AMMONIA/WATER ABSORPTION HEAT PUMP WITH "THERMALLY DRIVEN" SOLUTION PUMP

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Abstract: Only a few electrical solution pumps are available on the market suitable for small-capacity ammonia/water absorption heat pumps with an evaporator capacity below 20 kW. In order to improve this situation a "thermally driven" solution pump can be considered, which offers several advantages, as e.g. an oil-free, leak-proof and simple design. "Thermally driven" means that the pump is "internally" driven by a thermodynamic power process within the absorption heat pumping cycle instead of electricity. Further, this novel pump ("ThermoPump") can be easily integrated in any plant without special efforts regarding its basic layout. The experimental analysis of the ThermoPump operating in a commercially available absorption heat pump shows that this concept works without any relevant operating-limits, negative influence on the dynamic behaviour or electricity demand. However, the measured COP of the plant is decreased by at least 15% by the ThermoPump compared to the existing electrical pump. Nevertheless, thermal energy (e.g. CO_2-free waste heat) can be used to drive the solution pump instead of electricity, and this can be of interest from an economical and ecological point of view.

Key Words: absorption heat pumps, heat operated, solution pump

1 INTRODUCTION

Thermally driven absorption heat pumping systems (AHP) - for both cooling and heating applications - can contribute to the reduction of anthropogenic greenhouse-gas emissions by the integration of renewable energy, as e.g. solar heat, geothermal heat, ambient heat etc. or waste heat. Therefore, AHPs offer a large ecological potential for the energy supply, as discussed by e.g. (McMullan 2002), (Ziegler 2002) or (Li et al. 2012).

Recently, the combination of co-generation plants with AHPs - so called tri-generation systems, which are combined cooling, heating and power units are of a growing interest allowing a significant contribution to a sustainable energy supply (Denga et al. 2011), (Fumo et al. 2009) and (Ramming 2013). Besides the ecological benefit from an economic point of view, this combination offers the possibility to increase the degree of capacity utilization, because the heat from the prime mover can be used to drive an AHP for cooling purposes at times of no heat demand (Zotter and Rieberer 2014). Several prime movers for the AHP can be considered, as e.g. an internal combustion engine (Ramming 2013) or (in future) a fuel cell (Radermacher et al. 2013). Particular, small-capacity tri-generation systems offer various application possibilities in households, commercial buildings and the industry but small-capacity AHPs with e.g. an evaporator capacity about 20 kW are required. Furthermore, the use of ammonia/water as working fluid of the AHP offers the possibility of cold water temperatures below 0°C, which expands the number of application possibilities. However, the market success of ammonia/water AHPs depends generally on the investment cost, the energy price, the efficiency and the reliability of the plant. Therefore, since recent years the reduction of the investment cost of ammonia/water AHPs is focused.

The solution pump of an AHP is required to overcome the difference between low and high side pressure. Nevertheless, the solution pump, which is most commonly an electrically driven pump, is one of the cost drivers of ammonia/water AHP plants, especially of small-capacity plants accounting 10 to 25% of the system cost. Besides, the several technical requirements a solution pump of small-capacity ammonia/water AHP has to meet, the delivery rate is compared to the necessary pressure lift very low (Safarik 2003), (Zotter et al., 2011). Therefore, only a few electrically driven solution pumps are available on the market, which are suitable for small-capacity ammonia/water AHPs and most of them are relatively complex, expensive and have substantial potential for improvements (De Francisco et al. 2001), (Sakr et al. 1987) and (Safarik 2003).

In order to improve this situation a so called "thermally driven" or "heat operated" pump can be considered (Zotter et al. 2011). Such a pump is "internally driven" by a thermodynamic power process within the AHP cycle instead of electricity. The term "thermally driven" pump is based on the fact, that the required energy is generated by means of the heat supply and heat rejection of the AHP cycle itself (see Kahn 1995). Therefore, low-ex energy, as e.g. waste heat can be used for the operation of the solution pump. Furthermore, a pump concept requires no power transmission from an external energy source, as e.g. by a rotating shaft. I.e., it offers an oil-free, leak-proof and simple design, which can result to lower cost of the pump (Zotter et al., 2011 and Zotter & Rieberer, 2013). A thermally driven solution pump concept is not novel, because already the diffusion-AHP developed by Platen and Munters in 1928 operates autonomously without any further electricity demand for their bubble pump. However, based on a detailed patent and literature review several concepts of thermally driven solution pumps suitable for AHP cycles (working without any inert gas) have been found (Altenkrich 1954), (Dijkstra and Huizinga 1985), (Page 1986), (Vinz 1986 and 1988), (Knoche 1992), (Dawoud et al. 1993), (Karthikeyan 1993), (Kahn 1995), (Dawoud and El-Ghalban 2002), but none of these is available on the market up to now (Zotter et al. 2011). Hence, a new concept of a thermally driven solution pump, the so-called "ThermoPump", which can be integrated in any AHP "independent" of its design and working without any inert gas has been developed, constructed and experimentally investigated.

