A 10 kW INDIRECTLY FIRED ABSORPTION HEAT PUMP: CONCEPTS FOR A REVERSIBLE OPERATION

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Abstract: In the last decade several small and medium sized solar or waste heat driven chillers have been developed and brought to market. Nevertheless, in Central Europe where many of these chillers are installed, the required cooling period of buildings is rather short. By using them as a heat pump during winter time their operating period can be extended in order to shorten the payback period and increase the cost effectiveness, and at the same time, the benefit to the environment is increased.

From a thermodynamical point of view it is possible to run a chiller also as a heat pump, but in practice there are restrictions in application due to the dependency of the driving temperature and the temperatures of heat source and heat sink. Using the example of a 10 kW H_2O/LiBr absorption chiller, constraints of and demands on different possible peripheral systems (heat sources and heat sinks) for the reversible operation have been investigated. In the paper we present combinations which are favorable, and others which should be avoided from a primary energy point of view.

Key Words: absorption heat pump/chiller, reversible operation, heat sources/sinks

1 INTRODUCTION

The energy consumption for air-conditioning equipment is rising strongly worldwide. For this reason, in the last decade the development of thermally driven absorption or adsorption heat pumps and chillers has regained importance. Particularly, small and medium sized solar or waste heat driven chillers have been developed. In Germany alone, there were seven sorption chillers between 10 and 50 kW brought into market. But in most cases such systems are not yet competitive to compression chillers, especially in moderate climates where cooling seasons are rather short. Their market share can only be increased by improvement of the economic efficiency. This can be achieved by an extension of the operating period. In moderate climates sorption systems should not only be used for space cooling but primarily for space heating.

Up to now, there are two major approaches in the field of sorption systems: on the one hand solar or waste heat driven sorption chillers developed for space cooling (e.g. Association for sorption cooling e.V. (ASC 2011)) and on the other hand gas driven sorption heat pumps developed for space heating (e.g. Initiative Gaswärmepumpe (IGWP 2011)). From a thermodynamical point of view every chiller can also be applied as a heat pump; only the use of heat source and heat sink circuits is reversed, as for reversible compression heat pumps. But in practice there are restrictions in the application due to the dependency of the driving temperature and the temperatures of heat source and heat sink.

This paper explicates the interdependencies of the three temperature levels (driving heat, heat sink and heat source) and their impact on the operation in heating or cooling mode using the example of a small scale $H_2O/LiBr$ absorption heat pump. Two exemplary concepts for the use of a reversible absorption heat pump are presented.

2 BASIC SPECIFICATION OF THE HEAT PUMP

The absorption system presented here has been developed as small scale chiller with focus on compactness and low driving temperatures for solar cooling (Schweigler et al. 2001). Figure 1 shows the prototype and summarizes the technical data for nominal load.

Cooling mode

Working pair	$H_2O/LiBr$
EER	0.76
Cooling capacity	10 kW
chilled water in/out	18/**15**°C
flow rate	2.9 m³/h
Driving heat flow	13.2 kW
hot water in/out	**75**/65°C
flow rate	1.2 m³/h
Reject heat flow	23.2 kW
cooling water in/out	**27**/35°C
flow rate	2.6 m³/h

Figure 1: Prototype of a 10 kW absorption chiller and technical data

Even without changing the temperatures, the chiller can also be used as a heat pump. As can be seen from the technical data, under nominal conditions the heat pump already supplies water of 35°C for the use in panel heating systems such as floor heating. In this case (heat source temperature of 18°C, e.g. sewage water) a coefficient of performance (COP=useful heat/driving heat) of 1.76 would be achieved. Of course, heating supply temperature can be increased and heat source temperature level can be decreased adapting e.g. the driving temperature. To cope with a lower heat source temperature of 10°C (e.g. ground water) the driving temperature has to be increased to approximately 88°C.

3 INFLUENCE OF TEMPERATURES

3.1 The characteristic curve

Using the method of the characteristic equation (Ziegler 1997) a large range of potential operating conditions can be evaluated. The characteristic equation for the presented chiller has been determined by fitting measurement data (Kühn et al. 2005) to

$$\dot{Q}_E = 0.42 \cdot \Delta\Delta t' + 0.9 \,. \qquad (1)$$

The corresponding characteristic temperature function is

$$\Delta\Delta t' = t_D - 2.5\, t_{AC} + 1.8\, t_E \qquad (2)$$

where t_D, t_A, t_C, t_E are the external arithmetic mean temperatures at desorber, absorber, condenser and evaporator, and t_{AC} is the arithmetic mean value of t_A and t_C. To calculate also the heat output $\dot{Q}_{AC}$ and the COP/EER, characteristic equations for $\dot{Q}_{AC}$ and $\dot{Q}_D$ have also been fitted to

$$\dot{Q}_D = 0.53 \cdot \Delta\Delta t'' - 2.5 \tag{3}$$

$$\dot{Q}_{AC} = 0.94 \cdot \Delta\Delta t''' - 0.4 \,. \tag{4}$$

The corresponding characteristic temperature functions are

$$\Delta\Delta t'' = t_D - 2.1\, t_{AC} + 1.5\, t_E \tag{5}$$

$$\Delta\Delta t''' = t_D - 2.3\, t_{AC} + 1.6\, t_E. \tag{6}$$

The sum of the three heat flows does not necessarily result to zero, because - depending on load and ambient conditions - heat losses to the environment, heat input from the environment, and the internal pumps (solution and refrigerant pump) have to be accounted for when using the fits of all three characteristic equations.

