Final report

Contents

1. Project details

Project title	English: Development of Ultra-High Temperature Hybrid Heat Pump for Process Applications Dansk: Udvikling af ultra-høj-temperatur hybrid varmepumpe for procesapplikationer
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Name of the funding scheme	EUDP-2011-II
Project managing company / institution	**Innoterm A/S** Essen 32, 6000 Kolding Contact: Palle Lemminger, pl@innoterm.dk
CVR number (central business register)	40372938
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2. Summary

2.1 Dansk resumé

Det overordnede formål med projektet var at forøge driftsområdet og begrænsninger for hybrid varmepumper i forhold til at erstatte fossile brændsler i proces industrien, hvor højere temperaturer er nødvendige. I en hybrid varmepumpe er der intern væske cirkulation samtidig med kompressionprocessen (baseret på Osenbrück kredsproccesen). Der indgår dermed to kølemidler som en blandning i kredsproccesen, og ved at blande de to kølemidler er det muligt at opnå høje temperaturer ved et moderat tryk.

Projektet har succesfuldt udvidet vidensområdet for hybrid varmepumpe teknologien, hvilket har bidraget til kommercialisering af denne teknologi. Det teoretiske arbejde bestod bl.a. af grundlæggende forskning og udvikling af hybrid varmepumpens proces, hvor exergianalyser blev sammenholdt med økonomiske konsekvenser.

Analyser af muligheder for integration af en hybridvarmepumpe ved slutbrugerne blev udført sammen med flere andre analyser af potentielle anvendelsesområder. Et 1-trins pilot demonstrationsanlæg blev designet, men selvom der er et stort potentiale for teknologien og den har fået meget opmærksomhed fra industrien, var det ikke muligt at bygge et fuldskala demonstrationsanlæg, da prioriteter for slutbrugerne ændrede sig i løbet af projektet. I stedet blev ressourcerne i arbejdspakken med demonstrationsanlæget fokuseret på at udføre integrationsstudier for at supportere arbejdet med at finde en vært til et demonstrationsanlæg. Projektet er blevet formidlet bredt, hvilket har øget opmærksomheden omkring hybrid varmepumper igennem udgivelse af flere artikler og ved flere præsentationer på konferencer.

2.2 English summary

The overall objective for the project was to enhance the operating range and limits for hybrid heat pumps in order to replace fossil fuels in thermal processes in the industry, which require higher temperatures. In a hybrid heat pump process the vapor compression cycle also has internal liquid circulation (based on the Osenbrück cycle). Hence, two refrigerants are flowing in the hybrid process as a mixture, and by mixing the two refrigerants it is possible to reach high temperatures at a moderate pressure.

The project successfully expanded the knowledge of the hybrid heat pump process and hence supported the work to commercialize the technology. The theoretical work included fundamental research and development with the hybrid heat pump process, where advanced exergy analyses were connected with economic analysis in order to investigate technical optimization possibilities and economic consequences.

Studies about the integration possibilities for a hybrid heat pump at the end-users were made, and additional other application case studies were completed. A one-stage pilot demonstration plant was also designed, however even though the technology has a high potential and has gained a lot of interest from the industry, it was not possible to build a demonstration plant, as the scope for the project partners changed during the project. Instead the resources in the work package with the demonstration plant were focused on carrying out feasibility studies, in order to support the work with finding a demonstration host.

The project has been disseminated widely and has increased the awareness of the hybrid heat pumps as several journal publications, peer-review conference papers, and presentations have been made.

3. Project objectives

The consumption of primary energy for heating in the industry can be decreased by utilizing heat pumps, as heat pump process can upgrade excess energy at a low temperature level (low quality energy) to energy at a high temperature (high quality energy) using only a fraction of the primary energy. The maximum temperature is however limiting the implementation of high temperature heat pumps in the industry. A maximum temperature of approximately 100 °C is still the limitation for these processes based on the traditional heat pump cycles and fluids.

The hybrid absorption-compression heat pump (HACHP) technology has the potential to reach relatively high supply temperatures with standard equipment as well as the flexibility to design highly efficient systems for given applications by varying the refrigerant mixture composition. The overall objective for the project was therefore to enhance the operating range and limits for hybrid heat pumps in order to replace fossil fuels in thermal processes in the industry, which require higher temperatures. Furthermore, the project aimed at developing methods for optimally designing the cycle for given applications to exploit the technology's potentials most optimally.

Hybrid Energy (HE), which is a partner in this project, has developed a heat pump system based on a “hybrid process”. The hybrid process uses a mixture of water and ammonia as natural working refrigerant, which makes it possible to reach temperatures of 110 °C with standard industrial refrigeration components. HE has demonstrated this technology on a number of plants, e.g. in Norway and Denmark. The aim of this project was to increase the operating temperatures of the hybrid process by using the new standard components that are approved to higher pressures. Higher operating temperatures will open new markets in the food and process industry for utilizing heat pumps to recover excess energy at a lower temperature level and bring the energy back into processes at a higher temperature level.

The technology behind the hybrid process is vapor compression cycle with internal liquid circulation (based on the Osenbrück cycle). The two refrigerants are flowing in the hybrid process as a mixture. By mixing the two refrigerants it is possible to reach a high temperature at a moderate pressure. Compared to a conventional vapor compression cycle the evaporator is replaced by a desorber, and the condenser is replaced by an absorber. In the desorber heat is transferred from the heat source to the refrigerant, at low temperature (waste energy). In a conventional vapor compression cycle the refrigerant experiences a phase change at constant temperature. For a zeotropic mixture, in this case a mixture of ammonia and water, the refrigerant experiences a temperature glide during phase change. A principle diagram for the hybrid process and its main components can be seen in Figure 1.

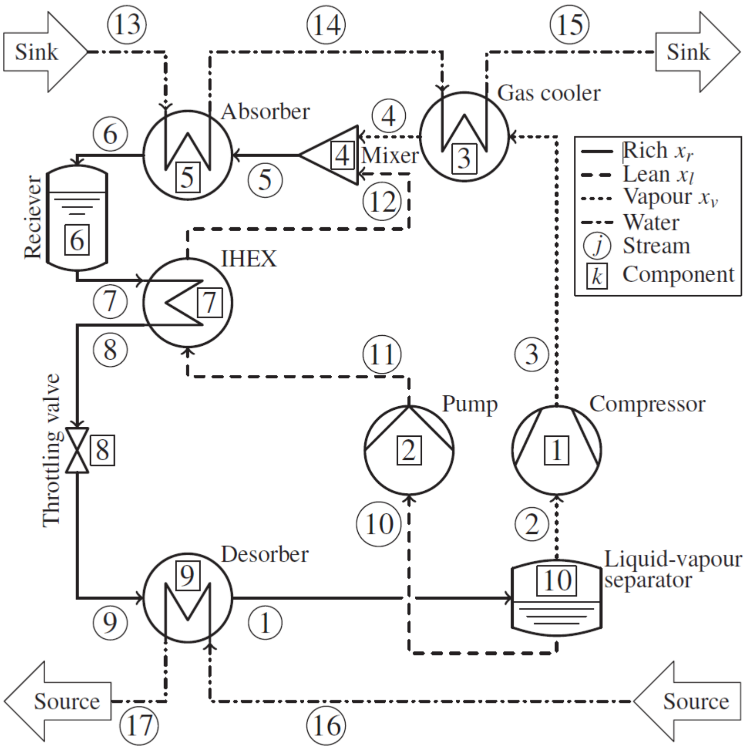

Figure 1 - Principle diagram for a one-stage hybrid heat pump [1].

As seen in Figure 1, the evaporated rich ammonia mixture is compressed in the compressor to a higher pressure and the water (still liquid) from the desorber is pumped to the same pressure and passed through and internal heat exchanger to increase the temperature before being mixed within or before the absorber. The heat is then rejected to the process stream while the refrigerant experiences a temperature glide like in the desorber. In most cases these temperature glides can be beneficial for the COP of the heat pump cycle, if the temperature glide matches the secondary streams.