Besides the working principal of the ThermoPump and an overview of the most important experimental results (compare with Zotter & Rieberer 2013 and Zotter & Rieberer 2014), this paper discusses also further potential for design optimization of the pump prototype according to simulations and the overall electricity consumption of the AHP using the ThermoPump including "parasitic" electricity consumers, as the ventilation system of a dry re-cooler are discussed.

2 THE THERMOPUMP – A THERMALLY DRIVEN SOLUTION PUMP

The ThermoPump is a new kind of thermally driven solution pump suitable for a small-capacity ammonia/water AHP, driven by a portion of vaporised refrigerant from the generator instead of electricity. Within this chapter an AHP cycle working with the ThermoPump (see chapter 3.1) as well as the working principal of the pump itself (see chapter 3.2) is explained.

2.1 AHP cycle working with the ThermoPump

Figure 1 shows a flow sheet of a conventional single-stage AHP cycle with an electrically driven solution pump and Figure 2 a single-stage AHP cycle operated by the ThermoPump. In comparison, the cycles are quite similar, but a defined portion of the vaporised high pressure refrigerant from the generator (m_{pump}) is used to drive the ThermoPump, as shown in Figure 2. After "transferring the pump energy" m_{pump} is expanded to the low pressure level and is absorbed by the poor solution in the absorber together with the refrigerant mass flow coming from the evaporator (m_{ref}). An additional heat input to the generator (Q_{Pump}) is required to generate the refrigerant vapour needed to drive the ThermoPump (m_{Pump}). Thus, if the ThermoPump is used the heat demand of the generator (Q^{*}_{GEN}) increases in comparison to a system using an electrical solution pump (Q_{GEN}) at the same evaporator capacity (Q_0) due to the fact that m_{pump} does not pass the refrigerant cycle. This fact has also a negative influence on the efficiency of the AHP. However, thermal energy can be used to drive the ThermoPump instead of electricity, which offers the possibility to operate the solution pump with low-ex energy.

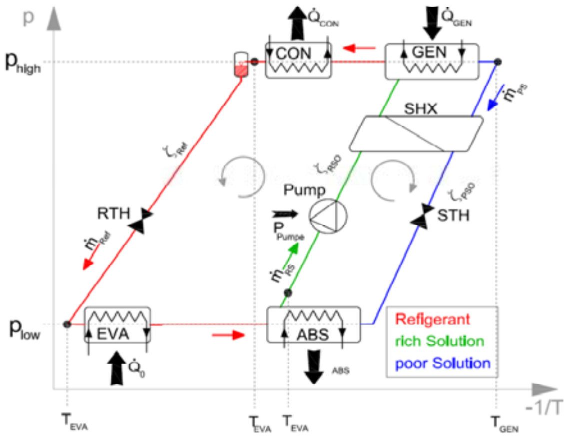

Figure 1: Principal drawing of a single-stage AHP with an electrical solution pump (Zotter & Rieberer 2013)

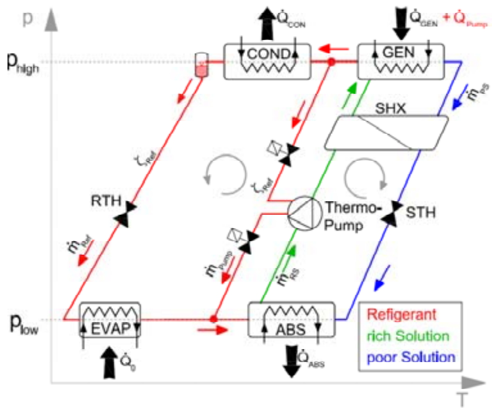

Figure 2: Principal drawing of a single-stage AHP with the ThermoPump (Zotter & Rieberer 2013)

2.2 The working principle of the ThermoPump (Zotter and Rieberer 2014)

The ThermoPump (see Figure 3) is a vessel consisting of following three chambers separated from each other by diaphragms:

- “Working chamber” contains the driving refrigerant vapor.
- “Pumping chamber” contains the rich solution.
- “Returning chamber” is essential for the function of the ThermoPump.