Figure 2 presents the characteristic curves of cooling capacity (chiller mode) and heat output (heat pump mode).

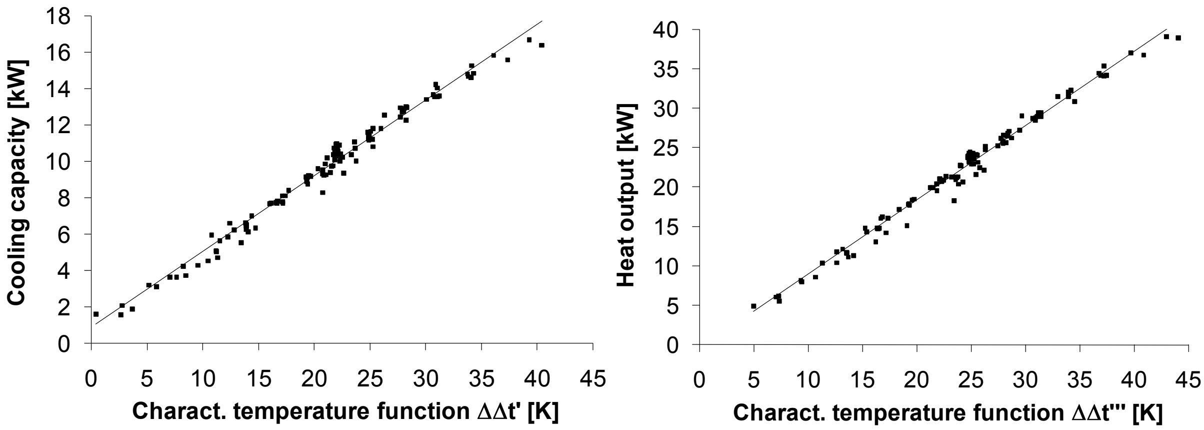

Figure 2: Characteristic curves of cooling capacity and heat output

Evidently, the characteristic equation method provides a simple linear correlation to describe the behavior of the heat pump at design flow rates and at a large range of possible external temperatures. Deviation of 96% of the measurement points from this equation is below 1 kW (in cooling mode) or below 2 kW (in heating mode) which is in each case around 10% of the nominal capacity. The characteristic equations (Eq. 1, 3 and 4) can be used to determine the load behavior and efficiency when changing the characteristic temperature function. It is even exact enough to serve as a control algorithm. As the heat output or cooling capacity does not change if the characteristic temperature function remains constant, it follows immediately how changes of an external temperature can be compensated by control of another external temperature. That means for example, a drop of the heat source temperature by 5 K can be compensated by an increase of the driving temperature by 8 K to keep the heat output constant (Eq. 6).

3.2 Improvement of the method

Using mean temperatures in the characteristic temperature functions (Eq. 2, 5 and 6) has the disadvantage that they usually are not known. They change with the heat flows. Usually only inlet or outlet temperatures are known by the user. This is different in heating and cooling mode. In the case of heating mode, the given values in equations (7) to (9) normally are the driving inlet temperature t_{Din}, the heat source inlet temperature t_{Ein}, and the heating supply temperature t_{Cout}. In the case of cooling mode, the driving inlet temperature t_{Din}, the chilled water outlet temperature t_{Eout}, and the cooling water inlet temperature t_{Ain} are given.

$$\Delta\Delta t' = \frac{t_{Din} - t_{Dout}}{2} - 2.5\frac{t_{Cout} - t_{Ain}}{2} + 1.8\frac{t_{Ein} - t_{Eout}}{2} \tag{7}$$

$$\Delta\Delta t'' = \frac{t_{Din} - t_{Dout}}{2} - 2.1\frac{t_{Cout} - t_{Ain}}{2} + 1.5\frac{t_{Ein} - t_{Eout}}{2} \tag{8}$$

$$\Delta\Delta t''' = \frac{t_{Din} - t_{Dout}}{2} - 2.3\frac{t_{Cout} - t_{Ain}}{2} + 1.6\frac{t_{Ein} - t_{Eout}}{2} \tag{9}$$

In both cases, we still have three unknown temperatures. They can be derived from the well-known equations (10) to (12) which determine the heat flows:

$$t_{Ein} - t_{Eout} = \dot{m} \cdot c_p \cdot \dot{Q}_E = 0.30\,\mathrm{K/kW} \cdot \dot{Q}_E \tag{10}$$

$$t_{Din} - t_{Dout} = \dot{m} \cdot c_p \cdot \dot{Q}_D = 0.74\,\mathrm{K/kW} \cdot \dot{Q}_D \tag{11}$$

$$t_{Cout} - t_{Ain} = \dot{m} \cdot c_p \cdot \dot{Q}_{AC} = 0.33\,\mathrm{K/kW} \cdot \dot{Q}_{AC} \tag{12}$$

By numerically solving the system of equations we can determine either the cooling capacity or the heat output and the EER/COP from any set of known external inlet or outlet temperatures.

