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TECHNICAL AND ECONOMIC WORKING DOMAINS OF INDUSTRIAL HEAT PUMPS: PART 2 - AMMONIA-WATER HYBRID ABSORPTION-COMPRESSION HEAT PUMPS

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Abstract The ammonia-water hybrid absorption-compression heat pump (HACHP) is a relevant technology for industrial heat supply, especially for high sink temperatures and high temperature glides in the sink and source. This is due to the reduced vapour pressure and the non-isothermal phase change of the zeotropic mixture, ammonia-water. To evaluate to which extent these advantages can be translated into feasible heat pump solutions, the working domain of the HACHP is investigated based on technical and economic constraints. The HACHP working domain is compared to that of the best possible vapour compression heat pump with natural working fluids. This shows that the HACHP increases the temperature lifts and heat supply temperatures that are feasible to produce with a heat pump. The HACHP is shown to be capable of delivering heat supply temperatures as high as 140 °C and temperature lifts up to 60 K, all with economical benefits for the investor.

Key Words Industrial heat pumps, working domain, absorption-compression heat pump, economic evaluation, natural refrigerants

1. INTRODUCTION

The hybrid absorption-compression heat pump (HACHP) or vapour compression heat pump with solution circuit is based on the Osenbrück cycle. The advantage of the HACHP is the reduction of vapour pressure and the temperature glides of the absorption and desorption processes, (Berntsson and Hultén, 1999, 2002). The reduction of vapour pressure allows the design of high temperature heat pumps with standard pressure refrigeration components, (Brunin et al., 1997; Jensen et al., 2014). The non-isothermal phase change allows the temperature profiles to match those of the external circuits, thereby attaining a reduction of the entropy generation caused by heat transfer over a finite temperature difference. Consequently, the HACHP is a measure of approaching the Lorenz cycle and thereby increasing the efficiency compared to a vapour compression heat pump (VCHP). These advantages make the HACHP a relevant technology, for industrial processes that require high temperature and large temperature glides. Brunin et al. (1997) evaluates the working domain of the HACHP based on maximum pressure, coefficient of performance (COP) and volumetric heat capacity. This shows that the HACHP can attain heat supply temperatures up to 140 °C with a maximum pressure of 20 bar. In recent years the maximum pressure for standard refrigeration components have reached 28 bar and further high pressure components have entered the market for R717, allowing a maximum pressure of up to 50 bar. Therefore it is relevant to re-evaluate the relevant working domains with the higher attainable pressures. Further Brunin et al. (1997) does not include the compressor discharge temperature as a constraint but, as discussed in Jensen et al. (2014) the compressor discharge temperature is a limiting factor for the development of a high temperature HACHP.

Ommen et al. (2011) compared the economy of the HACHP to the VCHP and showed that the HACHP is a competitive technology. However, the study did not consider the degradation of heat transfer coefficients caused by the application of a zeotropic mixture working fluid. As discussed in Radermacher and Hwang (2005) the two phase heat transfer coefficient is reduced due to mass diffusion resistance in the zone surrounding the liquid-vapour interface, which tends to be depleted of the volatile component. Consequently, the HACHP requires a larger heat transfer area compared to a pure R717 VCHP. Further the reduced vapour pressure of the mixture increases the specific volume of the low pressure vapour and consequently entails the need for an increased compressor displacement volume. It is therefore relevant to evaluate, whether this increased investment in heat transfer area and compressor can be justified by the increased efficiency of the heat pump.

