

THERMODYNAMIC ANALYSIS OF A SMALL-CAPACITY AMMONIA / WATER ABSORPTION HEAT PUMP FOR HEATING AND COOLING

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Abstract: At the Institute of Thermal Engineering a small-capacity ammonia/water absorption heat pumping unit has been developed which is able to operate at cold water temperatures from -10°C to 20°C and cooling water temperatures from 30°C to 50°C . For the generator a standard plate heat exchanger has been used where the rich solution is partly evaporated and the vapour and liquid phase flows in co-current flow through the component. This leads to a thermodynamic disadvantage compared to a counter-current flow component.

In this work two designs of an $\text{NH}_3/\text{H}_2\text{O}$ absorption heat pump, one with co-current flow generator and one with a counter-current flow generator are investigated using both first and second law analysis. Finally the results are compared to each other in order to identify the thermodynamic penalty of the co-current design and to find potential for improvements at different operating conditions.

Key Words: *second law analysis, co-current vs. counter-current generator*

1 INTRODUCTION

In recent years small capacity absorption heat pumps (AHP) attract increasing research interest due to the possible use of renewable energy, e.g. biomass, solar or process heat for cooling applications. But they are also considered to be used for residential heating application where usually the requirements regarding the temperature levels of the heat sources and the heat sink are different. Absorption heat pumping systems for refrigerating and air-conditioning with large cooling capacities (of some 100 kW) are more or less state-of-the-art. With respect to the working pairs mainly ammonia/water is used for applications with evaporation temperatures below 0°C , and water/lithium bromide for air-conditioning applications.

In contrast to large-capacity applications, up to now small-capacity absorption heat pumping systems have difficulties to get into the market due to relatively low efficiencies, control issues, and high first costs. Recently small-capacity absorption chillers with the working pair water/lithium bromide have been developed and are partially available on the market, but only a few installations of small-capacity ammonia/water applications can be found.

At the Institute of Thermal Engineering a small-capacity ammonia/water absorption heat pumping unit for heating and cooling has been developed and tested (Moser et. al., 2007). The heat pump has been designed to operate with cold water temperatures from -10°C to 20°C and cooling water temperatures from 30°C to 50°C . It is indirectly driven by a heat carrier loop (water or thermo-oil) in order to allow the use of different heat sources. For all heat exchangers – including absorber and generator – standard plate heat exchangers

(PHE) have been used. Plate heat exchangers offer very high heat transfer areas at compact dimensions, thus small temperature differences can be achieved. Furthermore plate heat exchangers are off-the-shelf products which offers a high prefabrication standard and compared to special designs a considerable cost reduction is expected. The capacity modulated heat pump uses a variable speed controlled solution pump in order to avoid start and stop losses and the generator temperature is controlled with respect to maximum efficiency (Fernández-Seara and Vázquez; 2001). For purification of the refrigerant a rectification column combined with a dephlegmator which is cooled by the rich solution has been developed.

Using a PHE for the generator can lead to a thermodynamic disadvantage compared to a special designed component. In a conventional generator the liquid solution enters the heat exchanger on the top and leaves the heat exchanger at the bottom and the generated vapour flows upwards in counter-flow to the liquid and leaves the heat exchanger on the top. If the temperature of the generator is higher at the bottom than on the top, the vapour leaves the generator at a lower temperature than the liquid. This can increase the efficiency of the system.

In standard plate heat exchanger the counter-flow of liquid and vapour requires large flow cross sections in order to avoid "flooding" of the heat exchanger. "Flooding" means that the velocity of the vapour becomes too high, resulting in a two-phase flow leaving the heat exchanger on the top. In the presented development the liquid and the generated vapour phase flows in co-current flow through the plate heat exchanger and are separated after the generator in the rectification column.

The aim of this work is to investigate the thermodynamic penalty of a co-current flow in the generator compared to a counter-current flow and to identify potential for improvements of the used design. Therefore a thermodynamic analysis of two different processes has been performed and compared to each other, including both the first and second law approach. According to the first law analysis all forms of energy are convertible to one another and the energy balance as well as the coefficient of performance can be calculated. The second law analysis limits the possible energy conversion and allows a more detailed view to the component losses which occur due to the energy conversion.

2 THERMODYNAMIC ANALYSIS

In Figure 1 the two different heat pump designs are shown where the components are indicated as they would appear in a $\log(p)$ vs. $1/T$ diagram. On the left hand side the Figure 1 shows the realised single-stage process with the co-current generator design. Starting at the absorber (ABS) the rich solution flows via the solution accumulator (SAC) to the solution pump (PUMP) where the pressure is increased to the high pressure level. Further the rich solution flows to the dephlegmator (DEP) where it is heated by the refrigerant at the top of the rectification column (REC) and to the solution heat exchanger (SHX) where it is preheated and possibly partly evaporated by the heat from the poor solution before it enters the generator (GEN). In the generator the rich solution is partly evaporated by the high temperature heat source (Q_{GEN}). The two-phase flow leaves the generator and enters the rectification column where it separates into liquid (poor solution) and vapour phase (refrigerant). The poor solution flows through the solution heat exchanger (SHX) via the solution throttle (STH) to the absorber. In the solution throttle it expands to the low pressure level. The refrigerant flows in the rectification column upwards through the stages in counter flow to the liquid fraction (condensed refrigerant). The vapour leaving the rectification column on the top flows to the dephlegmator (DEP) where it is partly condensed. The condensate flows back to the rectification column and the purified vapour streams to the condenser (CON) where it is totally condensed by rejecting the heat to the cooling water (Q_{CON}). After the condenser the refrigerant flows via the refrigerant accumulator (RAC) to the condensate