The method of the characteristic equation in principle allows an extrapolation. However, a good accuracy in the order of less than 10% of nominal capacity can only been guaranteed if the temperature values are not too far off the measurement range the equations have been fitted for. Else, further refinement of the method is required (Albers et al. 2011). Measurements have been carried out in the following ranges: t_{Ein}=8...20°C, t_{Eout}=5...18°C, t_{Din}=50...105°C, t_{Ain}=23...40°C and t_{Cout}=27...46°C.

In order to ensure save and stable operation, freezing of the refrigerant has to be avoided. To this end, the evaporation temperature T_E must be determined as well. It can be derived from the heat transfer equation

$$\dot{Q}_E = U_E \cdot A_E \cdot \Delta\vartheta_{log,E} = U_E \cdot A_E \cdot \frac{t_{Ein} - t_{Eout}}{\ln\frac{t_{Ein} - T_E}{t_{Eout} - T_E}}\,. \tag{13}$$

Since the temperature difference in the chilled water circuit in most cases is small, the arithmetic temperature difference can be used as an approximation of the logarithmic one which simplifies the equation to

$$\dot{Q}_E = U_E \cdot A_E \cdot ((t_{Ein} + t_{Eout})/2 - T_E)\,. \tag{14}$$

In Figure 3 the temperature difference $(t_{Ein}+t_{Eout})/2-T_E$ is plotted against the cooling capacity. From all measurement data we get the linear equation

$$(t_{Ein} + t_{Eout})/2 - T_E = 0.35\,\mathrm{K/kW} \cdot \dot{Q}_E \tag{15}$$

where 0.35 K/kW is the nearly constant overall heat transfer resistance ($1/U_EA_E$). It should be kept in mind that the approximation is not valid for very low cooling capacity.

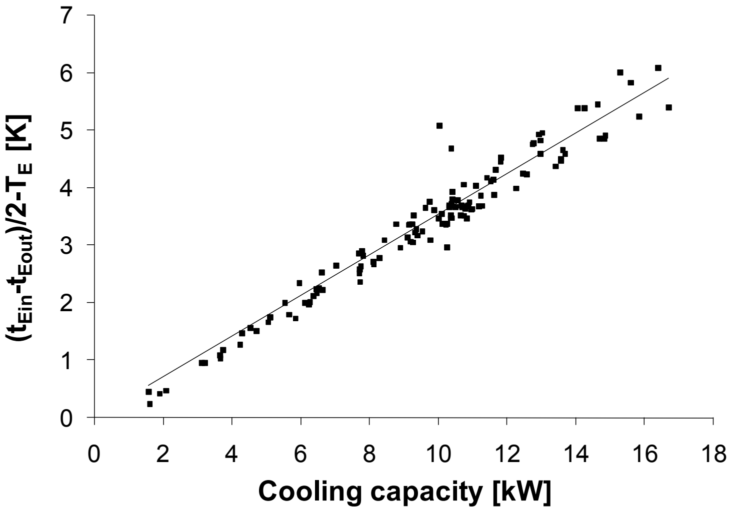

Figure 3: Arithmetic temperature difference $(t_{Ein}+t_{Eout})/2-T_E$ as function of the cooling capacity

4 REVERSIBLE OPERATION

As explained in chapters 1 and 2, the absorption heat pump is able to supply cold in summer time and heat in winter time. In the case of cooling mode, the heat source is the system which serves to distribute the cold in the building. It is connected to the evaporator of the heat pump. In terms of economic feasibility, the same system should be used to distribute the heat in the building in winter time. It is the heat sink, then. Thus, in heating mode this water circuit has to be connected to absorber and condenser. Equally, the heat sink of the cooling mode (water circuit between absorber/condenser and environment) should be the same as the heat source of the heating mode (water circuit between evaporator and environment). The switching of these two water circuits (heat source and heat sink) can easily be done by a valve, in the same way as in reversible compression heat pumps.

4.1 Operating conditions

Before evaluating the heat pump operation with different heat sources and heat sinks for cooling and/or heating mode operating conditions have to be specified. Two mandatory requirements have to be kept in mind in order to ensure a proper operation of the heat pump:

- The LiBr solution must not crystallize. Therefore, a minimum distance from the crystallization line of 3 percentage points has to be maintained.
- The refrigerant (water) must not freeze. Therefore, a minimum evaporation temperature of 3°C has been defined.