The limitations with standard refrigeration components for the hybrid heat pump were earlier maximum 110 °C, but with new commercially available compressors it should be possible to increase the maximum temperature towards 180 °C, and hence to increase the market potential for this type of heat pump. The new components, which are approved to up to 120 bar is based on a development driven by the introduction of the refrigerant CO_2 which requires high pressures for transcritical operating conditions.

Due to the freedom of varying the composition of the mixture and thereby determining the temperature glides, the technology has a unique flexibility to be adjustable to specific boundary conditions, enabling highest efficiencies for a large range of applications. This project has therefore furthermore focused on the understanding of the system design for given applications in order to exploit these potentials and reach optimal designs.

The overall project has consisted of three parts:

A. Theoretical and practical investigation of the hybrid heat pump process for high temperatures.

B. Investigation of possible implementation into the processes at the end users in the consortium and the conduction of a general market survey.

C. Demonstration at end-users in the consortium.

4. Project implementation

The initial working structure for the project was divided into and planned with 7 work packages which had the following objectives.

- WP1 - Knowledge collection and literature study (DTU, NTNU, HE, DTI):

  A Ph.D. student and MSc students at DTU made the fundamental research and development supported by NTNU. The research tasks consisted of:
  - Thermodynamic process simulation and optimization
  - Cycle and working fluid study
  - Applications in future fossil-free energy system
  - Basic thermodynamics and relations to other refrigeration cycles as well as to power cycles

  Work package 1 created the basis for the following work packages because the calculations program was used to design the process and the components for the lab scale hybrid process.

- WP2 - Design and calculation of lab scale pilot plant (HE, Innoterm, DTI, DTU):

  The design specifications for the components (heat exchangers, compressor, piping and valves) was supported by using the calculation program developed in work package 1 in order to purchase them from sub suppliers.

- WP3 - Buildup and commissioning of lab scale pilot plant (Innnoterm, DTI, DTU, HE):

  The purpose of the lab scale pilot plant was to prove that an extended operating range can be made with a hybrid heat pump, and identify possibilities for cost savings including COP optimization.

- WP4 - Measurement, data acquisition and optimization (DTU, DTI, HE, IM):

An important issue is the documentation of the efficiency (COP) of the hybrid process at elevated pressures and temperatures, and hence to achieve high temperatures with maximal COP.

- WP5 - System integration of full scale ultra-high temperature heat pump (all partners):

  In this work package the objective was to analyze the process of the end users in order to find possibilities to integrate a hybrid high temperature heat pump (HTHP) to recover excess energy and identify possible energy savings. A market survey was also done to find the market potential for high temperature heat pumps at elevated temperatures. Hence, this work package supports decisions on whether an investment in a heat pump gives enough energy savings to make a heat pump an economically feasible solution for a specific case.

- WP6 - Demonstration plant (PE, Innoterm, HE, DTI, host for demo plant):

  The objective for this work package was to run a scaled up version of the lab scale pilot plant, which was based on experience from the pilot plant, and where the components and control of the demonstration plant matched the chosen end-user’s process.

- WP7 – Dissemination (all partners):

  The objective for this work was to make an extensive dissemination of the knowledge built up in this project. A number of articles was published in which the results obtained in the project were described. Furthermore, presentations at conferences and workshops were made.

The project objectives and scope has changed during the course of the project, especially regarding the end-user demonstration plant as described in the following sections.

The theoretical work about identifying optimization and applications for the hybrid heat pump has overall been successful. But even though the technology has a high potential and has gained a lot of interest from the industry, it was not possible to build a demonstration plant, as the scope for the project partners changed during the project. It was a deliberate decision to extend the project several times, in order to make a thorough investigation about the opportunities for a suitable demonstration site at an end-user which was economically viable. The changes that were made to the project plan can be summarized as:

- The operating goal for the demonstration hybrid heat pump made most sense to be changed to 130 °C and capacity of 1 MW, as it early in the project was clear, that the expenses of especially valves and heat exchangers would be disproportionately high at higher temperatures, and hence would limit the market potential for this type of technology. In addition, the requirement for excess heat temperatures (source temperatures) would be too high and limit the market potential if higher sink temperatures are to be made at an acceptable COP.

- The end-date for the project was changed to facilitate changes in the project partner group, and to investigate possibilities at a new end-user demonstration host.

- The changes to the partner consortium have been:
  - Shortly after the project was started up the factory at Solae Company Denmark was closed down, and hence the company left the project.
  - Bigadan entered the project in 2016 as a substitute for Solae Company Denmark. Bigadan later decided to leave the project.
  - In 2017 Arla continued in the project with a reduced observer role in order to be able to continue following the development of the technology, but with a reduced amount of internal resources allocated to the project.

- Pilot plant (WP3): Medio 2018 it was decided to not install the pilot plant and instead focus directly on a more flexible demonstration plant with supplementary measurement equipment. One option for the pilot plant was to modify an existing plant located in Norway and combine it with the already purchased components. The decision to not install this pilot plant was primarily taken due to much larger expenses that anticipated for retrofitting this equipment from Norway to a new installation site, as it among other things included changing the power supply and the control system, while new safety approvals also were needed for the plant.

- Demonstration plant (WP6): During the project, different integration studies of a hybrid heat pump at the different project partners and potential project partners have been conducted. Despite the completed integration studies and dialogues with possible demonstration hosts, it was not possible to identify an end-user who was willing to invest in the technology in relation to this project. Hence, after a period where the project also has been on hold, and investigations about a possible demonstration host has been made, it was decided to not install an end-user demonstration plant

Further details about the project implementation for each work package are described in the following sections.

4.1 WP1 - Knowledge collection and literature study

DTU performed fundamental research and development with hybrid processes for heat pumps, and the work was also supplemented by NTNU. In this work advanced exergy analyses were connected with economic analysis in order to investigate technical optimization possibilities and economic consequences. A review of state-of-the art has been made among others about two-phase heat transfer in Ammonia-water systems.

The milestones were overall met and calculation programs were made that can be used as a design tool by the partners for analysis of customer inquiries and for design of the pilot plant.

4.2 WP2 - Design and calculation of lab scale pilot plant

The purpose of the lab scale pilot plant was to prove that an extended operating range can be made with a hybrid heat pump and identify possibilities for cost savings including COP optimization.

The calculation programs from WP1 were successfully used to design the process and the components for the lab scale pilot plant. The design of the pilot plant was prepared including a P&I diagram and a component list.

Originally a two-stage 30 kW flexible system was designed, however this turned out to be more expensive than anticipated, primarily due to much higher valve prices than expected, hence a one-stage system was designed. Focus for the cost optimization possibilities was also to investigate if the number of heat exchangers could be reduced as both the absorber and desorber consisted typically of two heat exchangers each.

4.3 WP3 - Buildup and commissioning of lab scale pilot plant

Based on the preliminary design, two piston compressors and two oil separators were ordered early in the project. The other main components were however not ordered as cost optimizations were being investigated, among other things the solution pump and possibilities for the heat exchangers. The combination of both relatively high pressure and temperature were limiting the choice of valves and increasing the prices, and hence re-design from a 2-stage configuration to 1-stage was made.

During the project an opportunity came to base the lab scale plant on an existing plant in Norway. This plant has a 150 kW heating capacity and is based on a screw compressor. It is located at Institutt for Energiteknikk (IFE). A plan was made to redesign this plant and build it together with the already purchased components to gain further insight into operation with a screw compressor and oil separator in a hybrid heat pump cycle. The required modifications of the plant turned out to exceed the project budget and the focus shifted again to constructing a lab plant with the initial components.

After the decision was made to not install the pilot plant based on the plant at IFE, the earlier version of the pilot scale was again investigated with focus on choosing the optimal solution pump for the high pressure. When reconsidering constructing the lab-scale plant with the initial components and the findings from the other work packages, it turned out that the cost was higher than expected. Based on these circumstances, it was decided to not build a pilot plant and instead focus directly on a more flexible demonstration plant with supplementary measurement equipment.