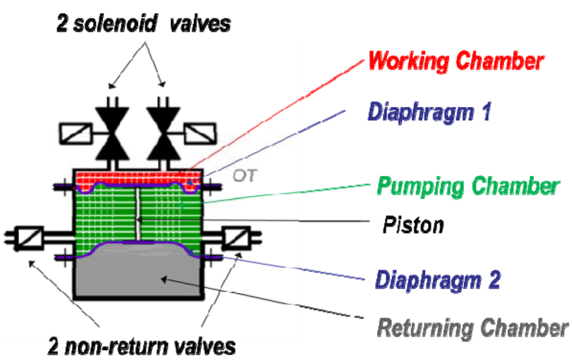

Figure 3: Principal drawing of the ThermoPump (Zotter and Rieberer, 2014)

The ThermoPump is basically an intermittently working diaphragm pump using two non-return valves (one at the inlet and one at the outlet of the pumping chamber) to ensure the flow direction of the rich solution from the absorber to the generator and two solenoid valves. One solenoid valve is connected to the generator (high pressure level) and the other solenoid valve to the absorber (low pressure level). The charging and discharging of the working chamber with the driving refrigerant mass flow of the ThermoPump can be controlled by opening one of two solenoid valves.

The pressure in the pumping chamber is be higher than the high side pressure or lower than the low side pressure of the AHP due to a special design of the ThermoPump, which is necessary to overcome the pressure losses inside the AHP. The geometry of the ThermoPump, majorly the volume ratio of the effective pumping to the working chamber ($V_{PC,eff}/V_{WC}$), determines the potential to overcome pressure losses (Δp_{losses}) and influences Λ_{eff}. But, unfortunately a geometry which allows to overcome a high Δp_{losses} leads to a low Λ_{eff}, which results in a lower COP^{*}_{C} of the AHP (see Eq. (5)).

The ”discharging” and “charging” of the ThermoPump with rich solution take place temporally one after another. When the pumping chamber is completely filled with rich solution at low pressure level, the discharging tact (see Figure 4) starts by opening the solenoid valve to the generator while the solenoid valve to the absorber (SV1) is closed (stroke… x = 0). Therefore a defined portion of refrigerant vapor at high pressure level flows inside the working chamber. Due to the pressure difference between the pumping and working chamber the rich solution is pressed through the non-return valve (NRV2) into the generator (full stroke). After the discharge of the rich solution out of the pumping chamber of the ThermoPump the solenoid valve to the generator (SV1) is closed and the “charging tact” (see Figure 5) starts by opening the solenoid valve to the absorber (SV2). The refrigerant vapor inside the working chamber flows out to the absorber of the AHP and the pressure level in the working chamber decreases to low side pressure. Due to the pressure difference between the pumping and working chamber rich solution from the absorber is sucked through NRV1 into the ThermoPump. When the pumping chamber is filled again with rich solution, SV2 is closed and the discharging tact starts again by opening SV1 (x = 0). (Zotter and Rieberer 2014)

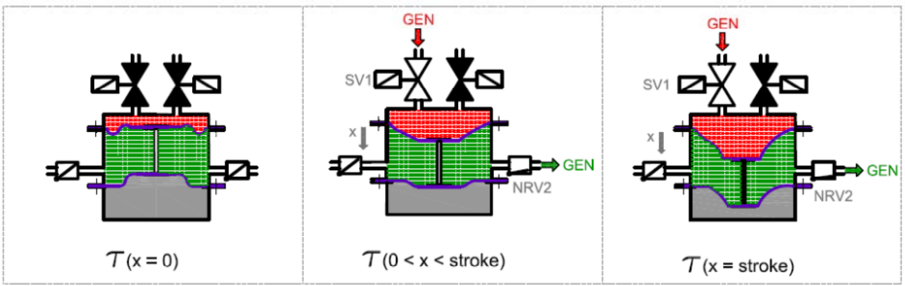

Figure 4: Principal drawing of the discharging tact of the ThermoPump (Zotter and Rieberer 2014)

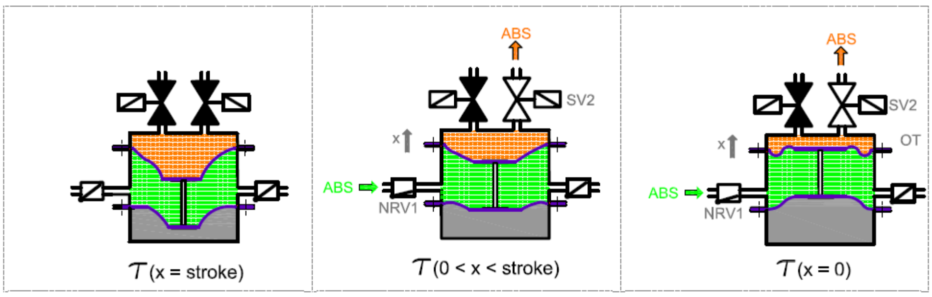

Figure 5: Principal drawing of the charging tact of the ThermoPump (Zotter and Rieberer 2014)

3 ENERGETIC EVALUATION OF THE AHP CYCLE WORKING WITH THE THERMOPUMP

In order to evaluate the influence of the ThermoPump on the Coefficient of Performance (COP), which is used to evaluate the efficiency of a single-stage AHP-cycle compared to an electrically solution pump have been investigated by means of simulation and experimental investigation. The most important figures for the evaluation are explained in this chapter.