Both Brunin et al. (1997) and Ommen et al. (2011) have restricted their HACHP analysis to a limited number

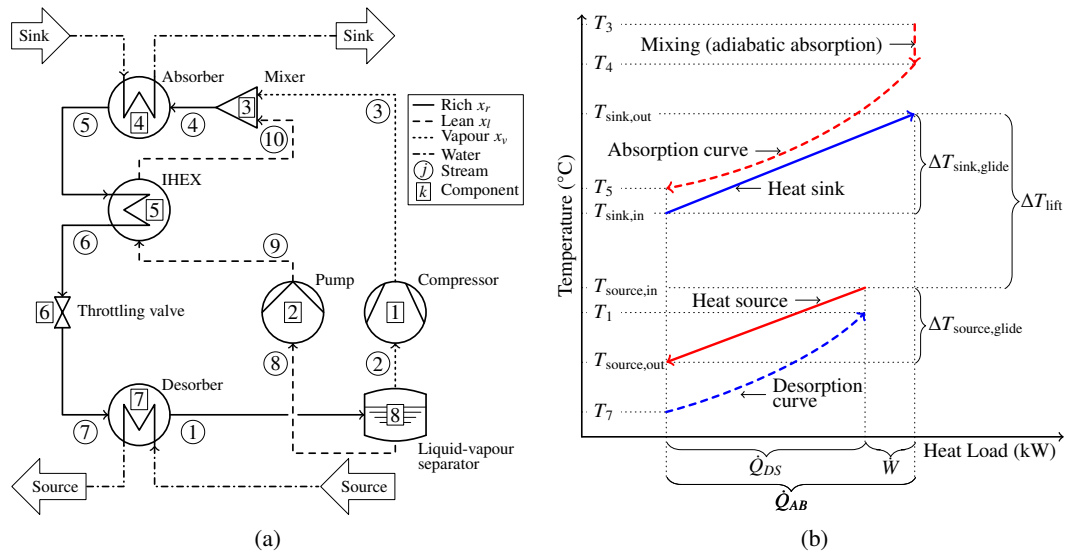


Figure 1. (a) Principle sketch of the HACHP, (b) HACHP process sketched in a temperature - heat load diagram of ammonia mass fractions and use a fixed value of concentration difference (difference between ammonia concentration in the rich mixture and the circulated lean mixture). However, as indicated by the results of Jensen et al. (2014): it is essential to determine the right combination of ammonia mass fraction and circulation ratio to attain a cost effective high temperature HACHP that complies with the constraints of commercially available components.

This study investigates the working domain of the HACHP based on a comprehensive analysis of the effect of varying the ammonia mass fraction and circulation ratio on both the investment and operating costs as well as the operating conditions and technological limitations. Finally the cost of the HACHP will be compared to the cost of the best available VCHP technology presented in Ommen et al. (2014).

2. METHOD

The same approach as used in Ommen et al. (2014) was applied to the investigation of the HACHP working domain. The working domain was evaluated at three different sink-source glide temperature differences. The working domain was bounded by several technical and economic constraints. The technical constraints are governed by thermodynamic behaviour of the working fluid and the limits of commercially available components, these limits are discussed in detail in section 2.3. The economic viability of the HACHP was defined as the configurations that attain a positive Net Present Value (NPV) when replacing a natural gas boiler as the source of heat supply. The basis of this economic evaluation is described in section 2.4. The HACHP was evaluated with a supplied heat load of 1000 kW, 3500 operating hours per year and a technical lifetime of 15 years, the same as in Ommen et al. (2014). Pressure drop in pipes and fittings and compressor heat loss were neglected.

2.1. Hybrid absorption-compression heat pump

The general layout of the HACHP can be seen in Fig. 1a, a principal sketch of the process is presented in the temperature-heat load diagram in Fig. 1b. The heat transfer fluid of the sink and source was assumed to be liquid water, pressurized to prevent evaporation.

A numerical model of a HACHP has been developed in Engineering Equation Solver (EES) (F-Chart, 1992). The thermodynamic properties of the ammonia-water mixture were calculated using equations of state developed by Ibrahim and Klein (1993). Transport properties were calculated using correlations developed by El-Sayed (1985) and Conde (2004). Each component was modelled based on steady state mass and energy balances.

The rich ammonia mass fraction x_r and circulation ratio f were inputs to the model. The circulation ratio was defined as the ratio between the mass flow rate of the rich solution and the lean mixture. Ammonia mass fractions between 0.3 and 0.9 and circulation ratios between 0.2 and 0.9 were investigated for the working domain.