4.4 WP4 - Measurement, data acquisition and optimization

The models used for exergy analysis and mapping of the losses in the hybrid process have been verified with available data with the first hybrid heat pump system in Denmark (EUDP project 64010-0026 ”Utilization of low grade industrial waste energy by means of new emerging high temperature heat pumps”).

Validation of the models made in the Master thesis [2] was made with operating from a hybrid heat pump installed at the production site at Nortura in Norway, which was designed by Hybrid Energy. The validation procedure performed on this simulation tool showed good overall concurrency with the measurement data obtained from the industrial plant.

4.5 WP5 - System integration of full scale ultra-high temperature heat pump

Studies about the integration possibilities for a hybrid heat pump at the end-users have been conducted, and additional case studies were completed. These include:

- Market surveys, including opportunities for installing hybrid heat pumps at industrial sites such as dairies, chemical process plants, and mink feeding centrals.
- Analysis of opportunities for implementing hybrid heat pumps at SPX-Anhydro spray-drying plants.
- Development of a method to determine optimal size and capacity of hybrid heat pumps were utilized to investigate a biogas plant at Bigadan (Horsens Bioenergi) and a biogas plant in Kalundborg (DONG and Bigadan)
- NTNU has also completed studies about the possibilities for using the HE Hybrid heat pump in relation to drying at temperatures above 150 °C, with direct heat transfer to the drying air.

4.6 WP6 - Demonstration plant

The possibilities for demonstration plants at the end-users were investigated in order to determine the economic feasibility with the capacity and temperature for the hybrid heat pump in scope of the project. It was however not possible to find a feasible solution to build a demonstration plant, and the scope for the project partners also changed during the project. After Bigadan replaced Solae Denmark in the project, a thorough investigation was also made regarding whether a hybrid heat pump could be integrated in their energy system when upgrading biogas to natural gas quality.

In 2018, Aalborg Forsyning and Industribejdsning Nord became interested in becoming a demonstration host. For this case, it was analyzed to upgrade district heating with a hybrid heat pump for supply of process heat to Industribejdsning Nord. Industribejdsning Nord had a tight time schedule for the establishment of the equipment in order to replace the existing equipment which was close to breakdown. However, even though the overall business case looked promising, the parties could not agree on contractual agreements and hence it was necessary for Industribejdsning Nord to invest in more traditional production equipment.

Several other potential demonstration hosts were also contacted and preliminary feasibility studies were conducted. However, as the investment in the hybrid plant would be close to commercial market terms, and comprehensive investigations would be necessary to carry out at a new demonstration host, it was not possible to find a host willing to participate under the defined terms. Hence, the resources in this work package has primarily been allocated to feasibility studies, in order to support the work with finding a demonstration host.

4.7 WP7 – Dissemination

The project has been disseminated as planed through journal publications and conference presentations.

5. Project results

The project has resulted in a wide range of novel studies on the subject of hybrid heat pumps, which are summarized in this chapter. For full description of the studies, please refer to the dissemination list in chapter 8.

5.1 State-of-the-art

When designing and analyzing the HACHP it is important to account for the two extra degrees of freedom of the HACHP, which can be attributed to the composition of the zeotropic mixture and the solution circuit design. The working fluid composition and the rate of solution circulation are thus values that can be chosen freely by the system designer. The choice of these values have been shown in literature to have a great influence on the size and performance of the system. Different approaches to satisfying and optimizing these extra degrees of freedom are suggested by several authors.

Stokar and Trepp [3] and Stokar [4] investigated the ammonia-water HACHP both experimentally and numerically. The hypothesis of this study was that, apart from the increased coefficient of performance (COP), the HACHP can ensure better capacity control than a vapor compression heat pump (VCHP). The improved capacity control is attributed to the adjustment of the ammonia mass fraction. Thus, by decreasing the ammonia mass fraction the vapour pressure of the working fluid is reduced and consequently a fixed speed compressor will provide a lower mass flow rate subsequently reducing the heat load. The analysis was performed at a fixed operating condition in which the sink was heated from 40 °C to 70 °C while cooling the source from 40 °C to 15 °C. It was concluded that for each capacity (ammonia mass fraction) there exists one circulation rate that will optimize the COP. It was further seen that the optimum COP is attained at the point where the temperature glide of the working fluid and the heat sink is close to identical as indicated by a glide match ratio (ratio of working fluid glide to the heat sink glide) close to 1, however optimum ratios between 0.8 and 1.2 are observed.

Åhlby et al. [5] conducted a numerical optimization study of the ammonia-water HACHP. The optimization was performed at four operating conditions all with a heat supply temperature of 80 °C but with varying values of ΔT_{sink} and ΔT_{source}. Further, the HACHP was compared to a R12 VCHP. The objective of the optimization was to determine the optimum internal temperature difference in the absorber (temperature difference between inlet and outlet of the absorber) for a given value of maximum allowable pressure. This showed that for each maximum pressure there exists one value of internal temperature difference that will optimize the COP. The optimum internal temperature difference depends strongly on the external conditions. Åhlby et al. [5] stated that the optimum internal temperature difference can differ significantly from the external temperature difference, especially if the external temperature difference is low. For low external temperature differences, it was shown that the internal temperature difference should be considerably higher than the external to optimize the COP. Further, Åhlby et al. [5] concluded that the highest COP was attained at the highest pressure. For all the investigated operating conditions the HACHP yielded a higher COP than the R12 VCHP.

Åhlby et al. continued their work in [6] by investigating the use of the ternary fluid NH_3-H_2O-LiBr. This showed to improve the COP up to 10 %, with the highest increase for the cases with large sink-source temperature differences.

Itard and Machielsen [7] discussed the issues that arise when modelling and comparing HACHP. It wa shown that it is important to account for the non-linear relation of temperature and enthalpy during equilibrium absorption and desorption. Not accounting for the non-linearity may lead to infeasible temperature profiles as cross overs may be encountered. The external conditions must therefore be included in the evaluation of the pressure needed to ensure feasible profiles. Itard and Machielsen [7] claimed that these considerations were not applied in previous studies such as Åhlby et al. [5], [6] and thus it is unclear whether the conclusions from these studies rely on infeasible temperature profiles. Itard and Machielsen [7] stated that temperature – enthalpy diagrams are useful to visualize the heat transfer process and ensure feasible profiles. Another consequence of the non-linearity, stated by Itard and Machielsen [7], is the inapplicability of the logarithmic mean temperature difference (LMTD). LMTD is based on an assumption of constant capacity rates. Thus, applying LMTD to absorption or desorption processes may lead to under or over estimation of the mean temperature difference. Further, Itard and Machielsen [7] compared the solution circuit design with the wet compression cycle and found that no clear conclusion can be drawn as to which cycle yields the best performance. Which cycle performs best depends on both the chosen mixture, concentration, temperature level and operating condition.

Brunin et al. [8] compared the working domain of the HACHP to several VCHP. The working domains were evaluated based on two technical constraints: a maximum high pressure and minimum low pressure as well as two economic indicators: minimum COP and minimum volumetric heat capacity (VHC). The HACHP was evaluated at three ammonia mass fractions 0.25, 0.35 and 0.45 and with a fixed concentration difference between the rich and lean streams of 0.10. The working domains were derived for a fixed sink-source temperature difference of 10 K. Brunin et al. [8] concluded that the only solution for high temperature heat pumps is either the HACHP or a hydrocarbon VCHP. The study by Brunin et al. [8] was carried out without the considerations of non-linearity discussed by Itard and Machielsen [7], further it is unclear how the constant concentration difference of 0.10 relates to the optimum circulation rates discussed by Stokar [4] or optimum temperature difference shown in Åhlby et al. [5].