3.1 AHP working with an electrically driven solution pump

The COP_C of an AHP for cooling purpose using an electrical solution pump is given in Equation (1), without considering the different exergetic values of the driving sources, i.e. heating capacity of the generator (Q_{GEN}) and electricity consumption of the solution pump (P_{Pump}).

$$COP_C = \frac{\dot{Q}_0}{\dot{Q}_{GEN} + P_{Pump}} \tag{1}$$

The electrical power consumption of the solution pump (P_{Pump}) is influenced by the theoretical pump power ($P_{Pump,theo}$) and the efficiency of the pump (η_{Pump}), as given in Equation (2).

$$P_{Pump} = \frac{P_{Pump,theo}}{\eta_{Pump}} = \frac{\dot{V}_{RS} \cdot (p_{high} - p_{low} + \Delta p_{losses})}{\eta_{Pump}} = \frac{1}{\eta_{Pump}} \cdot f \cdot \frac{\dot{m}_{ref}}{\rho_{RS}} \cdot (p_{high} - p_{low} + \Delta p_{losses}) \tag{2}$$

However, at that point it has to be mentioned that, P_{Pump} amounts only a few percent of the generator capacity (Q_{GEN}) of an AHP, which mainly depends on the circulation ratio (f), see Equation (3).

$$\frac{P_{Pump}}{\dot{Q}_{GEN}} = f \cdot \frac{1}{\eta_{Pump}} \cdot \frac{1}{COP_C} \cdot \frac{1}{\Delta h_0} \cdot \frac{1}{\rho_{RS}} \cdot (p_{high} - p_{low} + \Delta p_{losses}) \tag{3}$$

As defined in Equation (4) f is the ratio of the rich solution to the refrigerant mass flow rate. Based on a mass and species balance of a single-stage AHP with electrical solution pump f can be expressed by means of the ammonia-concentration of the refrigerant (ξ_{ref}), the rich (ξ_{RS}) and poor solution (ξ_{PS}). I.e., f depends mainly on the so called degassing ratio ($\Delta\xi$), which is the concentration difference between rich (ξ_{RS}) and poor solution (ξ_{PS}) and it is determined by the operating conditions of the AHP.

$$f = \frac{\dot{m}_{ref}}{\dot{m}_{RS}} = \frac{\xi_{ref} - \xi_{PS}}{\xi_{RS} - \xi_{PS}} \sim \frac{1}{\Delta\xi} \tag{4}$$

3.2 AHP working with the ThermoPump

The COP for an AHP working with the ThermoPump (COP_C^*) can be defined according to Eq. (5), as additional thermal capacity is required for the operation of the pump instead of electrical power.

$$COP_C^* = \frac{\dot{Q}_0}{\dot{Q}_{GEN}^*} \approx \frac{\dot{Q}_0}{\dot{Q}_{GEN} + \dot{Q}_{Pump}} \tag{5}$$

From an exergetic point of view, the minimal heat capacity to drive the solution pump ($Q_{Pump,min}$) has to be higher than $P_{Pump,theo}$ considering the temperature levels of the heat input (T_{GEN}) and output (T_{ABS}) of the solution cycle of the AHP, which is indirectly the power process of the ThermoPump (see Bosjankovic 1960). For example, at a generator temperature of 80°C and an absorber temperature of 30°C, $Q_{Pump,min}$ is about 7 times of $P_{Pump,theo}$ according to Eq. (6) by neglecting the temperature glides of the desorption and absorption process.

$$\dot{Q}_{Pump,\min} = \frac{P_{Pump,theo}}{\left(1 - \frac{T_{ABS}}{T_{GEN}}\right)} = P_{Pump,theo} \cdot \left(\frac{T_{GEN}}{T_{GEN} - T_{ABS}}\right) \tag{6}$$

Considering the different exergetic qualities of thermal (Q_{Pump}) and electrical power (P_{Pump}), for the same cooling capacity *(Q_0)* an AHP using the ThermoPump requires a higher generator capacity *(Q^*_{GEN} > $Q^*_{GEN,min}$)* than using an electrical solution pump *(Q_{GEN})*.This fact results in a lower $COP^*_{C,min}$ *respectively in a lower* COP^*_C *than* COP_C at the same operating conditions of the AHP, if η_{Pump} is higher than the ratio of $P_{Pump,theo}$ to $Q_{Pump,min}$. From an energetic point of view the relative negative influence on the COP *(ΔCOP_C and $\Delta COP_{C,min}$*, see Eq. (7) and (8)) of the AHP using the ThermoPump is one of the most important figures for the evaluation of the pump.