For reasons of energy efficiency we specified the following additional boundary conditions:

- In cooling mode, a minimum cooling capacity of 5 kW (50% of the nominal load) has been defined. It is also possible to achieve 25% part load or even less (see Figure 2, left), but even though cooling capacity is low, electric power consumption e.g. for the external pumps does not drop proportionally. This boundary condition is not mandatory for the operation but may be specified, depending on the system in order to achieve reasonable electrical EERs.
- In heating mode, a minimum thermal COP of 1.3 has been defined. Compared to a condensing boiler an additional energy input for external pumps like the heat source pump is required. To ensure a reasonable operation in terms of primary

energy use a high COP value is essential. Again, this condition is to be specified depending on the system.

Finally, we defined the following maximum differences between supply and return temperature of the air-conditioning equipment:

- 5 K for panel heating or cooling equipment like floor heating or chilled ceilings,
- 10 K for heating systems with supply temperatures above 40°C.

The last four conditions are not specific for the heat pump itself but they depend on the system.

4.2 Operational alternatives

Possible types of heat sources for the cooling mode have been classified into three groups based on the chiller outlet temperature:

- 7°C: central ventilation and air conditioning (VAC) systems (without downstream fan-coils), fan-coils (without upstream VAC system),
- 16°C: chilled ceilings or walls, passive cooling convectors, induction units and fan-coils with upstream VAC system, façade ventilation units,
- 20°C: concrete core cooling, floor cooling, radiators.

Possible heat sinks for the heating mode have been classified into five groups based on the heating supply temperature:

- 30°C: ceiling heating, concrete core heating,
- 35°C: floor heating,
- 40°C: wall heating, passive convection heaters,
- 45°C: central heating, ventilation and air conditioning (HVAC) systems, induction units and fan-coils with upstream HVAC system, fan-coils without upstream HVAC system, façade ventilation units, radiators,
- ≥ 50°C: central heating, ventilation and air conditioning (HVAC) systems, induction units and fan-coils with upstream HVAC system, fan-coils without upstream HVAC system, façade ventilation units, radiators.

As heat sinks for the cooling mode or heat sources for the heating mode respectively, air, water, and ground have been considered. While ground, surface or sewage water, bore holes, ground collectors, or dry or hybrid cooling towers can be used both for the cooling as well as for the heating mode, an open wet cooling tower is only feasible as a heat sink for the cooling mode. To increase the temperature of the heat source air, in the heating period a solar or air collector can be used.

The different possibilities have been combined in a matrix. For all combinations the heating and cooling power has been calculated for the presented heat pump using the model of chapter 3. Further assumptions have been made as follows:

- A constant driving temperature of 95°C has been assumed for cooling and heating mode. Doing so, of course, the power output of heat pump and chiller are restricted. The full potential of the chiller is not exploited as higher driving temperatures up to 120°C can be applied. However, in heating mode the driving temperature in some cases has been decreased below 95°C (e.g. by recirculation of the hot water return flow) if there was the risk of the evaporation temperature dropping below 3°C or the temperature difference in the heating system exceeding the maximum value of 5 or 10 K. In the case of cooling mode additionally an increase of the cooling water inlet temperature has been investigated in order to save the refrigerant from freezing. In this case the cooling tower fan speed is reduced. Considerable power savings can be achieved applying this alternative control strategy (Kühn et al. 2008a).
- The closest approach temperature difference of all heat exchangers in the peripheral systems, e.g. the dry cooler or bore hole heat exchanger, has been assumed to be 2 K.

- A heating limit temperature of 15°C has been assumed.
- In a first attempt, no heating characteristic has been regarded for the consumer. The heating supply temperatures have been assumed to stay constant with varying ambient temperature. Thus, the results turn out to be conservative.

As not all combinations can be presented in detail here we decided to demonstrate the most representative examples.

4.2.1 Bore holes/ground water as heat source/heat sink

A constant soil respectively ground water temperature of 10°C was assumed all year round. With this assumption water instead of brine can be used, also in case of bore holes. This assumption is reliable as the ground is cooled down (regenerated) during winter time. However, a detailed study on the ground temperature has not been performed yet.

Accordingly, in cooling mode we get a constant absorber inlet temperature of t_{Ain}=12°C. For cooling systems working with chilled water supply temperatures of 16°C or higher the ground water can be used directly; an operation of the heat pump in chiller mode is not required.