In the research of the present project the HACHP was modelled such that the rich ammonia mass fraction and the circulation ratio are inputs to the model and thus these parameters are applied to satisfy the two extra degrees of freedom. By choosing the ammonia mass fraction as one input all possible solutions can be accounted for ranging from a pure water cycle $x_r = 0.0$ to a pure ammonia cycle $x_r = 1.0$. By choosing the circulation ratio as the second input all solutions ranging from the VCHP cycle $f = 0$ to a cycle with no phase change, f = 1 (not a physical solution) can be modelled. Thus, simultaneously varying x_r and f from 0 to 1 result in the evaluation of all possible solutions for a given operating condition.

5.2 Modelling and process optimization of HACHP cycles

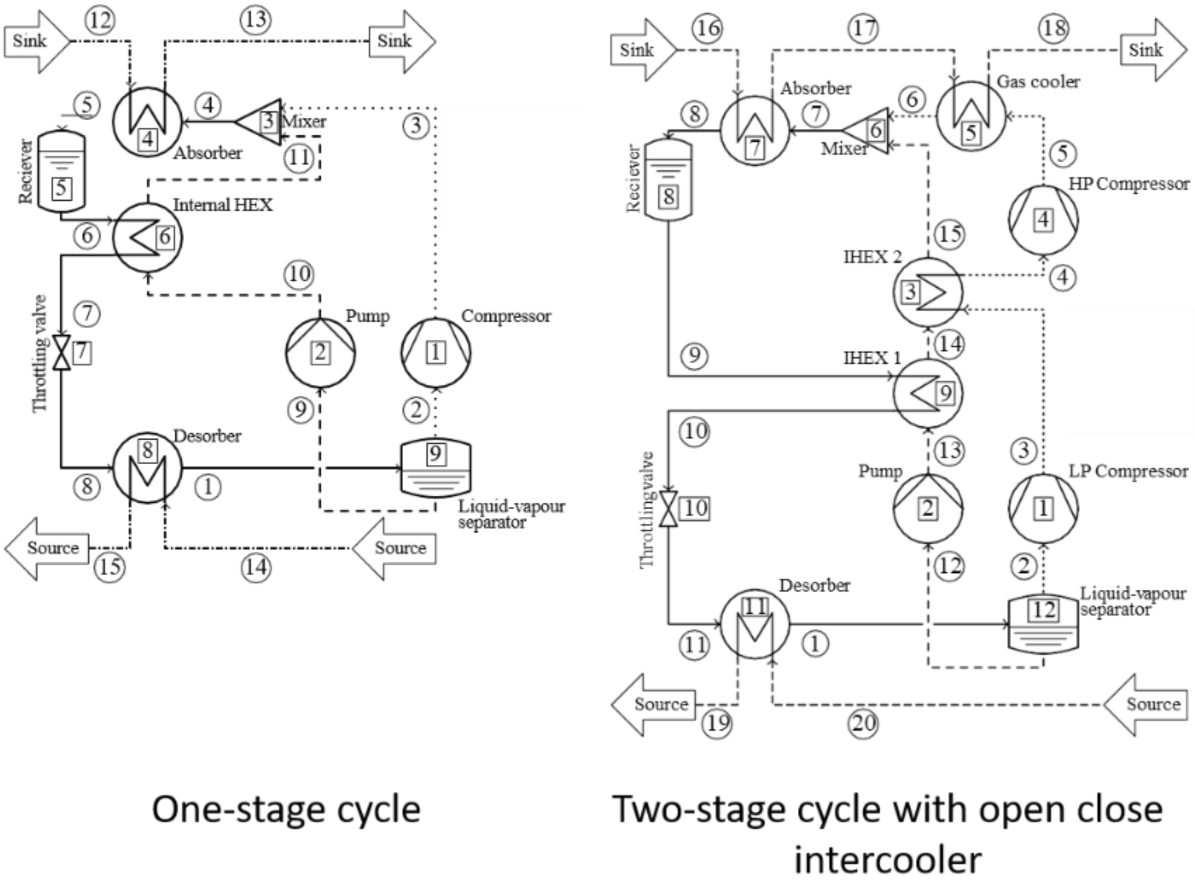

Figure 2 - Modelled one-stage HACHP and two-stage HACHP with closed intercooler [1].

A numerical model of a one-stage HACHP was developed in EES [9]. The thermodynamic properties of the ammonia-water mixture in the EES model were calculated using the Ibrahim and Klein equation of state [10]. A complete result for an exemplary simulation in the developed EES model may be seen in Figure 3 and in Table 1.

Further, numerical models of one one-stage and several two-stage HACHPs were developed in MATLAB 2015a [11]. The thermodynamic properties of the ammonia-water mixture in the MATLAB models were calculated using the Refprop [12] interface for MATLAB. The so-called 'Ammonia (Lemmon)' formulation was applied. As discussed by Modi and Haglind [13] this formulation significantly increases the robustness of the property calculations compared to the default Tillner Roth and Friend [14] formulation, especially in the two-phase region. The robustness is achieved without significantly compromising the accuracy of the property calculations [13].

In total five cycle configurations were modeled, one one-stage and four two-stage cycles. The two-stage cycles consisted of two cycles with closed intercoolers, one open intercooler cycle and one liquid injection cycle. The modelled one-stage cycle may be seen in Figure 2. The simulation showed that one of the two-stage cycles was always preferable compared to the other identified solution. This was the two-stage HACHP with closed intercooler also seen in Figure 2.

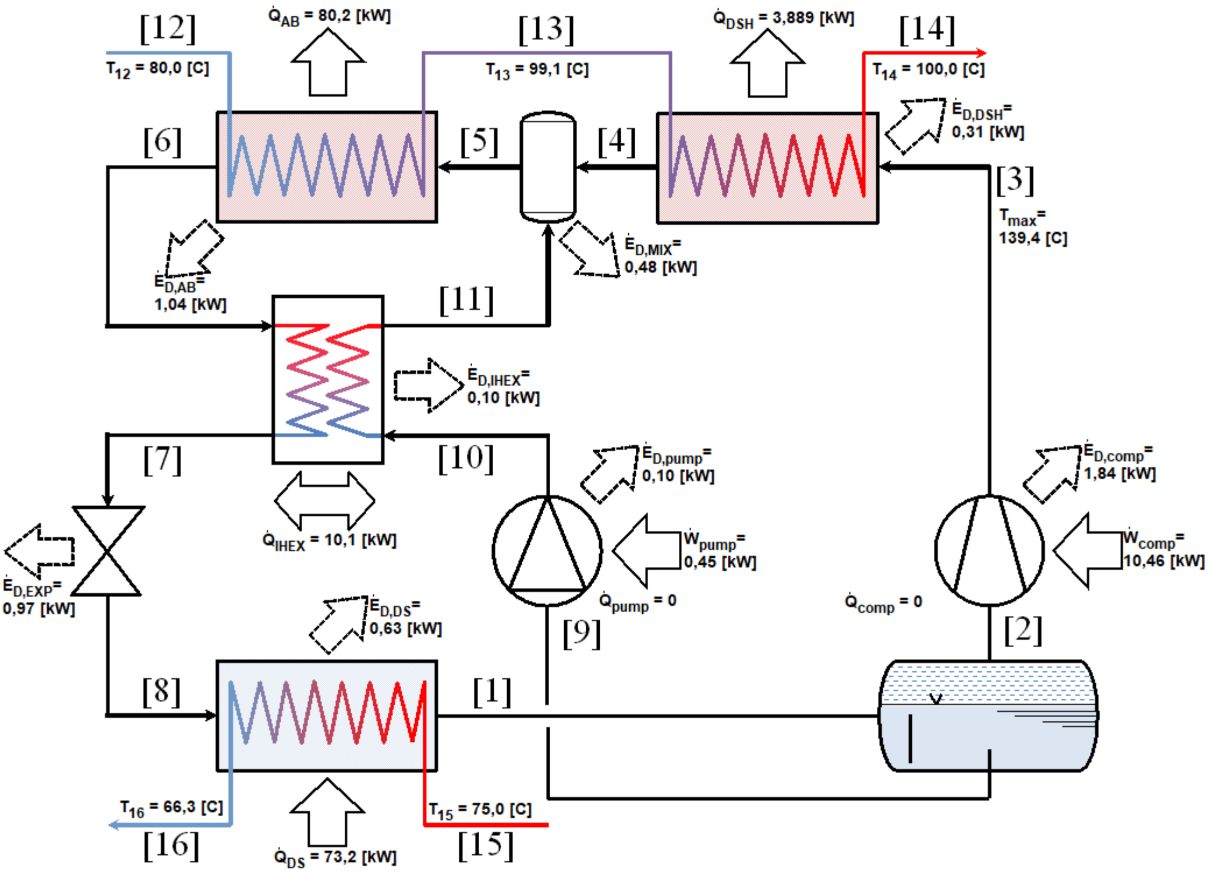

Figure 3 - Temperatures, energy flows and exergy destruction rates for a simulation of a HACHP [1].