$$COP^*_{C,\min} = \frac{\dot{Q}_0}{\dot{Q}_{GEN} + \dot{Q}_{Pump,\min}} \leq COP^*_C \tag{7}$$

$$\Delta COP_{C,\min} = \frac{COP_C - COP^*_{C,\min}}{COP_C} \cdot 100\% \leq \Delta COP_C = \frac{COP_C - COP^*_C}{COP_C} \cdot 100\% \tag{8}$$

ΔCOP$_C$ mainly depends on the amount of refrigerant mass flow required to drive the ThermoPump (m_{Pump}), due to the fact, that m_{Pump} doesn't pass the refrigerant cycle of the AHP (see Figure 2). m_{Pump} is determined by the required rich solution mass flow (m^*_{RS}), the density ratio of the refrigerant vapour coming from the generator to the rich solution coming from the absorber ($\rho_{ref,vapour}/\rho_{RS}$) and the so-called "effective delivery efficiency" (Λ_{eff}), which is a kind of volumetric efficiency mainly given by the internal geometry of the ThermoPump (see Eq. (9)).

$$\dot{m}_{Pump} = \dot{m}^*_{RS} \cdot \frac{\rho_{ref,vapour}}{\rho_{RS}} \cdot \frac{1}{\Lambda_{eff}} \tag{9}$$

Nevertheless, the required rich solution mass flow, which has to be delivered by the Thermopump (m^*_{RS}), is higher than the one, which has to be delivered by the electrical solution pump ($m_{RS,}$) for the same cooling capacity of the same AHP at the same operating conditions. The reason for this is that m_{ref} as well as m_{Pump} have to be desorbed in the generator. Hence, m^*_{RS} is given by the required refrigerant mass flow through the heat pumping cycle (m_{ref}) and the circulation ratio (f^*) for the AHP using the ThermoPump. f^* is higher than the circulation ratio for an AHP using an electrical solution pump (f). f^* is defined in Eq. (10) based on to the total mass- and NH_3 mass-balance for a single-stage AHP using the ThermoPump.

$$f^* = \frac{\dot{m}^*_{RS}}{\dot{m}_{ref}} = \frac{\xi_{ref} - \xi_{PS}}{\left(\xi_{RS} - \xi_{PS}\right) - \frac{1}{\Lambda_{eff}} \cdot \frac{\rho_{ref,vapor}}{\rho_{RS}} \left(\xi_{ref} - \xi_{PS}\right)} \tag{10}$$

3.3 Theoretical influence of the ThermoPump on the COP of an AHP

To determine the minimal influence of the ThermoPump on the COP of an AHP ($\Delta COP_{C,min}$) from a theoretical point of view, an existing semi-physical simulation model (Hannl & Rieberer 2012) for a single-stage AHP cycle (according to Figure 1) in EES (EES 2011) has been enhanced by means of the second law of thermodynamics (Zotter and Rieberer 2014). Therefore Eq.(6), (7) and (8) (see chapter 3.2) has been implemented in this existing model of the AHP cycle (Hannl & Rieberer 2012), which is based on total mass, ammonia mass and energy balances for each component as well as on experimental results of the PinkChiller PC19 to determine the UA-value of each heat exchanger, and the mass transport characteristic of generator and absorber, and the control algorithm for the low-side pressure and the rich solution mass flow rate. The input parameters of this simulations model were the hot (V_{HOT}), cooling (V_{COOL}) and cold (V_{COLD}) water flow rates and the hot water inlet ($t_{HOT,in}$), coolling water inlet ($t_{COOL,in}$) and the cold water outlet temperature ($t_{COLD,out}$). The water flow rates have been set according table 1 and the above mentioned temperatures have been varied.

According to the simulation results $\Delta COP_{C,min}$ amounts about 2 to 20% (see Figure 6). $\Delta COP_{C,min}$ and the specific circulation ratio (f) are strongly depending on $t_{COOL,in}$ at a certain $t_{HOT,in,}$ and both functions show a similar dependency on $t_{COOL,in}$ and $t_{HOT,in}$. As already mentioned, f depends mainly on the degassing ratio. A low degassing ratio results in a higher $\Delta COP_{C,min}$, due to the fact that more rich solution has to be pumped for the same cooling capacity.