pre-cooler (PRC) where it is cooled by the refrigerant leaving the evaporator. After the pre-cooler the refrigerant expands through the refrigerant throttle (RTH) to the low pressure level and enters the two-phase region and flows further to the evaporator. Receiving the heat from the low temperature heat source (Q_{EVA}) the refrigerant evaporates either partly or totally and flows to the condensate pre-cooler where the refrigerant is further evaporated or superheated and flows to the absorber. At the absorber inlet the refrigerant vapour and the poor solution are mixed. The mixture flows through the absorber where the refrigerant is absorbed and the heat is rejected to the cooling water (Q_{ABS}).

On the right hand side a similar process with a counter-current generator design is shown. For the comparison it is assumed that the liquid and the vapour flowing in counter-current flow in the generator are in thermodynamic equilibrium.

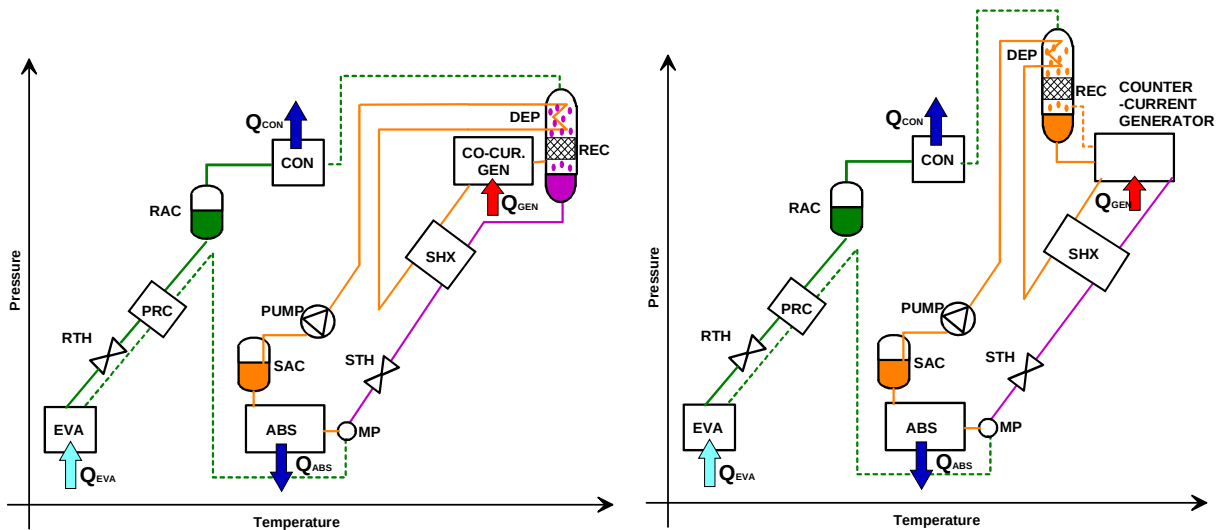


Figure 1 Single stage absorption heat pump with co-current generator design (left) and counter-current generator design (right)

A mathematical model has been developed for both process configurations with the software package EES (2006) in order to study the effects of component behaviors and perform calculation of variations.

2.1 Fluid Property Data

The thermodynamic properties of ammonia/water mixtures were calculated with the procedure „NH3H2O“ (Ibrahim and Klein, 1993), which is implemented in EES.

The external circuits are the cold water circuit for the low temperature heat source, the cooling water circuit for the medium temperature heat sink and the heating water circuit for the high temperature heat source. The specific heat capacities of the external circuits are calculated using Eq. 1

The used constants are given in Table 1. The values for the cold water circuit (brine, 50%_{Mass} Glysantine and 50%_{Mass} Water) are derived from BASF (2001). The values for the cooling water have been determined using the EES-procedure “Water”. For the heating water circuit thermal oil has been considered and the specific heat capacity has been derived from VDI Wärmeatlas (1997) for “BP Transcal N”.

$$c_p = a \cdot T^2 + b \cdot T + c$$

Eq. 1

Table 1: Coefficients for the calculation of c_p , Δh and Δs

Description	a	b	c	validity [K]
Cold Water (brine)	-2.4673E-05	2.0468E-02	-0.5101	243 < T < 383
Cooling Water (water)	1.2505E-05	-7.9884E-03	5.4548	273 < T < 423
Heating Water (thermal oil)	0	3.4131E-03	0.9825	273 < T < 590