Table 1 - Cycle state points for the HACHP simulation seen in Figure 3.

	$\dot{m}$ [kg/s]	*T* [C°]	*p* [bar]	*x* [-]	*h* [kj/kg]	*s* [kj/kg-K]	*q* [kg/kg]	*v* [m³/kg]
[1]	**0,2140**	**72,0**	**18,66**	**0,75**	**555**	**2,115**	**0,350**	**2,86E-02**
[2]	**0,0749**	**72,0**	**18,66**	**1,00**	**1381**	**4,360**	**1,000**	**7,91E-02**
[3]	**0,0749**	**139,4**	**36,35**	**1,00**	**1521**	**4,443**	**1,001**	**4,87E-02**
[4]	**0,0749**	**123,2**	**36,35**	**1,00**	**1469**	**4,318**	**1,001**	**4,56E-02**
[5]	**0,2140**	**102,1**	**36,35**	**0,75**	**635**	**2,243**	**0,325**	**1,44E-02**
[6]	**0,2140**	**89,0**	**36,35**	**0,75**	**260**	**1,226**	**0,000**	**1,59E-03**
[7]	**0,2140**	**79,9**	**36,35**	**0,75**	**213**	**1,095**	**-0,001**	**1,55E-03**
[8]	**0,2140**	**60,6**	**18,66**	**0,75**	**213**	**1,110**	**0,089**	**7,98E-03**
[9]	**0,1391**	**72,0**	**18,66**	**0,62**	**110**	**0,906**	**0,000**	**1,38E-03**
[10]	**0,1391**	**72,5**	**36,35**	**0,62**	**113**	**0,909**	**-0,001**	**1,38E-03**
[11]	**0,1391**	**87,4**	**36,35**	**0,62**	**185**	**1,113**	**-0,001**	**1,42E-03**

	$\dot{m}$ [kg/s]	*T* [C°]	*p* [bar]	*h* [kj/kg]	*s* [kj/kg-K]
[12] Sink inlet	**1,0000**	**80,0**	**2,00**	**335**	**1,075**
[13] Sink intermed.	**1,0000**	**99,1**	**2,00**	**415**	**1,297**
[14] Sink outlet	**1,0000**	**100,0**	**2,00**	**419**	**1,307**
[15] Source inlet	**2,0000**	**75,0**	**10,00**	**315**	**1,015**
[16] Source outlet	**2,0000**	**66,3**	**10,00**	**278**	**0,909**

In both the EES and MATLAB models, each component was modelled based on steady state mass and energy balances. Further, the model ensured that the Second Law of thermodynamics was fulfilled in all components. For the absorption and desorption processes this required a discretization of the heat transfer process to account for the non-linearity of the absorption and desorption temperature profiles. Heat and pressure losses in heat exchangers, vessels and pipping were neglected. 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 solution. Hence, the circulation ratio was directly linked to the vapour quality exiting the desorber, such that: $q = 1 - f$.

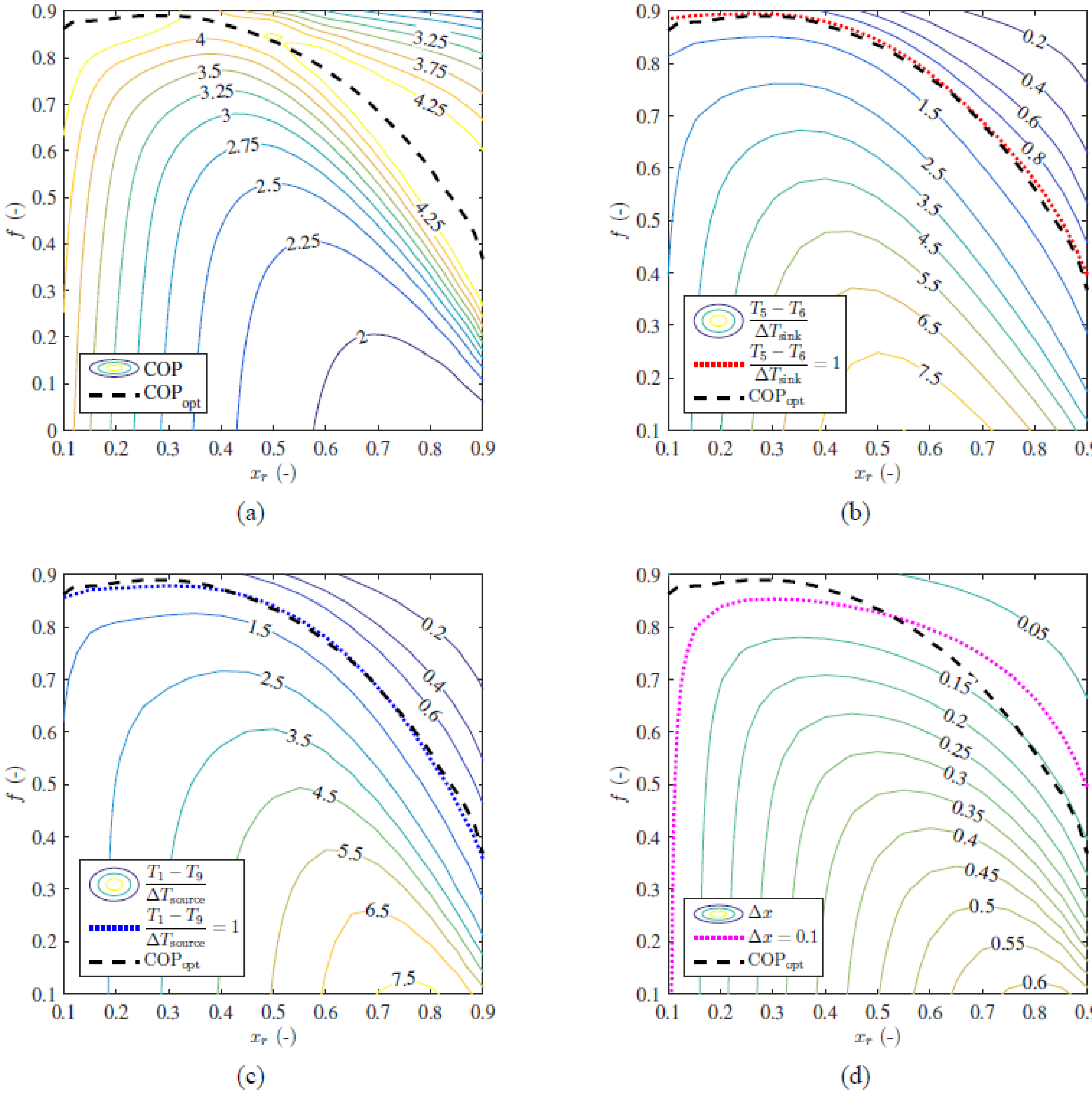

Figure 4 - The variation of COP (a), sink temperature glide match (b), source temperature glide match (c) and concentration difference (d) for an operating condition of $DT_{sink} = DT_{source}$ = 10 K, DT_{lift} = 30 K and a heat supply temperature of $T_{sink,out}$ = 100 °C [1].