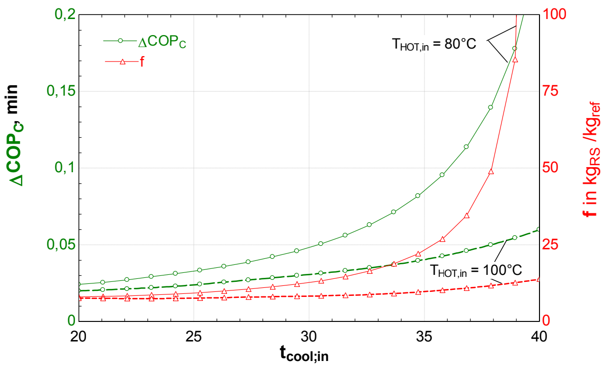

Figure 6: Estimation of $\Delta COP_{C,min}$ and f vs. $t_{COOL,in}$ for two different driving temperatures of the AHP ($t_{HOT,in}$) and a constant cold water outlet temperature ($t_{COLD,out}$) of 15°C

4 EXPERIMENTAL ANALYSIS

A first prototype of the ThermoPump has been designed (see Figure 8) based on a very detailed theoretical analysis of the concept by means of fluid- and thermodynamics as well as kinematics. After this, the prototype has been integrated in a test bench and experimental analyzed at different operating conditions concerning operational behavior, cooling capacity and COP.

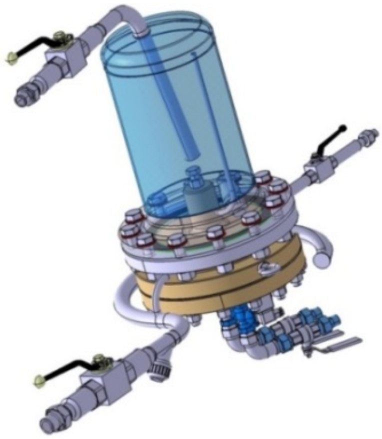

Figure 7: 3D-model of the ThermoPump (Handler, 2012)

4.1 Test bench

An AHP available on the market (PinkChiller PC19) has been chosen as test bench of the prototype. It offers the possibility for a direct comparison of the same AHP working with the ThermoPump and an electrical solution pump. At that point it has to be mentioned, that the prototype has been integrated into the PinkChiller (see Figure 9) without any extra efforts regarding the design of the AHP. The PinkChiller PC 19 has a nominal cooling capacity of ca. 19 kW_{th} working with the existing solution pump, which is an oil diaphragm pump with a maximal delivery rate of ca. 600 l/h. The electrical power consumption of this existing solution amounts only about 2% of the generator capacity (Q_{GEN}), depending on the operation conditions.

Figure 8: Picture of the ThermoPump test bench (PinkChillerPC19) with the prototype of the Thermo-Pump (red circle) (Zotter & Rieberer 2013)

The PinkChiller including the ThermoPump has been equipped with the required measurement equipment and integrated to a heat sink / heat source system in the laboratory of the Institute, and analyzed at different operating conditions in comparison to the electrical solution pump. For that, the hot, cooling and cold water temperature level has been varied at constant volume flow rates according to table 1. All energy flows from and to the PinkChiller PC 19 have been measured at the internal AHP cycle as well as at the external hot, cooling and cold water cycles. In order to achieve smaller measuring uncertainties for the evaluation, the measured data of the external cycles have been chosen. After calibration of the measurement equipment the uncertainties of the used resistance temperature sensors amount only ± 0.05 K, of the volume flow meter only about ± 0.01 m³/h (see table 1) and of mass flow meters only ± 0.1 % of the measured value. (Zotter and Rieberer 2014)

Table 1: Hot, cooling and cold water flow rates of the test rig

V_{HOT}	V_{COOL}	V_{COLD}
2.4 ± 0.007 m^3/h	3.6 ± 0.012 m^3/h	2.9 ± 0.015 m^3/h

4.2 Results

According to the experimental investigations the pump worked without any relevant operating-limits or negative influence on the dynamic behaviour of the AHP. Furthermore an adequate control strategy has been developed by modulating the switching time of the two solenoid valves of the pump (Zotter and Rieberer 2013). As shown in Figure 9 there is one switching time, for which the delivered mass flow has a maximum. But the optimum switching time is different for each operating point of the AHP.

However, the measurement results showed that the delivered solution mass flow of the ThermoPump was rather low, due to the additional pressure losses caused by the measurement equipment. To compare both pumps to each other, the delivery mass flow of the existing electrical solution pump of the PinkChiller has been also reduced to comparable values.