The performance of the HACHP was evaluated based on given, constant values of efficiencies for the com-

Table 1. Applied heat transfer and pressure drop correlations for the absorber, desorber and internal heat exchanger

k^{th} -comp.	Media		Heat transfer	Pressure drop
(4) AB	H ₂ O		Martin (1996)	Martin (1996)
(4) AB	NH ₃ /H ₂ O	vapour only:	Martin (1996)	
		liquid film:	Yan et al. (1999)	
		two-phase:	Silver (1947); Bell and Ghaly (1972)	Yan et al. (1999)
(6) IH	NH ₃ /H ₂ O		Martin (1996)	Martin (1996)
(8) DS	H ₂ O		Martin (1996)	Martin (1996)
(8) DS	NH ₃ /H ₂ O	liquid only:	Martin (1996)	
		two-phase:	Táboas et al. (2012)	Táboas et al. (2012)

pressor, pump and their electric motors ($\eta_{is}=0.80$, $\eta_{vol}=0.80$ and $\eta_{el}=0.95$), pinch point temperature differences for the absorber and desorber ($\Delta T_{pp}=5$ K) and heat exchanger effectiveness for the internal heat exchanger ($\epsilon=0.90$).

2.2. Calculation of heat transfer and pressure drop

All three heat exchangers are assumed to be of a plate type with a chevron corrugation. The use of plate heat exchangers for industrial processes have increased significantly over the last decades and is therefore assumed to be the preferred option. The applied plate dimensions correspond to the V120T (absorber and desorber) and the V10T (internal heat exchanger), SWEP (2014).

Detailed heat transfer coefficient correlations have been applied to both the absorber, desorber and internal heat exchanger. The applied heat transfer and pressure drop correlations are listed in Table 1. Both heat transfer coefficients and friction factors of the two-phase flows depend on the vapour quality. Therefore, the absorber and desorber were discretized to account for this dependency. The logarithmic mean temperature difference was averaged over 50 steps of the absorber and desorber calculation. This is done to account for that enthalpy and temperature are not proportional during absorption and desorption.

2.3. Compressors and operating conditions

The maximum heat supply temperature achievable by a heat pump is restricted by the maximum pressure of the applied compressor technology and the thermodynamic behaviour of the working fluid. Further the maximum achievable temperature lift, i.e., the difference between the source inlet temperature and the sink outlet temperature, is restricted by the compressor discharge temperature, for positive displacement compressors. This is due to the thermal stability of the lubricating oil and to reduce wear by thermal stress. As discussed in Ommen et al. (2014), the compressor discharge temperature is of special concern for ammonia compressors.

In the present study two compressor technologies were investigated: low pressure ammonia (Ommen et al. (2014): Type 3) and high pressure ammonia (Ommen et al. (2014): Type 4). The low pressure ammonia compressor (LPR717) has a maximum pressure of 28 bar and can sustain discharge temperatures up to 180 °C. The LPR717 is available in a capacity range from 5-180 m³/h. The high pressure ammonia compressor (HPR717) has a maximum pressure of 50 bar and can sustain discharge temperatures up to 180 °C. The HPR717 is available in a capacity range from 90-200 m³/h.

Generally the water content in the compressed vapour stream is negligible but as shown by Jensen et al. (2014): for low ammonia mass fractions the water content can be substantial for some operating points. These solutions should be avoided and therefore solutions with a vapour ammonia mass fraction $x_v < 0.95$ were deemed infeasible.

2.4. Economic evaluation

Product equipment cost (PEC) functions have been developed based on Danish intermediate trade business prices H. Jessen Jürgensen A/S (2013) and individual producers Sørensen (2013). The cost function were constructed as proposed by Bejan et al. (1996). A number of assumptions have been made to estimate the total investment of the HACHP system. These are as follows:

- Total capital investment of a component is 4.16 higher than the PEC of the component Bejan et al. (1996).
- PEC for a compressor is a function of the compressor displacement volume.
- PEC for an electrical motor with a fixed efficiency is dependent only on the shaft power.