Figure 4 presents a parameter variation of the ammonia mass fraction and circulation ratio for a one-stage HACHP, both parameters from varied 0.1 to 0.9. An operating condition of $\Delta T_{sink} = \Delta T_{source}$ = 10 K, ΔT_{lift} = 30 K and $T_{sink,out}$ = 100 ˚C was applied. Figure 4 (a) presents the contours of the COP as a function of the ammonia mass fraction and circulation ratio. Figure 4(b) shows the corresponding sink glide while the source glide match ratio is presented in Figure 4(c). The glide match ratios are defined as the ratios of the average internal temperature gradient to the average external temperature gradient. Thus, when these ratios attain a value so one the temperature profiles of the absorption or desorption processes matches the one of the sink or source. The non-linearity is not accounted for in these ratios and thus the overall minimization of the temperature difference may differ.

The contours of concentration difference, Δx, is seen Figure 4(c), Δx is defined as the difference between the lean and rich ammonia mass fractions.

As seen from Figure 4(a), the choice of ammonia mass fraction and circulation ratio has a large influence of the COP. For each choice of ammonia mass fraction one value of circulation ratio maximizes the COP. This is represented by the black dashed line. As seen the lower the ammonia mass fraction the higher the circulation ratio should be applied to optimize the COP.

Comparing the contours of the COP to the contours of the sink and source glide match ratios, it is clear that the maximum COP coincides with the matching of temperature profiles. If the circulation ratio is chosen below the optimum value, the glide match ratios are too high and vice versa. Hence, for a circulation ratio below optimum: the temperature difference over the absorption/desorption process is too large compared to the sink/source temperature difference. Choosing a circulation ratio above the optimum causes a temperature difference in the absorption/desorption process that is too low compared to the sink/source temperature difference.

Comparing the contours of the COP to those of the concentration difference, Figure 4(d), it can be seen that some discrepancy exists between the optimum COP and the constant concentration difference. However, it should be noted that the choice of $\Delta x = 0.1$ does keep the COP close to its optimal value.

It should be noted that although both the sink and source glide match ratios in the present case as well as the concentration difference can be used to identify close to optimal COPs then this may not be the case for all operating conditions. It was found that over a large range of operating conditions, attaining a glide match ratio of unity in the desorber typically attained a COP closest to the optimum.

5.3 Feasibility of high temperature development

Brunin et al. [8] showed that it is technically and economically feasible to use the HACHP up to a heat supply temperature of 140 ˚C. This, however, is based on a high pressure constraint of 20 bar corresponding to the limitations of standard refrigeration components at the time of the study. In the meantime new compressor types ofr high pressure NH_3 (50 bar) and transcritical CO_2 (140 bar) applications have become commercially available and further standard refrigeration components now operate at 28 bar. It is therefore of interest to evaluate how the application of these components changes the working domain of the HACHP.

One design constraint that is not discussed by Brunin et al. [8] is the compressor discharge temperature. However, most compressor manufacturers require this to be lower than 180 °C [15]. This is mainly due to the thermal stability of the lubricating oil and the thermal stress of the materials surrounding the compressor discharge line. The material constraints are mainly an issue for reciprocating compressors.

Changing the lubricant from a mineral oil to a synthetic oil could relax the constraint due to thermal stability. This however requires that a synthetic oil that meets the requirements of miscibility etc. is identified. Adjustments to the gasket materials and alike could also make the compressor durable at higher discharge temperatures. It is assumed to be a realistic estimate that compressor discharge temperatures up to 250 °C can be sustained with minor adjustments.

To evaluate the working domain of the HACHP using the recently developed high pressure equipment a set of design constraints is defined. A solution that satisfies this set of constraints will constitute an economically and technically feasible solution. The technical limitations are: the high pressure, governed by the choice of compressor technology, the low pressure, set to eliminate entrainment of air, and the compressor discharge temperature, as discussed above. Further, for the ammonia compressors a constraint is set on the vapour ammonia mass fraction. The economic constraints are: the Coefficient of Performance (COP) and the volumetric heat capacity (VHC), calculated as the ratio of the compressor displacement volume to the heat output of the HACHP [8].

As shown the rich ammonia mass fraction, x_r, and the circulation ratio, f, influence the design values of the constraining parameters significantly. The combination of these two govern the system pressure and thereby the VHC. Also, the slope of the absorption-desorption curve and thereby the performance (COP) is influenced by these parameters.

Here the working domains were analyzed by investigating the set of feasible combination of x_r and f at heat supply temperatures of 100 °C, 125 °C, 150 °C and 175 °C. Working domains will be evaluated for all three mentioned types of refrigeration components. The three sets of components have the constraints listed in Table 1.

Table 1 - Design constraints for standard refrigeration, high pressure NH_3 and transcritical CO_2 components.

	Unit	Standard ref.	HP NH_3	Transcrit. CO_2
$p_{H,\mathrm{max}}$	bar	28	50	140
$p_{L,\mathrm{min}}$	bar	1	1	1
$T_{H,\mathrm{max}}$	°C	170	170	250
$\mathrm{COP}_{\mathrm{min}}$	-	4	4	4
$\mathrm{VHC}_{\mathrm{min}}$	$\mathrm{MJ\ m^{-3}}$	2	2	2
$x_{v,\mathrm{min}}$	-	0.95	0.95	0.00

The constraint on COP and VHC ensures the economic feasibility of the heat pump as a high COP ensures a low running cost and a high VHC indicates a low investment cost. The applied values for the COP, VHC and low pressure constraints are set in agreement with those presented by Brunin et al. [8]. The high pressure constraints for the different compressor technologies are summarized by Ommen et al. [16].

For the two ammonia compressors a constraint was imposed on the vapour ammonia mass fraction, x_v. It was assumed that 5 % water is acceptable in an ammonia compressor. For the modified transcritical CO_2 compressor no constraint was imposed on x_v as they may just as well be modified to handle the needed ammonia-water composition.

Two HACHP cycles were investigated, the one-stage HACHP and a two-stage HACHP, the layout of both cycles is seen in Figure 2. Both the one-stage and two-stage cycles were analyzed at a fixed operating condition governed by the sink temperature difference, $\Delta T_{\mathrm{sink}} = 20$ K, temperature lift, $\Delta T_{\mathrm{lift}} = 25$ K and the mass flow rates of the sink and source of 1 $\mathrm{kg \cdot s^{-1}}$ and 2 $\mathrm{kg \cdot s^{-1}}$, respectively.

Figure 5 shows the contours of COP, VHC, compressor discharge temperature, high pressure, low pressure and vapour ammonia mass fraction for the one-stage HACHP.

The general trend of all constrained variables with x_r and f are similar for both the one and two-stage HACHP. Although it can be noted that the COP is generally highest for the two-stage while the one-stage generally attains the highest VHC. Further, the compressor discharge temperature is lower for the two-stage HACHP compared to the one-stage.

All the constrained design parameters are greatly influenced by both ammonia mass fraction and circulation ratio and thus the behavior of these cannot be attributed solely to one of the two. Thus, the main conclusion derived from Figure 5 is that the correct combination of ammonia mass fraction and circulation ratio is needed in order to identify a feasible design.