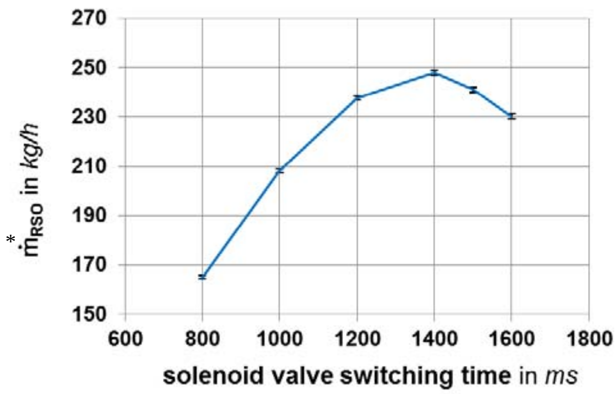

Figure 9: Measured rich solution mass flow delivered by the ThermoPump ($\dot{m}^*_{RSO}$) vs (charging) switching time of SV2 at a constant (discharging) switching time of SV1 at 1.4 s for the operating point: t_{HOT_in} = 90°C, t_{COOL_in} = 28°C and t_{COLD_out} = 15°C; (Zotter and Rieberer 2014)

For the AHP manufacturer besides the costs the influence of the ThermoPump on their plant is mainly of interested. Therefore, in addition to the operational behavior of the ThermoPump, the cooling capacity and the COP of the PinkChiller using both pumps has been investigated in detail within this experimental analysis and compared to each other. Figure 11 shows the measured cooling capacity (Q_0) of the PinkChiller PC19 using the electrical solution pump (solid lines) and the ThermoPump (dashed lines) at different hot and cooling water temperature levels and a cold water temperature (t_{cold_out}) of 15 °C. Based on the fact that the refrigerant mass flow required for driving the ThermoPump does not pass the refrigerant cycle, the cooling capacity of the PinkChiller is lower using the ThermoPump. Furthermore, based on the same fact, the COP of the AHP is reduced by at least 15% by the ThermoPump (Figure 10). Higher cooling water temperatures leads to higher ΔCOP_C due to the fact that, f* increases at higher cooling water temperatures (t_{cool_in}) and a high *f** leads to a high demand of refrigerant mass flow to drive the ThermoPump.

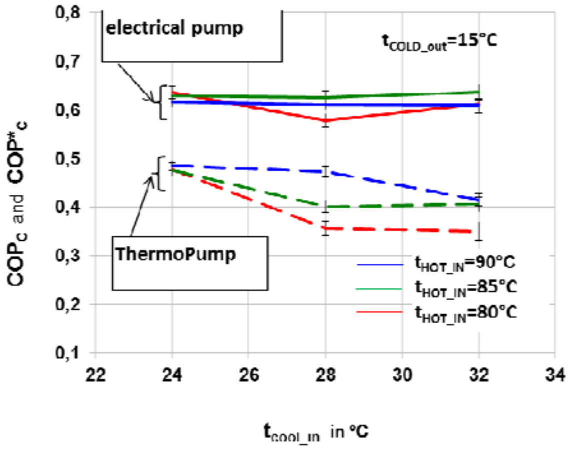

Figure 10: Comparison of the COPs of the PinkChiller PC19 using the electrical solution Pump (COP$_C$) and the ThermoPump (COP$^*_C$) at different operating conditions (Zotter & Rieberer 2013)

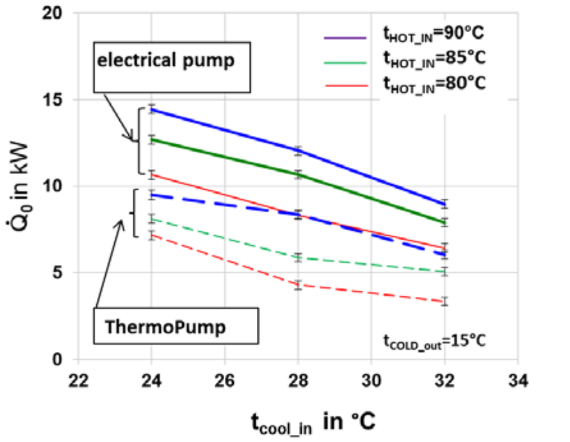

Figure 11: Measured cooling capacity of the PinkChiller PC19 using the electrical solution pump (solid lines) and the ThermoPump (dashed lines) at different operating conditions (Zotter & Rieberer 2014)

Based on the measured results it can be pointed out, that the so called energetic quality grade of the ThermoPump (υ^*) amounts not more than 30%. As defined by Equation (11), υ^* is the ratio of the minimal required thermal power to drive the ThermoPump according to theory ($Q_{Pump,min}$, see chapter 3.2 and 3.3) to the measured thermal power consumption of the pump (Q_{Pump}) at the same temperature level.

$$\upsilon^* = \frac{\dot{Q}_{Pump,\min}}{\dot{Q}_{Pump}} = \frac{\dot{Q}^*_{Gen,\min} - \dot{Q}_{Gen}}{\dot{Q}^*_{Gen} - \dot{Q}_{Gen}} \qquad (11)$$

Based on this rather low energetic quality grade of the ThermoPump, there is still room for improvements concerning the layout and design of the prototype to come up to the theoretical potential of a thermal driven solution pump.