In case of 7°C chilled water supply temperature (i.e. for air dehumidification) the chiller is able to provide a cooling capacity of nearly 23 kW with an EER of 0.82. As the evaporation temperature drops below 3°C to 2.4°C, we have to limit the cooling capacity. This can be achieved either by decreasing the driving temperature to 84.6°C (e.g. by recirculation of the desorber outlet, see Figure 4, left side) or by increasing the cooling water temperature to 15.9°C (e.g. by recirculation of the condenser outlet, see Figure 4, centre). Cooling capacity in both cases is 20 kW. As the thermal EER has its maximum at medium driving temperatures, the EER in the first case is 0.83 and drops at rising cooling water temperatures to 0.80. But as already mentioned, the second control strategy can offer considerable advantages in terms of the electrical EER. Another possibility which was not investigated in this work is to reduce flow rates.

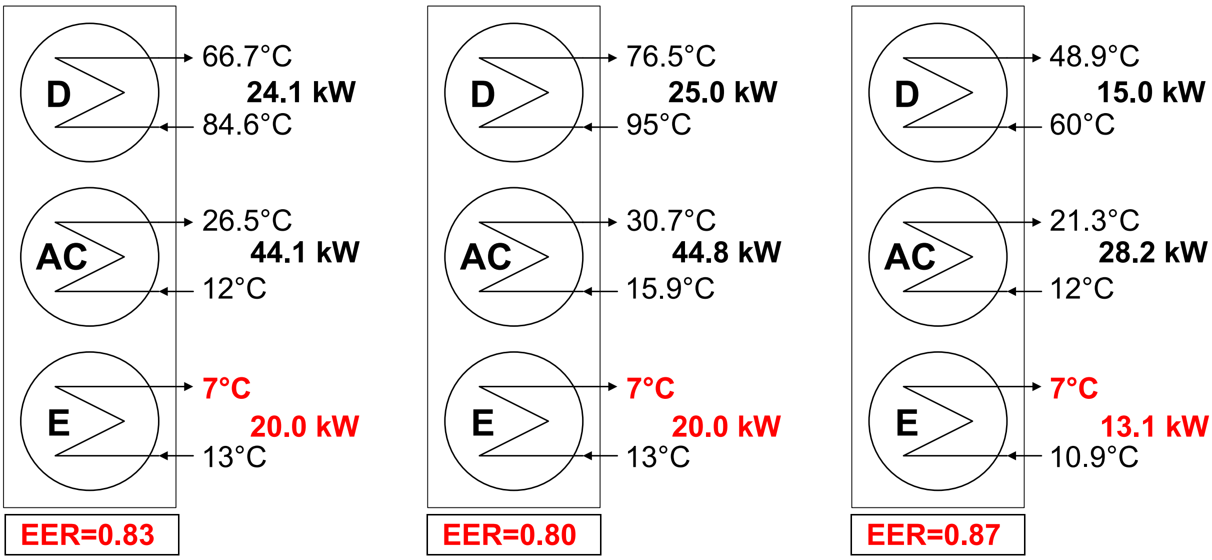

Figure 4: Chiller operation (t_{Eout}=7°C) with bore holes or ground water as heat sink (left: driving temperature decreased, centre: cooling water temperature increased, right: solar cooling)

If a solar collector is used as driving source instead of waste heat from e.g. CHP units or district heat, driving temperature in moderate climate like in Central Europe will not be that high (especially when using economic flat plate types). A chiller with only 60°C driving temperature, 12°C absorber inlet and 7°C evaporator outlet temperature delivers a cooling capacity of 13 kW with an EER of 0.87 (see Figure 4, right side). It has to be admitted that these calculations have not been validated experimentally yet. The minimum absorber inlet temperature applied so far was 23°C. In this example, we assume an absorber inlet temperature of only 12°C. So the quoted EER might be too high. Nevertheless, the temperature lift in such point of operation is very small so a high EER is reasonable.

In heating mode (Figure 5), a minimum evaporator inlet temperature of t_{Ein}=5°C has been assumed (in case of bore holes). With this boundary condition a maximum heat supply temperature of 40°C can be achieved. Heat output in this case is around 10 kW; the COP is still acceptable with a value of 1.4. It increases with decreasing heating supply temperature to 1.65 at t_{Cout}=30°C. In the cases of 30°C and 35°C heating supply temperature the heat output had to be limited (by decreasing the driving temperature) as the evaporation temperature fell below the lower limit of 3°C. Measures to depress the freezing point of the refrigerant would help to use the full potential of the heat pump but have not been implemented here.

