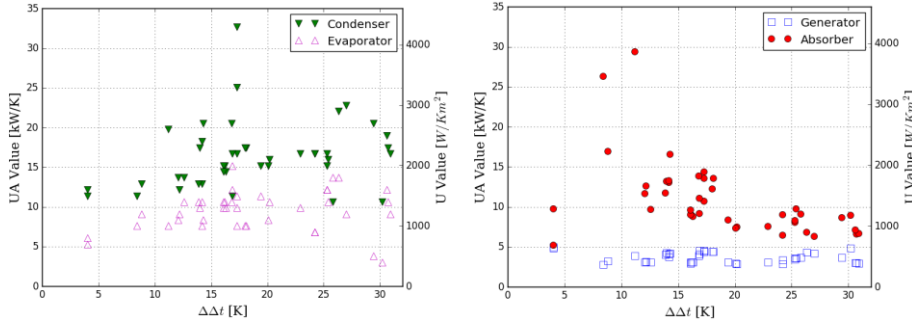


Fig. 7. UA and U values

The UA values for condenser and evaporator are obtained using the externally determined heat flows, the external temperatures ( $t_{eva}$  and  $t_{con}$  from Figure 2, and the saturation temperatures calculated with the measured pressures  $P_{high}$  and  $P_{low}$ ). The values for the condenser are mostly in a range between 13 and 20 kW/K. It seems that for values of  $\Delta\Delta t < 15$  most of the values are in range 13–15 kW/K and for  $\Delta\Delta t > 15$  they are more likely in the range 15–20 kW/K. The UA values obtained for the evaporator are in a range between 5 and 15 kW/K, but most of them can be found in a tighter range between 8 and 11 kW/K.

For the determination of the UA value of the generator, the measured external carrier temperatures  $t_{gen}$  and internal solution temperature  $T_{gen,in}$  are used. Additionally the saturation temperature of the solution leaving the generator is calculated with measured  $P_{low}$  and the LiBr concentration obtained with the density measured of the solution leaving the generator.

The UA values obtained for the generator are all in a range between 3 and 4.5 kW/K, with only three outliers around 5. Most of the UA between 3 and 3.5 kW/K have been obtained from measurements with a condenser inlet temperature between 20 and 30°C, and a resulting desorber vessel pressure between 30 and 50 mbar. For measurements with measured pressure levels between 120 and 180 mbar (condenser inlet temperature 45 or 50°C) the UA values are mostly between 4 and 4.5 kW/K. More measurements are needed to conclude if there is any kind of relationship between the desorber UA value and the pressure level.

The UA values for the absorber are much higher than those measured for the generator. However, for the calculation of the absorber UA values it was necessary to make some assumptions that may be doubted: it has been assumed that the solution contacts the absorber tube bundle at a saturation temperature depending on the vessel pressure and solution mass fraction after adiabatic absorption. Additionally, the measured temperature of the solution leaving the absorber was used and not the saturation temperature. This solution is expected to be subcooled referred to the corresponding saturation temperature. By using the measured solution outlet temperature, the UA value will most probably be overestimated as compared with one calculated using the saturation temperature. For these reasons for the simulations later presented in this paper the measured average UA values for each heat exchanger will be considered, but the generator UA value will be used also for the absorber.

On the right axis of figure 7 the overall heat transfer coefficient or U values are scaled considering a heat exchanger surface (estimated from construction drawings) of 7.6 m<sup>2</sup>. This equivalent U value must be carefully considered; the obtained values can be smaller than expected since the reduction in useful heat exchanger area due to partial wetting is not considered for its calculation. The obtained U values for the condenser are in the range between 2000 and 2300 kW/m<sup>2</sup>K, for the evaporator between 900 and 1200 W/m<sup>2</sup>K and for the generator between 400 and 600 W/m<sup>2</sup>K. The estimated U values for the absorber spread out between 900 and 2000 W/m<sup>2</sup>K, but in order to use this value the subcooling effect must be considered by the model.



## 5. Energy balance simulations and comparison with experimental results

The measurement results shown above are compared with a model based on energy and mass balances. This model is written in the programming language Modelica using the modeling tool Dymola. The model is based on the same equation as the Modelica-model shown in [7], but it was rewritten in a more modular way (see figure 8). The absorber was divided in adiabatic absorption and “general” absorption and the generator in flashing and desorption. Furthermore, in this paper the measurement and simulation results of 18 different working point with varying inlet temperature of the external heat carrier was compared, in [7] models written in three different modeling tools was compared with measurement results for one working point.

For the energy and mass balance model following assumptions were taken:

- Constant UA values at the absorber, desorber, condenser, evaporator and solution heat exchanger
- Steady state heat and mass transfer
- Saturated thermodynamic conditions at the solution outlet of the absorber, desorber, adiabatic absorber and flashbox
- Saturated thermodynamic condition at the refrigerant outlet of the evaporator and condenser
- No heat losses to ambient
- Isenthalpic pumps
- Constant mass flow rate of the pumped poor solution leaving the desorber

In addition to the boundary conditions of the experiment (see Table 1) the following parameters were used:

Table 2 Additional simulation conditions

UA Absorber	UA Desorber	UA Evaporator	UA Condenser	UA Solution heat exchanger	Mass flow rate poor solution
3.6 kW/K	3.6 kW/K	9.4 kW/K	15.9 kW/K	1.3 kW/K	650 kg/h

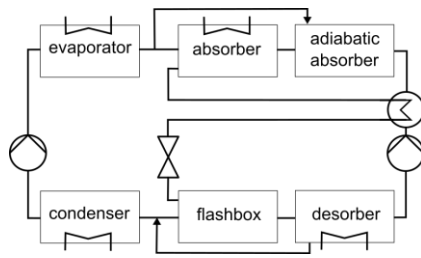


Fig. 8. Flow chart of the model

In figure 9 the results of the heat flows at the absorber, desorber, condenser and evaporator of simulation and experiment are compared. To simulate the 18 different working points, the same varying inlet temperatures and constant volumetric flow rates of the external heat carrier were used. In these figures the arithmetic mean values of the experimental results of the heat flows and coefficients of performance of every experiment for each working point are drawn.