Applying Eq. 1 to Eq. 2 the enthalpy difference, and applying Eq. 1 to Eq. 4 the entropy difference of the external circuits can be calculated from T_1 to T_2 (at constant pressure) according to Eq. 3 and Eq. 5 (Lüdecke and Lüdecke, 2000).

$$\Delta h^{1 \rightarrow 2} = \int_{T_2}^{T_1} c_p \cdot dT \quad \text{Eq. 2}$$

$$\Delta h^{1 \rightarrow 2} = \frac{a}{3} \cdot (T_1^3 - T_2^3) + \frac{b}{2} \cdot (T_1^2 - T_2^2) + c \cdot (T_1 - T_2) \quad \text{Eq. 3}$$

$$\Delta s^{1 \rightarrow 2} = \int_{T_2}^{T_1} \frac{c_p}{T} \cdot dT \quad \text{Eq. 4}$$

$$\Delta s^{1 \rightarrow 2} = \frac{a}{2} \cdot (T_1^2 - T_2^2) + b \cdot (T_1 - T_2) + c \cdot \ln\left(\frac{T_1}{T_2}\right) \quad \text{Eq. 5}$$

2.2 First Law Analysis

For the thermodynamic analysis of the process a steady state model has been set up. The model solves energy, overall mass and ammonia mass balances for all components. The left hand side of Figure 2 shows the absorption heat pump cycle for the co-current generator and the labeling of the state points and the right hand side shows the section of the counter-current generator design. Except the generator no other changes have been assumed for the AHP-process.

For the co-current design the liquid separator (LS) is located downstream the generator and separates the refrigerant and the poor solution and for the counter-current design the liquid separator is located upstream the generator and separates the refrigerant and the rich solution. In both cases thermodynamic equilibrium of the vapor and liquid phase has been assumed.

At the absorber inlet a mixing point (MP) has been considered where the refrigerant and the poor solution are adiabatically mixed.

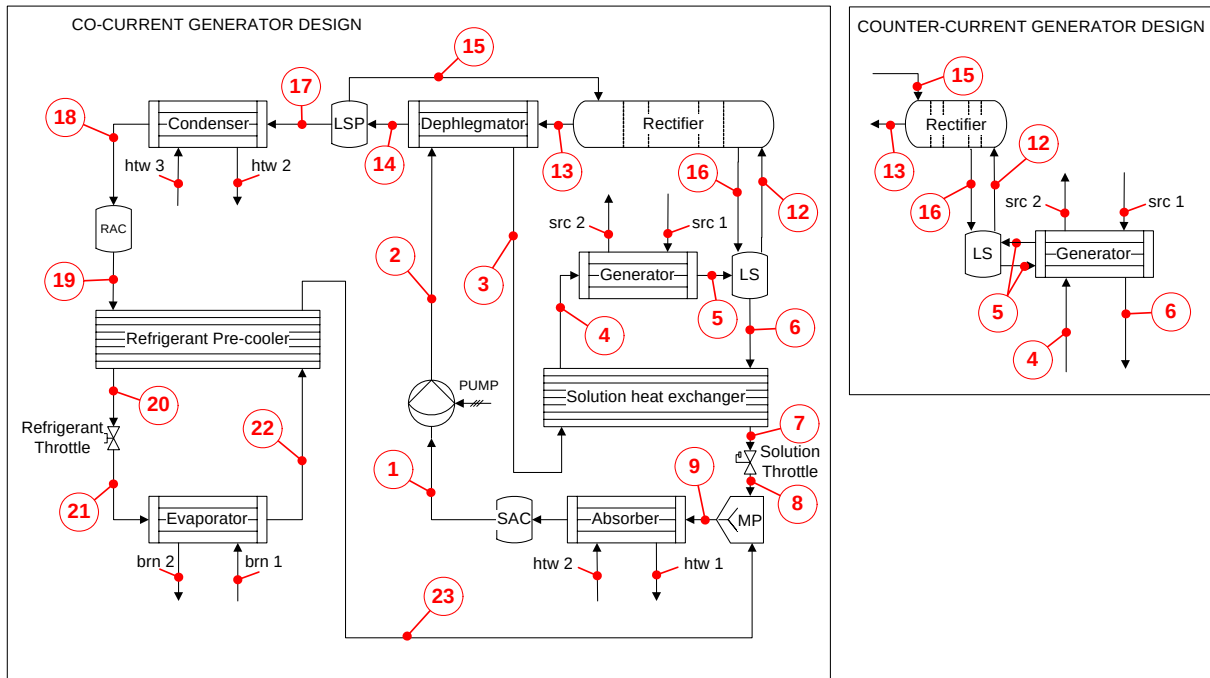


Figure 2: State pints for the absorption heat pump cycle for the co-current generator (left) and the generator section of the counter-current generator design (right)