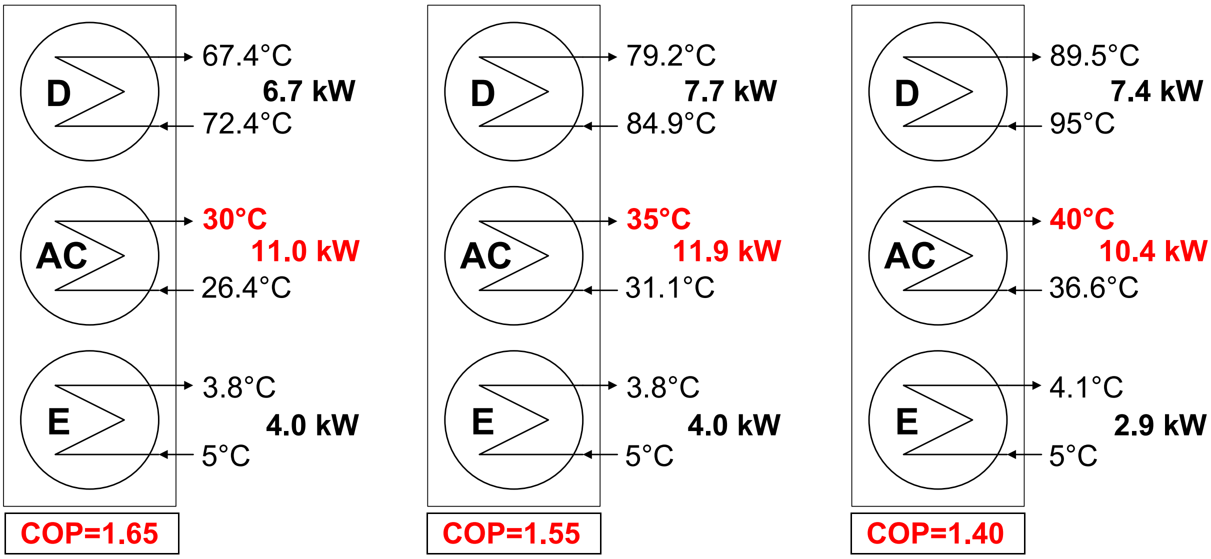

Figure 5: Heat pump operation with bore holes as heat source (t_{Ein}=5°C, heating supply temperature left: 30°C, centre 35°C, right: 40°C)

An increase of the heat source temperature has a significant influence (compare Figures 5 and 6). With an increase of only 3 K in the evaporator inlet temperature from 5°C to 8°C (e.g. ground water) the heat output is roughly between 30% and 45% higher. The COP is also increasing considerably.

The all year round use of bore holes is favourable as the ground which cools down during heat pump operation is regenerated in summer by the reject heat of the chiller. Nevertheless, heat transferred to the ground and heat removed from it should be balanced. Cooling capacity and heat output first have to be adapted to the building's requirements. There are buildings with very good insulation standard. There, cooling load might be higher than heating load. In this case the reject heat flow transferred to the ground in cooling mode is much higher than the heat source flow removed from it in heating mode. Consequently, the ground temperature will increase which is favourable for the heating mode; the system will adapt. However, there are also cases where heating load will be considerably higher. The number of bore holes required fits better for the reversible operation in the latter case. This applies in particular given that in moderate climate the heating period is longer than the cooling period. It further has to be considered that in cases of 16°C or 20°C chilled water supply temperature where the heat source is directly used to cool the building only approximately half of the heat is transferred to the ground.

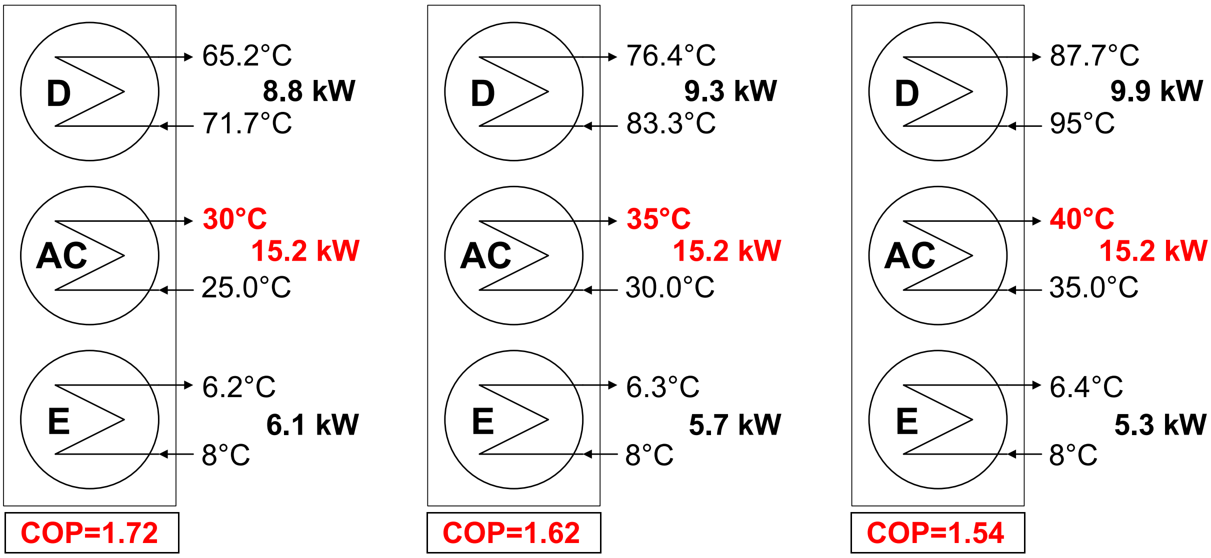

Figure 6: Heat pump operation with bore holes or ground water as heat source (t_{Ein}=8°C, heating supply temperature left: 30°C, centre 35°C, right: 40°C)