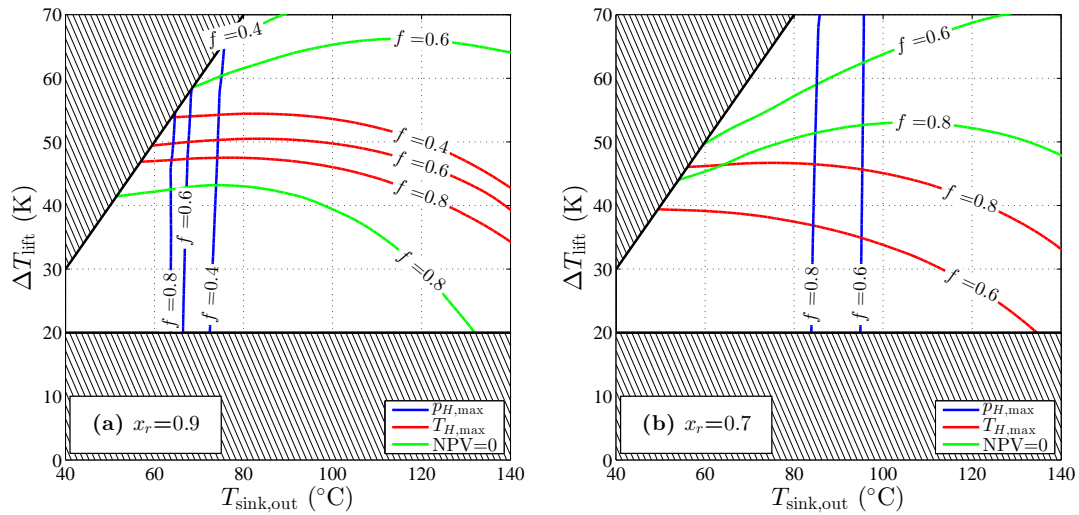


Figure 2. Feasible working domain boundaries for variable design parameters x_r and f , $\Delta T_{\text{source}}=10$ K, $\Delta T_{\text{sink}}=20$ K, compressor type LPR717

- PEC for a heat exchanger is a function of the heat exchange area.
- PEC of the pump is equivalent to its electrical motor.
- PEC of the expansion valve, mixer and the liquid vapour separator are neglected.

To determine the levelized capital cost rates, the capital recovery factor Bejan et al. (1996) was used. An interest rate of 7% and inflation rate of 2% were assumed, further the technical life time was set to 15 years. The maintenance cost rate was assumed to be 20% of the levelized capital cost rates, Bejan et al. (1996). Electricity and natural gas prices correspond to an industrial consumer in the Danish fiscal environment, prices were found in Danish Energy Agency (2013).

The criteria for the economic viability of the heat pump investment was a positive NPV when replacing a natural gas boiler as the source of heat supply. The cost of a natural gas boiler was evaluated with a gas burner efficiency of 90% and the without considering the gas burner investment.

3. RESULTS

The working domain of the HACHP is greatly influenced by the choice of the two design parameters x_r and f . Figure 2 shows an example of the working domain for several design options, Figure 2 (a) is for $x_r=0.9$ and $f=0.4, 0.6$ and 0.8 , (b) is for $x_r=0.7$ and $f=0.6$ and 0.8 . The example shown here is for a LPR717 compressor. The hatched areas define operating points where the analysed heat pump implementation is not applicable. The upper right corner is infeasible as this requires $T_{\text{source,out}} < 0$ °C which implies a phase change for the chosen heat transfer fluid. Operating points at which $T_{\text{sink,in}} < T_{\text{source,in}}$ are neglected as for these conditions, direct heat transfer should be applied first, as suggested by the principles of pinch analysis, (Bejan et al., 1996).