The general behavior of the experimental analyzed AHT in heat flows and coefficient of performance is reproduced by the model. The deviation of the heat flows is around 10% (capacity difference related to the experimental capacity), but near to  $\Delta T = 17$  K there are two working points with a deviation between 20% and 30%. These two working points have high external heat carrier inlet temperatures at the condenser of 45°C and 50°C and large UA values of the desorber between 4 and 4.5 kW/K (see chapter before).

Furthermore, for  $\Delta T < 20$  K the absolute values of the experimentally determined heat flows are bigger than the simulated heat flows. The biggest deviation appears at the evaporator. For  $\Delta T > 20$  K the values of the experimentally determined heat flows are scattered around the simulated heat flows.

The simulated thermal COP (figure 10) is always bigger than the experimentally determined one. The reason for the overestimation of the thermal COP is related to the heat losses neglected in the simulation. In further studies this should be analyzed in detail. The deviation for  $\Delta\Delta t < 10$  K is around 10% related to the simulated thermal COP and for  $\Delta\Delta t > 10$  K smaller than 7 %. The relative deviation of the thermal COP is smaller than the heat flows itself, because the deviation of the heat flows for each working point have the same direction.

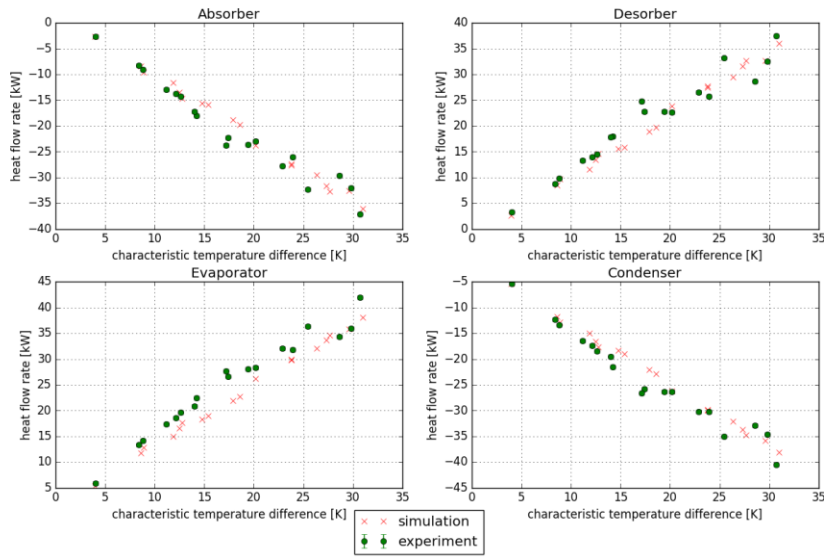


Fig. 9. Comparison of the capacity of the experimental and simulative results

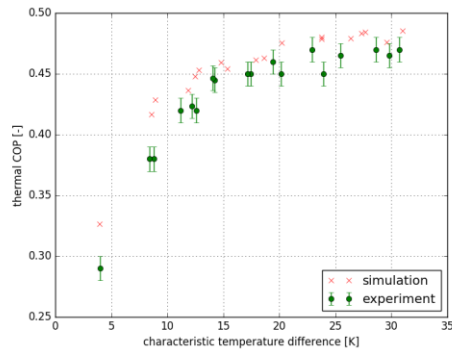


Fig. 10. Coefficient of performance comparison of the experimental and simulative results

## 6. Conclusions

The measured absorber heat flows are between 17 and 40 kW. For the investigated temperature levels with inlet heat temperatures around 90°C and cooling water temperatures around 25°C, it is possible to obtain temperature lifts between 25 and 50 K with thermal COP values that lie mostly over 0.42 and up to 0.48. Both the temperature thrust and lift play a major role in the resulting thermal COP, and the they affect the AHT performance can be represented using the characteristic temperature difference  $\Delta\Delta t$ . The thermal COP falls below 0.4 at  $\Delta\Delta t$  values below 10 K.

The highest value of revaluated heat flow has been obtained using solution mass flow rates between 400 and 500 l/h. The thermal COP has been shown to be the higher the lower the solution flow rate in the considered range between 200 and 600 l/h is.

The obtained experimental results have been validated against an energy and mass balance model. It is shown the general behavior of the experimental device could be reproduced. The deviation of the heat flows of the absorber, generator, evaporator and condenser is typically around 10%, but two working points with high inlet temperatures of the heat carrier at the condenser show deviations between 20 and 30%. The simulated thermal COP for characteristic temperature differences  $\Delta\Delta t < 10$  K is around 10% and for  $\Delta\Delta t > 10$  K less 7% higher than the experimentally determined thermal COP.

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