4.2.2 Air as heat source/heat sink

Ambient air can be used as a heat source or heat sink by means of a cooling tower. A wet cooling tower presents a very effective possibility to reject heat in cooling mode as cooling water temperatures below ambient temperature can be provided. Nevertheless, the design of an open wet cooling tower does not allow the use of ambient air as a heat source in winter time. Therefore, only dry or hybrid coolers (which also offer the advantages of a wet cooling tower at hot summer days) or closed cycle wet cooling towers can be applied for a reversible operation of the heat pump.

Cooling operation has been defined to start at an ambient temperature of 24°C (corresponding to an absorber inlet temperature of 26°C). Up to this ambient temperature window ventilation has been assumed to be sufficient. The cooling capacity delivered by the chiller and the EER depending on ambient temperature are presented in Figure 7. The parameter t_{Eout} is the chilled water temperature delivered to the building's cooling system.

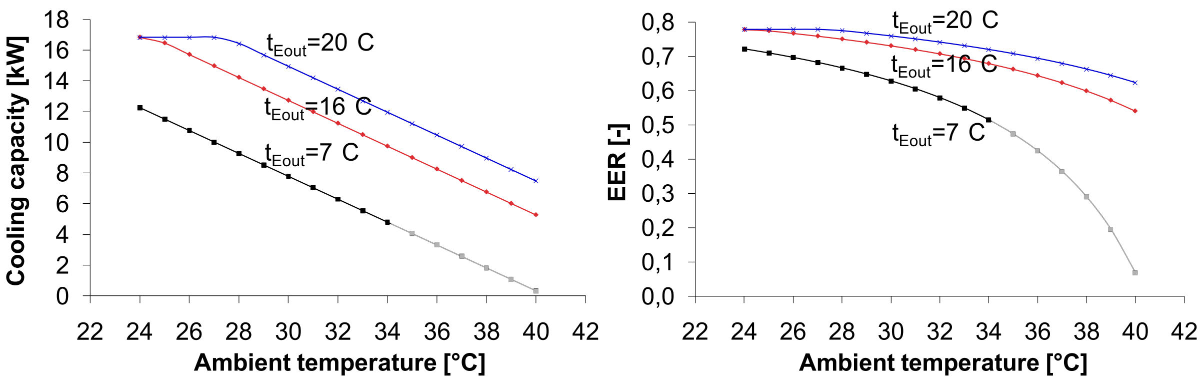

Figure 7: Chiller operation with air as heat sink (dry or hybrid cooler)

In the cases of 16°C and 20°C chilled water temperature a maximum temperature difference in the chilled water circuit of 5 K has to be maintained. Therefore, the maximum cooling capacity is limited to approximately 17 kW at 24°C ambient temperature (EER=0.8). Minimum cooling capacity is 5.3 kW at 16°C chilled water und 40°C ambient temperature (EER=0.5), and 7.5 kW at 20°C chilled water temperature (EER=0.6), respectively. Maximum cooling capacity at 7°C chilled water temperature is 12.3 kW at 24°C ambient temperature (EER=0.7). In this case, the cooling capacity falls below the defined minimum value of 5 kW at ambient temperatures above 34°C. That means that in hot climate HVAC-systems or fan-coils used for air humidification cannot be combined with a dry cooler but a hybrid cooler has to be used. The EER of the chiller at 7°C chilled water temperature and 34°C ambient temperature is still above 0.5.

In heating mode, air can only be used as a heat source at ambient temperatures from 6°C and above which limits application considerably. This is due to the impending freezing of the refrigerant water. In (Kojima et al. 2003, Richter et al. 2007, and Kühn et al. 2008b) a possibility to enlarge the operation limits to heat source temperatures even below 0°C has been presented. This method is not considered in this paper, as stated before.

In case of low temperature heating systems (30°C and 35°C heating supply temperature) the temperature difference in the heating circuit is limited to 5 K. This is achieved by a decrease of the driving temperature. Therefore, the heat output is limited to maximum 15 kW (COP=1.8 at t_{Cout}=30°C and 1.7 at t_{Cout}=35°C, see Figure 8). In case of heating systems with supply temperatures of 40°C (or higher) a temperature difference of 10 K is allowed. Therefore, the maximum heat output of such a system at 15°C ambient temperature (corresponding to an evaporator inlet temperature of 13°C) is 23 kW (COP=1.7). The lowest heat output in all three heating systems is between 7 and 9 kW at 6°C ambient temperature (corresponding to an evaporator inlet temperature of 4°C). The COP at this operating point is higher than the requested 1.3 in case of 40°C heating supply temperature and nearly 1.6 in case of 30°C heating supply temperature. A heating supply temperature of 45°C can only be delivered at ambient temperatures near to the heating limit temperature and is therefore not very reasonable.