As may be seen some general trends can be concluded from these examples. A reduction of the ammonia mass fraction increases the range of available supply temperatures, such that the maximum heat supply temperature is increased from approximately $T_{\text{sink,out}}=75$ °C to $T_{\text{sink,out}}=95$ °C when reducing the ammonia mass fraction from $x_r=0.85$ to $x_r=0.65$. Further it can be observed that the reduction of ammonia mass fraction increases the compressor discharge temperature T_H . Therefore a reduction of the ammonia mass fraction reduces the range of feasible temperature lifts. Reducing x_r from 0.85 to 0.65 reduces the maximum lift from approximately $\Delta T_{\text{lift}}=54$ K to approximately $\Delta T_{\text{lift}}=46$ K. Reducing the ammonia mass fraction moves the economic constraint towards a lower temperature lift. This is due to the increased investment in heat transfer area and compressor displacement volume.

In order to assess the working domain of the HACHP technology as a whole a selection criteria must be derived to choose the best possible design at each operating point (defined by the heat supply temperature $T_{\text{sink,out}}$ and temperature lift ΔT_{lift}).

In the following the best possible design has been found for all operating points as the design that maximizes the NPV while still complying with the constraints on maximum pressure, maximum compressor discharge temperature and maximum vapour water content. The results are presented in Figure 3, on page 5. As may