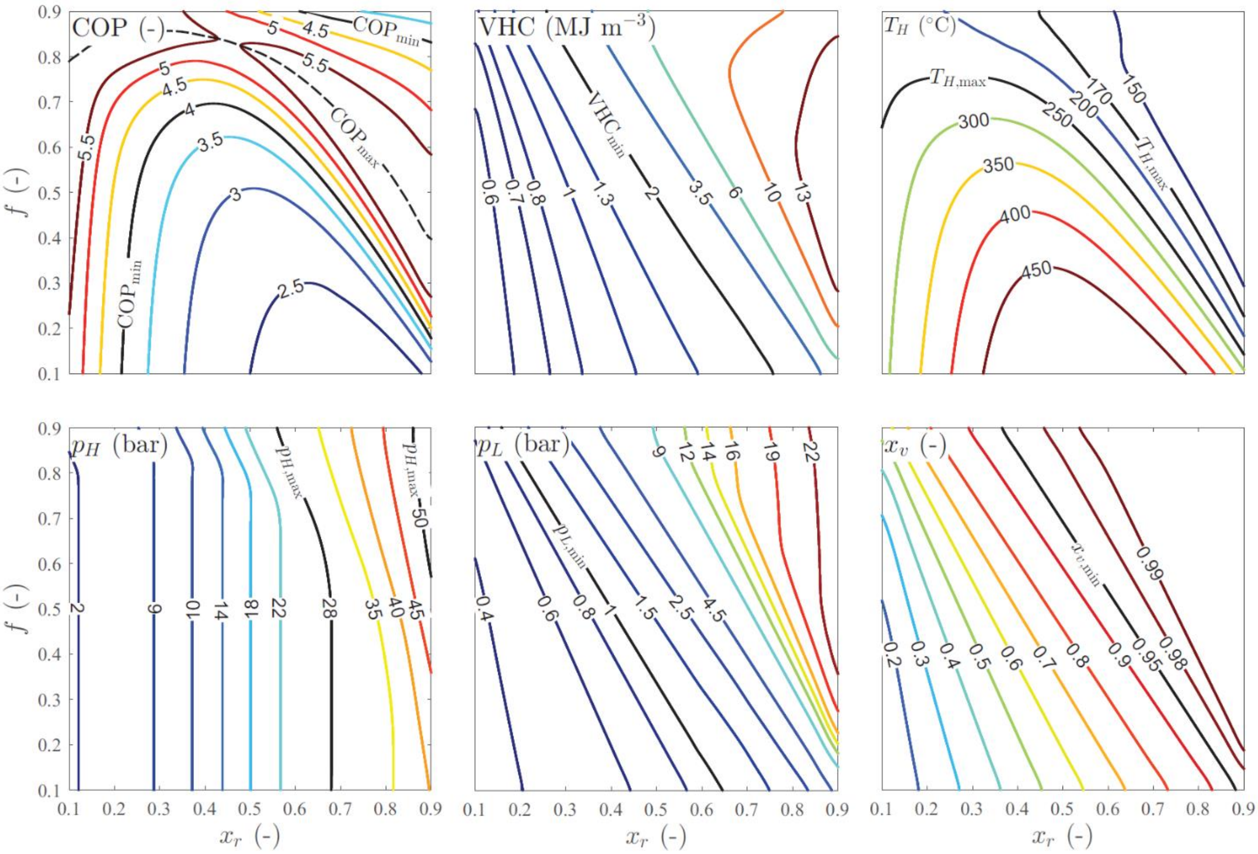

Figure 5 - COP, VHC, compressor discharge temperature, T_H, high pressure, p_H, low pressure, p_L, and vapour ammonia mass fraction, x_v, as a function of the circulation ratio, *f*, and the ammonia mass fraction, x_r for the one-stage HACHP. Heat supply temperature $T_{sink,out}$ = 100 °C [1].

Thus, In order to evaluate the applicability of the HACHP for high temperature operation ($T_{sink,out}$ > 100 °C) the feasible combinations of ammonia mass fraction and circulation ratios must be identified in this temperature range. The feasible design combinations are given by the design constraints listed in Table 1. As these are specific to the three technologies (standard, high pressure NH_3 and transcritical CO_2) three sets of feasible combinations are identified. These sets of feasible combinations have been determined at four heat supply temperatures: $T_{sink,out}$ = 100 °C, 125 °C ,150 °C and 175 °C.

Figure 6 shows the feasible combinations of the three compressor technologies at the four heat supply temperatures for the one-stage HACHP. For the one-stage HACHP with $T_{sink,out}$ = 100 °C a feasible set of combinations exists for all three compressor technologies. The set belonging to the standard refrigeration components is the smallest of the three and is constrained by the high pressure, p_H, and compressor discharge temperature, T_H. The set belonging to the high pressure NH_3 technology is significantly larger than that of the standard components. This set is constrained by p_H, T_H and COP. The largest set is that of the transcritical CO2 components. This set is constrained by the VHC, T_H and COP. It can be noted that the x_v constraint is dominant over both the low pressure and VHC constraint in the entire range, thus these constraints are effectively extraneous.

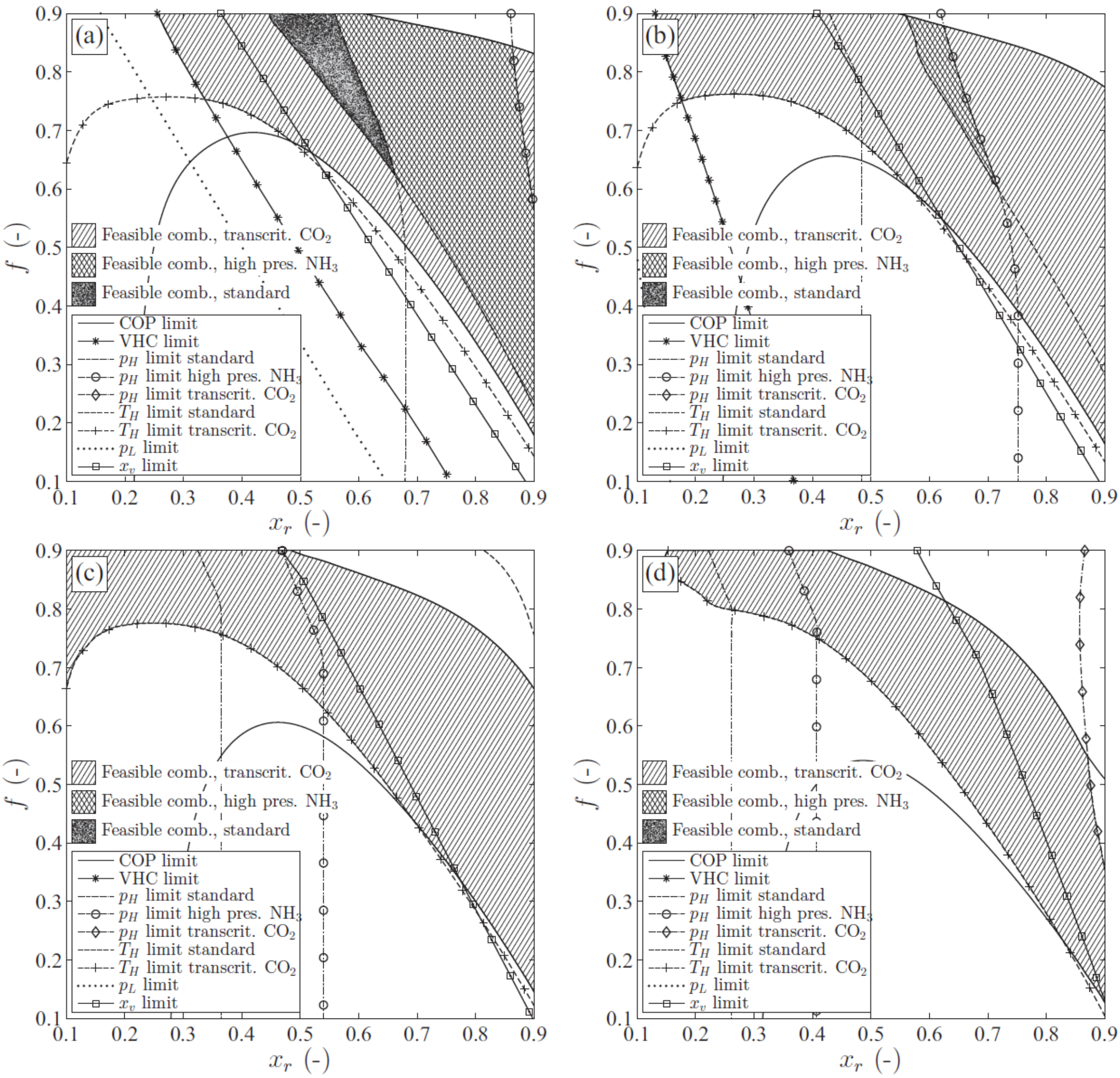

Figure 6 - Feasible combinations of the circulation ratio, *f*, and the ammonia mass fraction, *xr*, for standard, high pressure NH_3 and transcritical CO_2 components. Heat supply temperature (a): $T_{sink,out}$ = 100 °C, (b): $T_{sink,out}$ = 125 °C, (c): $T_{sink,out}$ = 150 °C, (d): $T_{sink,out}$ = 175 °C [1].