As an example of the first law analysis the used equations for the overall mass, ammonia mass and energy balances of the co-current generator are given by Eq. 6, Eq. 7 and Eq. 8

$$\dot{m}_4 = \dot{m}_5 \quad \text{Eq. 6}$$

$$\dot{m}_4 \cdot \xi_4 = \dot{m}_5 \cdot \xi_5 \quad \text{Eq. 7}$$

$$\dot{Q}_{\text{GEN}} = \dot{m}_5 \cdot h_5 - \dot{m}_4 \cdot h_4 = \dot{m}_{\text{src}} \cdot (h_{\text{src};1} - h_{\text{src};2}) \quad \text{Eq. 8}$$

The effect of the rectification process on the water fraction of the refrigerant and on the dephlegmation capacity was calculated using the theoretical plate model (ponchon method) acc. to Herold and Pande (1996) and two theoretical plates has been assumed.

The internal heat exchangers efficiencies (ε) were considered to be constant acc. to Eq. 9. Where $\dot{Q}_i$ means the actually transferred capacity and $\dot{Q}_{i,\text{max}}$ the maximum possible capacity determined by the inlet temperatures. Pressure drops and heat losses in components and pipes were neglected.

$$\varepsilon = \frac{\dot{Q}_i}{\dot{Q}_{i;\text{max}}} \quad \text{Eq. 9}$$

In order to compare the results to another, the coefficient of performance (COP) is calculated using Eq. 10 and Eq. 11. The external temperature lift - which is the temperature difference between the low temperature heat source and the medium temperature heat sink - is calculated using the Eq. 12 and the internal temperature lift is calculated using Eq. 13. $\Delta\xi$ means the concentration difference between the rich solution and the poor solution (Eq. 14).

$$\text{COP}_C = \frac{\dot{Q}_{\text{EVA}}}{\dot{Q}_{\text{GEN}} + P_{\text{PUMP}}} \quad \text{Eq. 10}$$

$$\text{COP}_H = \frac{\dot{Q}_{\text{ABS}} + \dot{Q}_{\text{CON}}}{\dot{Q}_{\text{GEN}} + P_{\text{PUMP}}} \quad \text{Eq. 11}$$

$$\Delta T_{\text{LIFT;ext}} = \frac{T_{\text{htw;1}} + T_{\text{htw;3}}}{2} - \frac{T_{\text{brn;1}} + T_{\text{brn;2}}}{2} \quad \text{Eq. 12}$$

$$\Delta T_{\text{LIFT;int}} = T_{18} - T_{21} \quad \text{Eq. 13}$$

$$\Delta \xi = \xi_{\text{rso}} - \xi_{\text{pso}} \quad \text{Eq. 14}$$

2.3 Second Law Analysis

According to the first law of thermodynamics all forms of energy are convertible in one another. But different forms of energy have different potential to produce useful work, thus the second law of thermodynamics limits the possible energy conversion. The second law analysis of the absorption heat pump have been performed by determining the exergy losses of each process component acc. to Aphornratana and Eames (1995), the procedure is described hereafter.

A reversible process is one that can be reversed completely and after reversing leaves no change in the system or its surroundings. All real processes are not reversible due to e.g. friction, heat transfer, hysteresis, mixing or diffusion processes.

Heat and work interaction can be defined in terms of exergy, which always decreases during real irreversible processes. Exergy is defined as the maximum possible reversible work that can be obtained from a source with respect to the surrounding environment and is given for work in Eq. 15, for heat in Eq. 16, and for a mass flow in Eq. 17, if changes in the potential and kinetic energy can be neglected.

$$\dot{E}_p = P \quad \text{Eq. 15}$$

$$\dot{E}_Q = \left(1 - \frac{T_{\text{ENV}}}{T}\right) \cdot \dot{Q} \quad \text{Eq. 16}$$

$$\dot{E}_M = \dot{m} \cdot [h - h_{\text{ENV}} - T_{\text{ENV}} \cdot (s - s_{\text{ENV}})] \quad \text{Eq. 17}$$

For each process component the exergy loss can be calculated applying Eq. 18.

$$\Delta \dot{E} = \sum \dot{E}_p + \sum \dot{E}_Q + \sum \dot{E}_M \quad \text{Eq. 18}$$

In the following heat losses to the environment have been neglected and except the solution pump no work occurs, thus only the mass flows are to be considered. As an example the exergy loss of the co-current generator is given by Eq. 19

$$\Delta \dot{E}_{\text{GEN}} = \dot{E}_{\text{src;1}} + \dot{E}_4 - \dot{E}_{\text{src;2}} - \dot{E}_5 \quad \text{Eq. 19}$$

2.4 Assumptions and boundary conditions

In order to solve the system of equations different assumptions and the definition of boundary conditions for the external circuits are necessary. Table 2 summarizes the used assumptions.