5 OPTIMIZATION POTENTIAL ACCORDING TO SIMULATIONS

In order to determine further potential for increasing the COP$^*_C$ of the AHP using the ThermoPump the detailed simulation model covering all relevant geometric parameters of the ThermoPump has been enhanced and validated with experimental data.

After including the internal heat and pressure losses of the ThermoPump in the simulation model, the simulations fit rather well to the experiments (see in Figure 12).

Additional potential for improving COP^*_C has been detected. E.g., so far the experimental results are based on an operation with non-optimal switching times of the solenoid valves, which results in a higher demand on refrigerant vapour for driving the pump than with optimal switching times. Furthermore, reduced internal pressure and heat losses as well as an optimization of the design may increase COP^*_C.

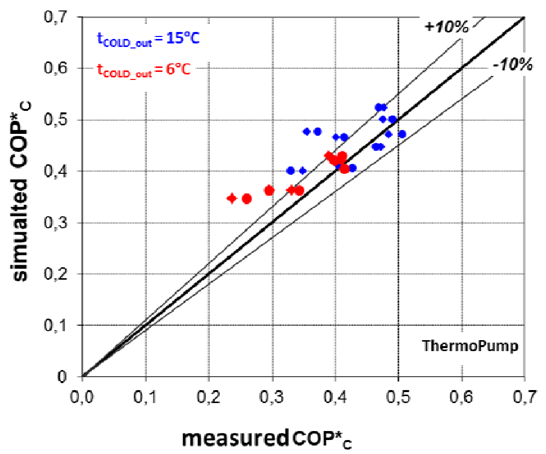

Figure 12: Measured COP*$_C$ vs. simulated COP*$_C$ values for the PinkChiller PC19 using the ThermoPump

6 CONCLUSIONS

A thermally driven solution pump, i.e. a pump driven by a power process instead of electricity, is of interest because only a few electrical solution pumps are available on the market, which are suitable for NH_3/H_2O-AHPs with an evaporator capacity below 20 kW. Within this work a new concept of a thermally driven solution pump, the so called ThermoPump, which is driven by a portion of refrigerant mass flow, has been developed and experimentally investigated.

From a technical point of view, a major advantage of this concept compared to other thermally driven solution pumps is, that the ThermoPump can be easily integrated in an AHP (e.g. PinkChiller PC19) "without" special efforts regarding the layout of the system. Furthermore, according to the experimental investigations of the first ThermoPump prototype integrated in a commercially available AHP it can be pointed out, that this pump operates

- without any leakage problems
- without relevant operating-limits
- without negative influence on the dynamic behavior of the AHP

Within this work also an adequate control strategy has been developed by modulating the switching time of the two solenoid valves. However, the measured COP_C is lower by at least 15% in comparison to an electrical pump. In this context it should be mentioned that the additionally required generator heat could be delivered by e.g. CO_2-free thermal energy instead of electricity.

However, from an ecological point of view, also the additional electricity consumption for the rejection of more waste heat has to be accounted for. For example, using the ThermoPump to drive an AHP with a nominal cooling capacity of about 19 kW_{th}, it saves about 0.5 kW_{el} in comparison to the electrical pump for delivering nominal cooling capacity. Nevertheless, the decrease of the COP results in an increase of waste heat. A ΔCOP_C of e.g 15% leads to a decrease of the COP_C from e.g. about 0.6 to a COP^*_C of 0.51. This means that about 5.5 kW_{th} of additional waste heat of a chiller with 19 kW_{th} cooling capacity have to be rejected to the ambient. Assuming a value of 0.05 kW_{el} for the specific electrical consumption of the ventilation system of a dry cooler per kW_{th} of rejected heat (Nienborg et al. 2013) an additional electrical consumption of 0.3 kW_{el} is required to reject the additional waste heat. Hence, it can be pointed out, that the electrical power consumption of the overall AHP cooling system can be reduced using the ThermoPump instead of an electrical pump as long as ΔCOP_C is lower than 25%.

Furthermore, from an economical point of view, the production cost of the ThermoPump – expected for a small production series of 100 pieces – will nearly amount the same as the cost of a comparable electrical solution pump.

However, additional potential for increasing the COP^*_C and reducing the production cost has been detected within further analysis. Finally it should be emphasized that by this optimization (e.g. reducing internal pressure and heat losses as well as improving the pump layout), the ThermoPump concept could offer a very interesting alternative to a "standard" solution pump for small-capacity AHPs.

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