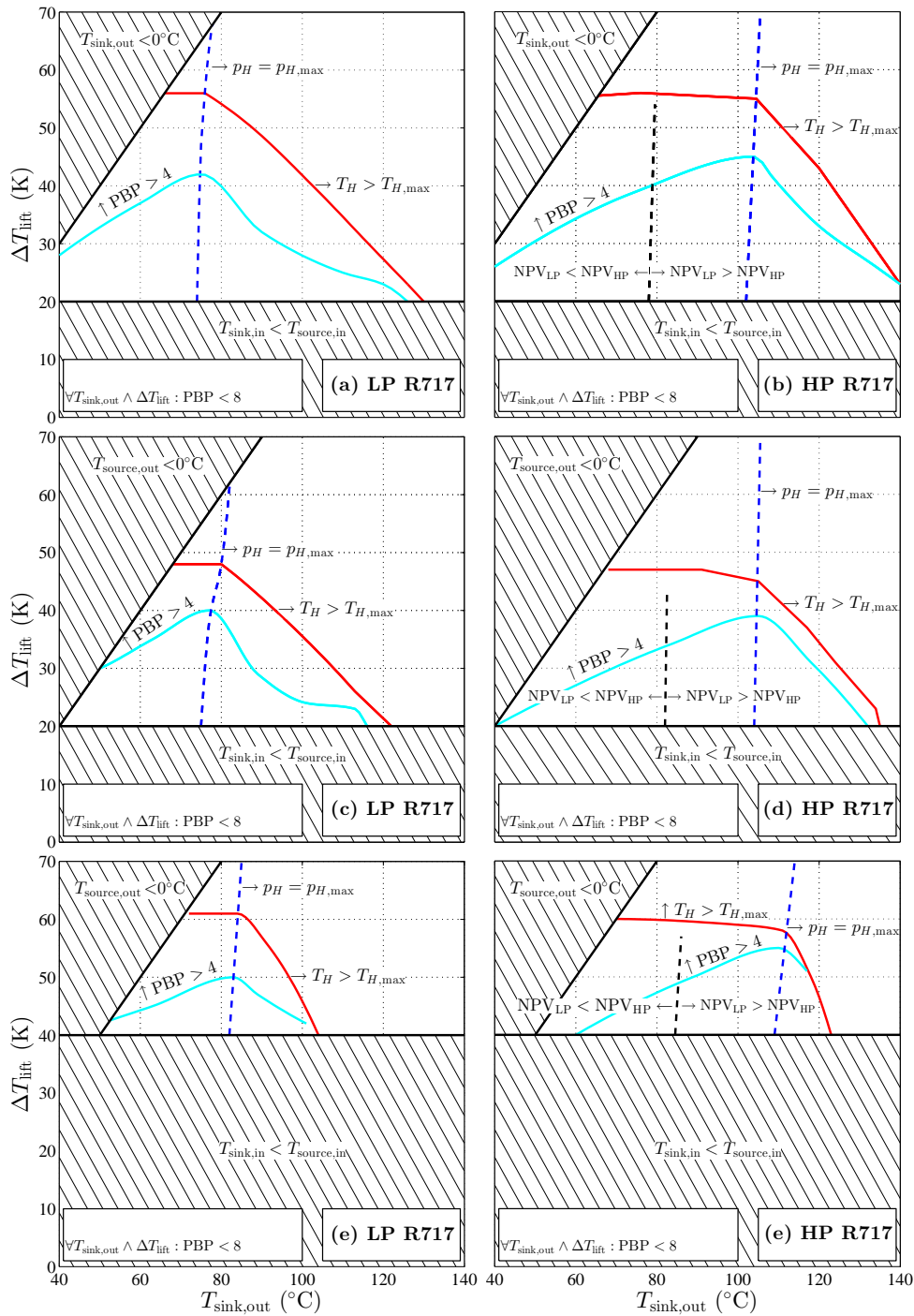


Figure 3. Working domains for HACHP using LPR717 and HPR717 compressors at three different sink source configurations

be seen from Figure 3 (a) the HACHP using the LPR717 compressor can deliver a maximum heat supply temperature of $\Delta T_{\text{sink,out}}=125\text{ }^{\circ}\text{C}$ and a maximum temperature lift of $\Delta T_{\text{lift}} = 55\text{ K}$. This with a sink-source configuration of $\Delta T_{\text{source}}=10\text{ K}$ and $\Delta T_{\text{sink}}=20\text{ K}$. When increasing the source glide to $\Delta T_{\text{source}}=20\text{ K}$ the maximum supply temperature is slightly reduced to $T_{\text{sink,out}}=120\text{ }^{\circ}\text{C}$ and the maximum lift is reduced to $\Delta T_{\text{lift}} = 48\text{ K}$, see Figure 3 (c). Increasing the sink glide to $\Delta T_{\text{sink}}= 40\text{ K}$ reduces the maximum temperature to $T_{\text{sink,out}}= 105\text{ }^{\circ}\text{C}$ but increases the maximum lift to $\Delta T_{\text{lift}} = 61\text{ K}$, see Figure 3 (e).

As seen the working domain of the best possible solutions all exhibit the same trend: the maximum temperature lift remains close to constant until a point indicated by the dashed blue line where after the maximum lift descends. This is because the solution that minimizes the cost is always the solution with the highest allowable ammonia mass fraction. Hence the solution to the left of the dashed blue line all have $x_r=0.90$ without compromising the pressure constraint. Moving to the right of the blue dashed line the ammonia mass fraction must

be reduced to comply with the pressure constraint. The minimal cost is still attained by the solution with the highest ammonia mass fraction and thus all solutions to the right of the dashed blue line have the maximum allowable pressure. This result is in alignment with conclusion of Berntsson and Hultén (1999, 2002). Figure 4a graphically presents the optimal choice of ammonia mass fraction for LPR717 and $\Delta T_{\text{source}}=10$ K and $\Delta T_{\text{sink}}=20$ K. Figure 4b shows the optimal choice of circulation ratio, here it may be seen that when ammonia mass fraction is reduced, the circulation ratio should be increased to minimize the cost.

By applying the HPR717 compressor, the high ammonia mass fraction can be retained to a higher range of supply temperatures, see Figures 3 (b), 3 (d) and 3 (f). Consequently higher heat supply temperatures can be reached with the HPR717 compressor. For $\Delta T_{\text{source}}=10$ K and $\Delta T_{\text{sink}}=20$ K, heat supply temperatures of $T_{\text{sink,out}}=140$ °C can be reached and still attain a simple pay back time (PBT) under 4 years. Increasing the source glide to $\Delta T_{\text{source}}=20$ K reduces the maximum heat supply temperature to $T_{\text{sink,out}}=135$ °C while increasing the sink glide to $\Delta T_{\text{sink}}=40$ K reduces the maximum heat supply temperature to $T_{\text{sink,out}}=122$ °C. As the investment in the HPR717 compressor is larger compared to the LPR717 it is not advantageous to use the high pressure solution for the low heat supply temperature range. The dashed black line in Figures 3 (b), 3 (d) and 3 (e) indicate where the low pressure solution is the preferable option. As may be seen this limit occurs shortly after the maximum pressure is attained for the LPR717 compressor.