Table 2: Assumptions and boundary conditions used for the process calculation

State point	Value	Unit	Description
t_{ENV}	17	°C	Temperature of the environment
m_{brn}	0,35	kg/s	Cold water mass flow rate
m_{htw}	0,5	kg/s	Cooling water mass flow rate
m_{src}	0,187	kg/s	Heating water mass flow rate
ΔT_1	5	K	Temperature difference between t_1 and $t_{htw;2}$
ΔT_{18}	4	K	Temperature difference between t_{18} and $t_{htw;3}$
ΔT_{21}	5	K	Temperature difference between $t_{brn;2}$ and t_{21}
ΔT_{GEN}	14	K	Logarithmic mean temperature difference in the generator
ΔT_{22-21}	3	K	Temperature difference between t_{22} and t_{21}
ΔT_{17-18}	10	K	Temperature difference between t_{17} and t_{18}
$q_1, q_6, q_{15}, q_{16}, q_{18}$	0	-	Vapor fraction at state points 1, 6, 15, 16 and 18
q_{12}, q_{13}, q_{17}	1	-	Vapor fraction at state points 12, 13 and 17
ε_{SHX}	0,88	-	Heat exchanger efficiency of solution heat exchanger
ε_{PRC}	0,5	-	Heat exchanger efficiency of refrigerant pre-cooler
η_{PUMP}	0,5	-	Efficiency of solution pump

3 RESULTS AND DISCUSSION

The process simulation has been performed for the co-current flow and the counter-current flow generator at different temperature levels of the heat sources and the heat sink.

Figure 3 shows one result for the coefficient of performance for cooling depending on the generator outlet temperature of the poor solution (t_6) for four different load cases with evaporation / condensation temperatures of 10/30, 0/30, 10/50 and -10/50 °C and varying $\Delta\xi$ starting at 0.05 up to 0.15.

The temperature of the poor solution at the generator outlet increases from about 70°C to about 170°C and the COP_c decreases from about 0.75 to about 0.4 for the 4 load cases with increasing $\Delta\xi$. The deviation of the results between the co-current and counter-current generator increases with increasing generator temperature whereas the difference is small for a small $\Delta\xi$ and becomes bigger with increasing $\Delta\xi$. The results for the co-current generator show a maximum of the COP_c at $\Delta\xi$ approx. 0.1 for all load cases while for the counter-current generator the COP_c has a maximum at $\Delta\xi > 0.15$.

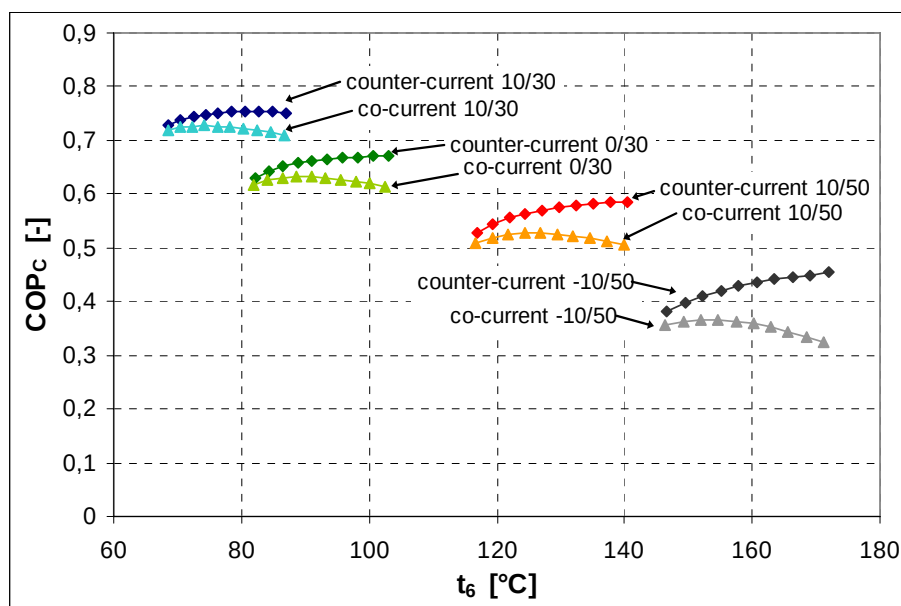


Figure 3: Coefficient of performance for cooling depending on the generator outlet temperature of the poor solution of co-current and counter current flow generator design for four different load cases with evaporation / condensation temperature of 10/30, 0/30, 10/50 and -10/50 °C and varying $\Delta\xi = 0,05$ to $0,15$.

In order to gain a more detailed insight to the origin of the process losses the exergy loss of each process component has been calculated for both generator designs as discussed in chapter 2.

Figure 4 shows the simulation result for an evaporation temperature of -10°C and varying condensation temperature (20 up to 50°C) for co-current generator design (left) and counter-current generator design (right). The exergy losses in the Figure 4 have been normalized in order to show which component contribute most to the total exergy loss. Depending on the condensation temperature the exergy losses of the main components dominate at low temperature lifts. At high temperature lifts the losses in the internal components and in the main components are approx. of the same size. If one compares the normalized exergy losses of the two designs to each other it appears that the losses of the dephlegmator and rectification column are higher for the co-current design, for the counter-current design the exergy loss of the solution heat exchanger is higher.