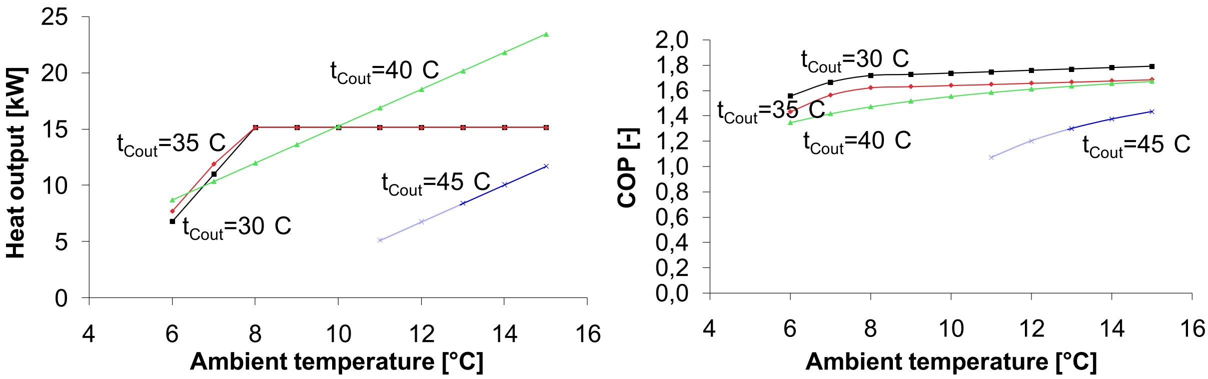

Figure 8: Heat pump operation with air as heat sink (dry or hybrid cooler)

An option in case of solar cooling, i.e. if a solar collector is used to provide the required driving heat in cooling mode, is to use the collector as heat source equipment, also. It is supposed that the operating range with air as a heat source can be extended considerably. In a draft of the German norm VDI 4650-2 a temperature increase of 5.6 K above ambient temperature is reported when using a collector with an aperture area of 8 m² (VDI 2010).

5 SUMMARY

In this paper a reversible H_2O/LiBr absorption heat pump is presented for which all possible combinations of heat source and heat sink have been evaluated using an improved version of the characteristic equation method. The most representative examples have been presented in detail; influencing parameters have been described.

The reversible water-water absorption heat pump can be classified in accordance with the EN 12309-2 (CEN 2000) as W10/W35 with a heat output of 15.2 kW and a COP of 1.6 (heating mode) and as W30/W7 with a cooling capacity of 9.4 kW and an EER of 0.7 (cooling mode). The use of the heat pump is possible up to heating supply temperatures of 40°C (W10/W40) without limitations. The supply of higher heating supply temperatures is not reasonable. Air can also be used as a heat source or heat sink, respectively, but not directly. Heat has to be transferred to/from a water circuit. The use of air as a heat source has been limited to a minimum ambient temperature of 6°C if pure water is used as a refrigerant. Below this temperature the driving heat must directly be used to heat the building. The use of a solar collector as a heat source can extend the operating time of the heat pump considerably. The use of air as a heat sink in summer is limited to an ambient temperature of 34°C only in case of a low chilled water supply temperature of e.g. 7°C. In cases of higher chilled water supply temperatures, like e.g. 16°C, heat of the chiller can efficiently been rejected up to an ambient temperature of 40°C. The prototype can also be used as a brine-water heat pump as explained in (Kühn et al. 2008b) but to a limited extent.

It can be concluded, that the absorption chiller is very flexible and is able to deliver a heat output and a cooling capacity very different from the rated conditions. The integration into a building energy supply system consisting of a heat source and a heat sink imposes some restrictions, especially if a reversible operation is requested. There, deliverable useful heat flows have to be adapted to the requirements of the peripheral circuits, e.g. a maximum temperature difference of the heating or cooling system. The heat pump already allows the use of many combinations of heat sink and heat source, but by using an intelligent system integration there are still more options (e.g. the use of a solar collector as driving source in cooling mode and as heat source in heating mode).

For a final assessment, in a further step seasonal COPs (SCOP) and seasonal EERs (SEER) of the reversible absorption heat pump will be calculated and the different possibilities will be evaluated regarding cost, applicability and comfort for the user of the heating and cooling system.

6 NOMENCLATURE

t, T	temperature [°C]
$\dot{Q}$	heat flow [kW]
$\Delta\Delta t$	characteristic temperature functions [K]
$\dot{m}$	mass flow rate [kg/s]
c_p	specific heat capacity [kJ/kgK]
U	heat transfer coefficient [kW/m²K]
A	heat transfer area [m²]
$\Delta\vartheta_{log}$	logarithmic temperature difference [K]

Indices

D	desorber
A	absorber
C	condenser
AC	absorber/condenser in series
E	evaporator
in	inlet
out	outlet

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