Comparing the working domain of the HACHP to the working domain of the best possible VCHP solution presented in Ommen et al. (2014), it is clear that the HACHP expands the range of operating points at which heat pump implementation is technically feasible and economically viable. The HACHP allows both higher temperature lifts and higher heat supply temperatures. When comparing the life time cost of the HACHP to the best possible VCHP it is found that, under the examined conditions, the HACHP always has a lower cost than the VCHP. This is also the conclusion of Ommen et al. (2011) although the two technologies are only compared for a small range of temperature lift and heat supply temperature. Likewise, Berntsson and Hultén (1999, 2002) conclude that the HACHP heat pump is advantageous over the VCHP for sink-source glide over 10 K.

Figures 4c, 4d and 4e shows the relative percentage difference between the NPV of HACHP and VCHP, in the range of operating points where these compete. The cost difference ranges from approximately 5-30%. The highest difference is found when the HACHP is competing against the transcritical R744 heat pump with $\Delta T_{\text{source}}=10$ K and $\Delta T_{\text{sink}}=40$ K and for the low lift solutions of $\Delta T_{\text{source}}=10$ K and $\Delta T_{\text{sink}}=20$ K. The latter is caused by favourable heat transfer conditions in the desorber resulting in a slightly lower investment for the HACHP, while still attaining a higher COP.

4. DISCUSSION

Comparing R717 VCHP to the HACHP, the relative difference for most operating points are between 5-10%. Under these conditions a R717 VCHP might be an advantage due to the simplicity of the system and the broader range of suppliers and contractors. This increased competition may cause the actual price of a R717 VCHP to be lower than that of the HACHP. Further it should be noted that lifetime cost of the HACHP has been minimized by the optimal choice of the ammonia mass fraction and circulation ratio. Thereby the best trade-off between running cost and investment has been determined to some extent. A further optimization of this could be imposed by finding the optimal size for all three heat exchanger. This could equally have been done for the VCHP where the potential for optimization might be higher as no consideration to this trade-off have been made.

It should be noted that the application of the high pressure compressor to the HACHP did not increase the range of feasible heat supply temperatures compared to those concluded from Brunin et al. (1997). This is mainly due to the applied constraint on the compressor discharge temperature, which was neglected in Brunin et al. (1997). The use of a cooled screw compressor or an oil free compressor could relax the constraints on the compressor discharge temperature and allow heat supply temperatures above 140 °C. Also, a two stage compression could be a measure for attaining these improvements.

The results presented here stem solely from simulation results by verified models but are not validated with experimental results. However, the same approach has been applied for all investigations and thus it is assumed that if any fault occurs it will mainly effect the absolute values. The comparison will therefore still be valid.