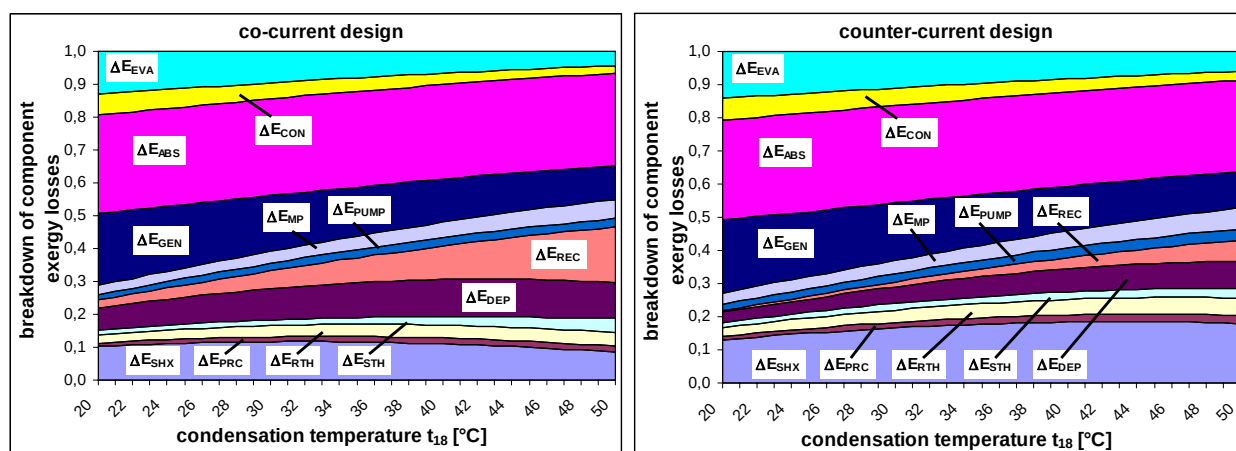


Figure 4: Normalized exergy losses for evaporation temperature of -10°C and varying condensation temperature (20 and 50°C) for a co-current (left) and a counter-current generator design (right); $\Delta\xi = 0.1$

Figure 5 to Figure 8 show the exergy losses depending on the $\Delta\xi$ for the load case 10/30 and -10/50, whereas the exergy losses of the main components are shown on the left hand side and the exergy losses of the remaining components are shown on the right hand side.

Figure 5 shows the results for the co-current and Figure 6 for the counter-current generator design for the load case 10/30. At this load case the losses of the main components are more than three times higher than the exergy losses due to other components and especially the generator and absorber losses are dominant. The generator loss decreases slightly with increasing $\Delta\xi$ due to the fact, that with increasing $\Delta\xi$ the solution mass flow rate decreases and the outlet temperature of the poor solution becomes higher, thus the temperature difference between poor solution outlet and heating water inlet decreases. The absorber loss increase significant because of the increased $\Delta\xi$ also the inlet temperature of the poor solution in the absorber increases which causes a higher temperature difference for the heat transfer in the absorber.

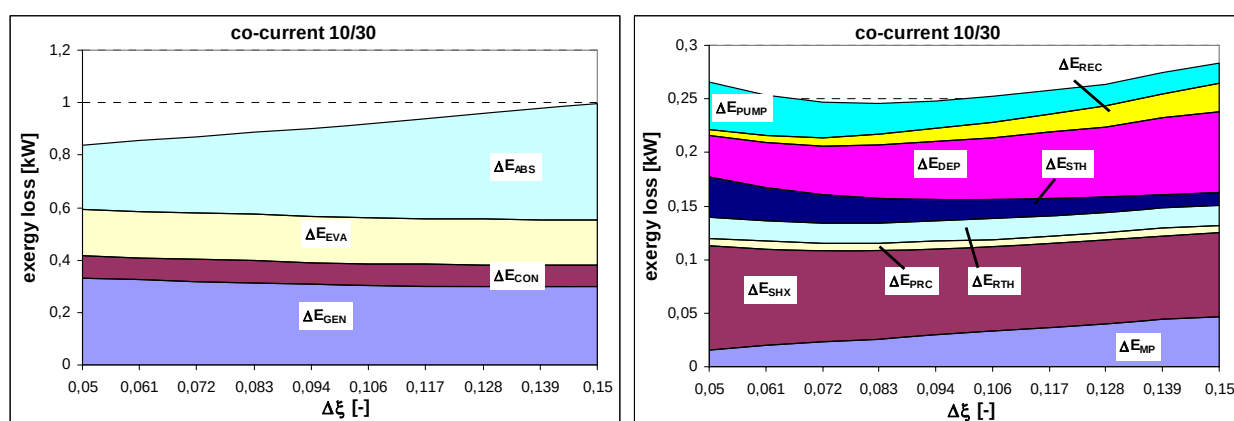


Figure 5: Co-current generator design - exergy losses caused by the main components (left) and by the internal components (right) for the load case 10/30 depending on $\Delta\xi$