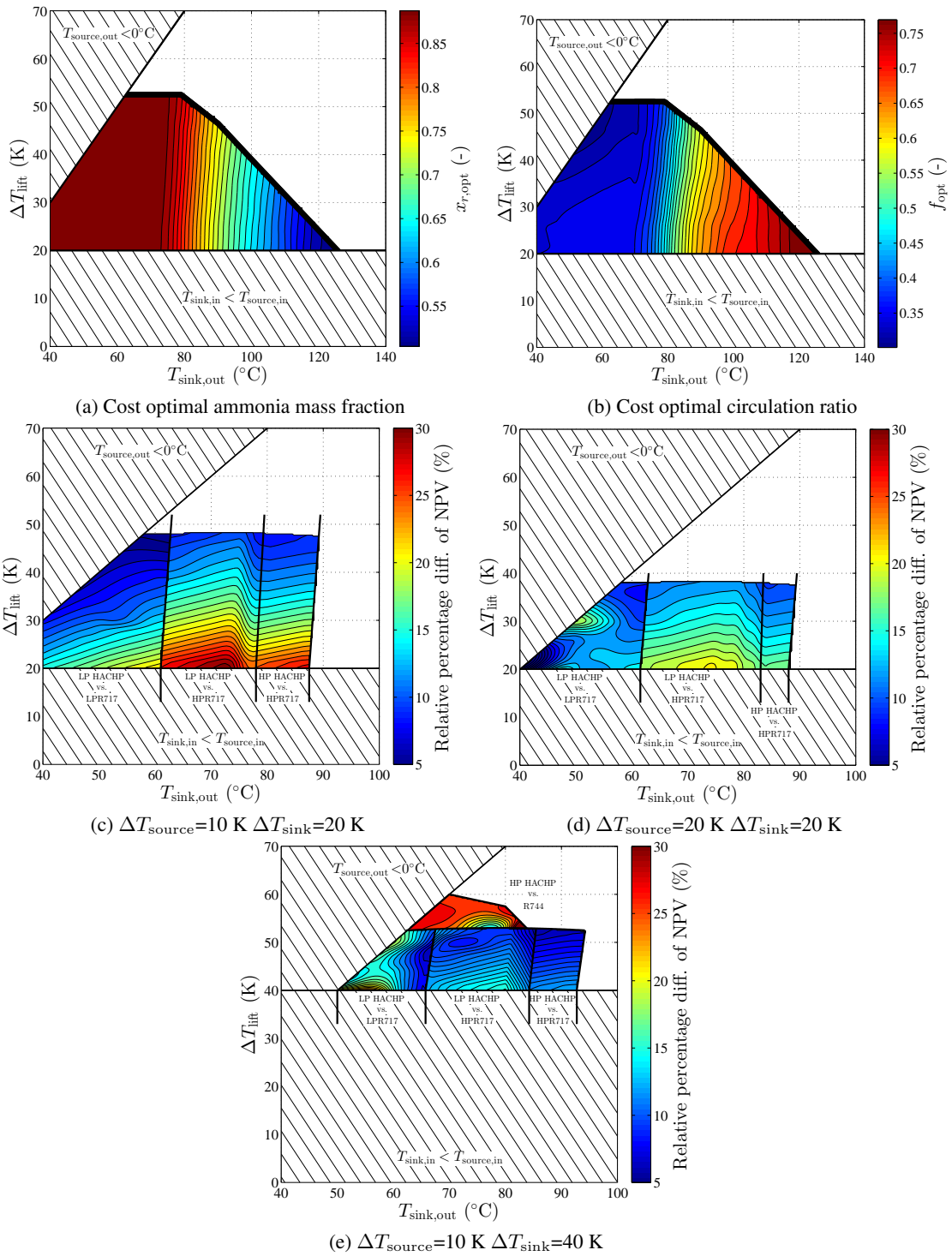


Figure 4

5. CONCLUSION

The feasible working domain of a HACHP has been evaluated based on a detailed economic analysis and a comprehensive investigation of the design variables: ammonia mass fraction and circulation ratio. The results show that the HACHP is capable of delivering both higher heat supply temperatures and higher temperature lifts than conventional VCHP. Heat supply temperatures up to 140 °C can be attained with commercially available components and with a significant economic benefit. When comparing the lifetime cost of the HACHP with the VCHP at the operating points where both are applicable: the cost of the HACHP is lower for all operating points. The relative difference to the closest competitor is between 5 and 30%, for R717 VCHP typically only 1-10%. Given this small difference, the HACHP may not be advantageous in the operating range, where R717 VCHP are feasible.

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NOMENCLATURE

Abbreviations

EES engineering equation solver
HACHP hybrid absorption-compression heat pump
VCHP vapour compression heat pump
PEC product equipment cost
NPV net present value

Symbols

p pressure (bar)
 T temperature ($^{\circ}\text{C}$) (difference (K))
 f circulation ratio (-)
 x ammonia mass fraction (-)

Greek symbols

Δ difference
 η efficiency
 ϵ effectiveness

Subscripts & Superscripts

HP high pressure
LP Low pressure
 r rich
 v vapour
is isentropic
vol volumetric
el electrical

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