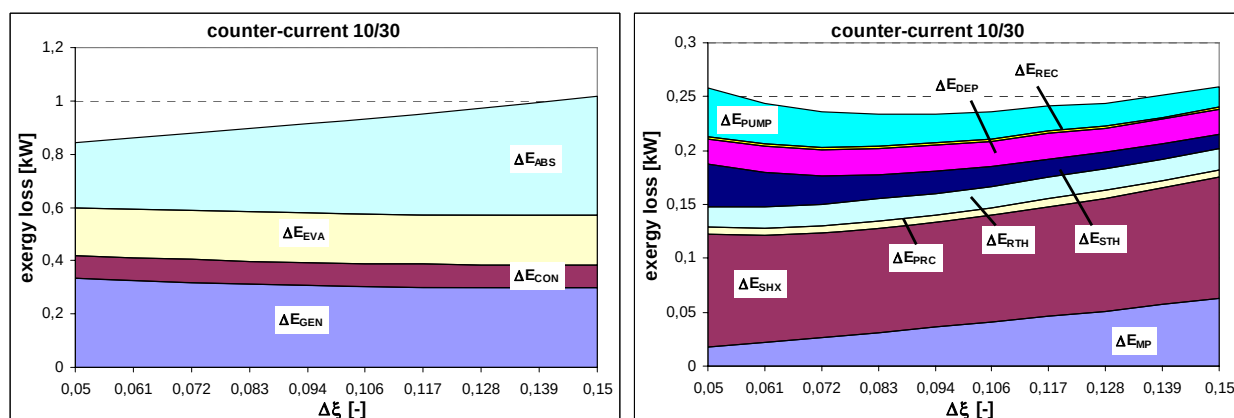


Figure 6: Counter-current generator design - exergy losses caused by the main components (left) and by internal components (right) for the load case 10/30 depending on $\Delta\xi$

Acc. to the right diagram of Figure 5 (internal losses of the co-current design), the losses of the solution heat exchanger, mixing point upstream the absorber and dephlegmator have the largest share on the internal exergy loss. The results for the counter-current design (Figure 6) are similar, but the losses of the dephlegmator and the rectification column are lower compared to the co-current generator design.

Figure 7 and Figure 8 show the result for the load case -10/50 for both co-current and counter-current generator design. At this load case a high internal temperature lift occurs, thus the losses due to the internal irreversibilities are significantly higher than for the load

case 10/30. The main part of the exergy losses of the main components occurs in the absorber which increases with increasing $\Delta\xi$.

The process with the co-current generator design seems to have a strong interaction between the exergy losses of the solution heat exchanger, dephlegmator and the rectification column. With increasing $\Delta\xi$ the losses of the rectification column increases dramatically, while the losses of the solution heat exchanger and the dephlegmator decreases.

The solution heat exchanger losses decrease with increasing $\Delta\xi$ because, the solution mass flow rate decreases and the dephlegmator capacity increases, this leads to a higher pre-heating of the rich solution before it enters the solution heat exchanger. The dephlegmator loss decreases because the temperature difference in the dephlegmator becomes smaller with increasing $\Delta\xi$ which is caused by the used assumptions of the simulation model i.e. the assumption of constant temperature difference between t_{17} and t_{18} .

The exergy analysis of the different load cases and generator designs helps to determine the magnitude of the irreversibilities which occur in the different components, but it does not indicate the interactions of the different components and how the irreversibility of the whole system changes if the irreversibility of one component varies. In order to investigate these relations it is intended to perform a structural analysis (Boer, 2005) of the absorption heat pump designs in the near future.

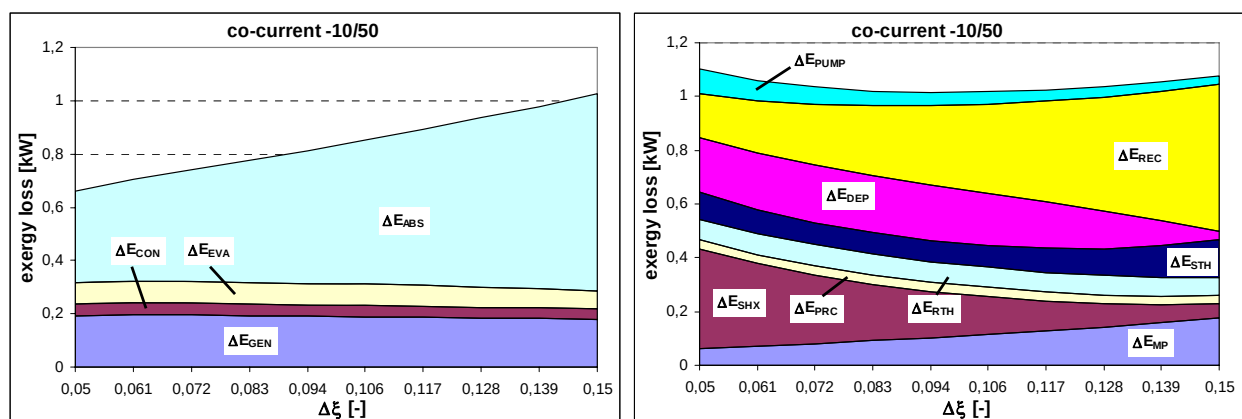


Figure 7: Co-current generator design - exergy losses caused by the main components (left) and by the internal components (right) for the load case -10/50 depending on $\Delta\xi$

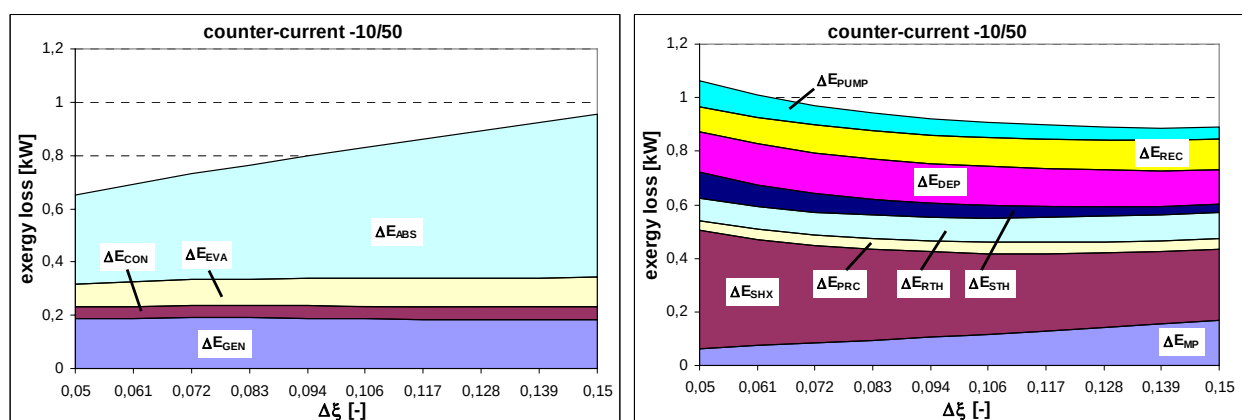


Figure 8: Counter-current generator design - exergy losses caused by the main components (left) and by the internal components (right) for the load case -10/50 depending on $\Delta\xi$

4 CONCLUSIONS

The thermodynamic analysis of two $\text{NH}_3/\text{H}_2\text{O}$ absorption heat pump processes with co-current and counter-current generator design have been presented using both, first and second law analysis.

The results show that the component exergy losses depend on the operating condition especially on the temperature level of the heat sources and the heat sink. At low temperature lifts (about 20 K) the exergy losses in the main components (absorber, generator, condenser and evaporator) are more than three times higher than the exergy losses in the other (internal) components. Thus it can be concluded, that for a small temperature lift the improvement of the main components - i.e. by optimization of the heat transfer - seems to be promising for the optimization of the investigated process.

At high temperature lifts (about 60 K) the losses in the internal components and in the main components are approx. of the same size and the absorber, solution heat exchanger, dephlegmator and the rectification column are dominant. Thus these components shall be considered for process optimization. The losses of the co-current generator are significantly higher than those of the counter-current generator design.

The exergy analysis of the different load cases and generator designs helps to determine the magnitude of the irreversibilities which occur in the different components, but it does not indicate the interactions of the different components and how the irreversibility of the whole system changes if the irreversibility of one component varies. In order to investigate these relations it is intended to perform a structural analysis of the absorption heat pump designs in the near future.

NOMENCLATURE

ENV	environment	brn	Cold water (brine)
ext	external	htw	Cooling water (water)
in	inlet	pso	Poor solution
int	internal	ref	Refrigerant
out	outlet	rso	Rich solution
		src	Heating water (thermal oil)
c_p	Specific heat capacity [kJ/(kg K)]	ABS	Absorber
$\dot{E}$	Exergy [kW]	CON	Condenser
ε	Efficiency [-]	DEP	Dephlegmator
h	Specific enthalpy [kJ/kg]	EVA	Evaporator
$\dot{m}$	Mass flow rate [kg/s]	GEN	Generator
p	Pressure [kPa]	LS	Liquid separator downstream GEN
P	Mechanical power	LSP	Liquid separator downstream DEP
$\dot{Q}$	Thermal capacity [kW]	MP	Mixing point upstream ABS
q	Vapour fraction[-]	PUMP	Solution pump
s	Specific entropy [kJ/(kg K)]	PRC	Condensate precoolers
t	Temperature [°C]	RAC	Refrigerant accumulator
T	Temperature [K]	REC	Rectification column
ξ	Ammonia fraction [-]	RTH	Refrigerant throttle
		SAC	Solution accumulator
		SHX	Solution heat exchanger
		STH	Solution throttle

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