Figure 6(b) shows the results with a heat supply temperature of 125 °C. As seen, this has the consequence, that the high pressure constraint for the standard components moves beyond the compressor discharge temperature constraint, deeming the application of the standard refrigeration components infeasible. A small set of feasible combinations exists for the high pressure NH_3 components. This set is constrained by p_H and T_H. The set of feasible combinations for the transcritical CO_2 components is slightly increased as the VHC constraint is moved towards lower ammonia mass fractions. It should be noted that although the VHC and low pressure constraints have moved significantly towards lower x_r the vapour ammonia mass fraction constraint actually moved slightly towards higher x_r allowing fewer solutions to satisfy this constraint.

Figure 6(c) has a heat supply temperature of 150 °C. Here no combinations that satisfy the COP constraint also satisfy the constraint of T_H < 170 °C. Thus, neither standard refrigeration nor high pressure NH_3 components are applicable at a heat supply temperature of 150 °C. The VHC constraint is no longer present due to the high temperature of the heat source and subsequently high desorber pressure. This again causes the set of feasible combinations for transcritical CO_2 components to expand. This set is only constrained by T_H and COP.

Figure 6(d) has a heat supply temperature of 175 °C. Again, only the transcritical CO_2 components can be applied due to the high compressor discharge temperature. For the first time the high pressure constraint for transcritical CO_2, $p_H < 140$ bar, is present, while the VHC constraint is no longer present. The set of the feasible combinations is reduced compared to a heat supply temperature of 150 °C due to the increased pressure. The set of feasible combinations is constrained by T_H, COP and p_H.

It should be noted that the increased heat supply temperature attained by the use of the transcritical CO_2 components is only possible because of the increased tolerance on the compressor discharge temperature. As seen in Figure 6(c) & (d), a design that keeps the compressor discharge temperature below 170 °C is not possible. Hence, judging from Figure 6, if the transcritical CO_2 compressor cannot be modified to sustain these temperatures, these components cannot attain higher supply temperatures than the high pressure NH_3 components. Conversely, it can be seen that if the standard refrigeration and high pressure NH_3 compressors are also modified to sustain 250 °C compressor discharge temperature and vapour ammonia mass fractions below 0.95, these can also attain heat supply temperatures up to 175 °C, although the set of feasible combinations of x_r and f is considerably smaller.

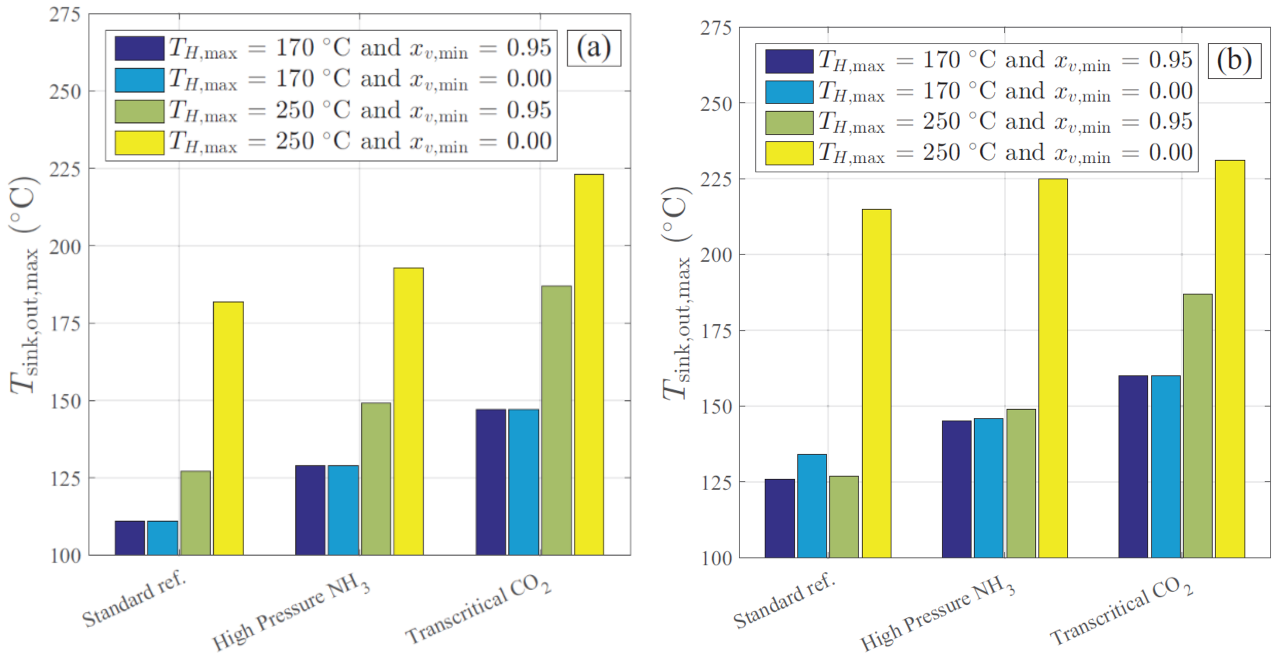

Figure 7 - Maximum attainable heat supply temperature for the three investigated component types. Two maximum compressor discharge temperatures are imposed. Further, $T_{\text{sink,out,max}}$ is presented with and without the vapour ammonia mass fraction constraint [1].

To get a more detailed view of the temperature levels that can be achieved by the different components, the maximum heat supply temperature was derived. The maximum heat supply temperature was defined as the temperature level at which no combinations of x_r and f result in a feasible solution.

Figure 7 shows the maximum heat supply temperature for the one-stage HACHP (a) and two-stage HACHP (b). The maximum temperature is presented for all three component types with a compressor discharge temperature constraint of both $T_{H,\max} = 170$ °C and $T_{H,\max} = 250$ °C and with and without the vapour ammonia mass fraction constraint.

For the one-stage HACHP, with $T_{H,\max}$ = 170 °C and $x_{v,\min}$ = 0.95 a maximum heat supply temperature of 111 °C can be attained with 28 bar components, 129 °C with 50 bar components and 147 °C with 140 bar components. Using the two-stage HACHP this is increased to 126 °C 145 °C and 160 °C

Removing only the vapour ammonia mass fraction constraint does not lead to a further increase in attainable temperatures for the one-stage HACHP. For the two-stage HACHP the maximum temperature is increased by 8 °C for the standard components and 1 °C for the high pressure NH3 components. No increase is observed for the transcritical CO_2 components.

Increasing $T_{H,\max}$ to 250 °C for the one-stage HACHP increases the maximum temperature by 16 °C for standard components, 20 °C for high pressure NH_3 and 40 °C for transcritical CO_2. For the two-stage HACHP, increasing $T_{H,\max}$ alone does not increase the maximum temperature for the standard refrigeration components. For the high pressure NH_3 components a slight increase of 4 °C is attained while the transcritical CO_2 components yield an increase of 27 °C.

As seen the largest increase in maximum heat supply temperature is attained when increasing $T_{H,\max}$ to 250 °C and simultaneously removing the vapour ammonia mass fraction constraint. This allows the one-stage HACHP to operate at heat supply temperature up to 182 °C, 193 °C and 223 °C for the respective pressure constraints. While, the two-stage HACHP allows 215 °C, 225 °C and 231 °C, respectively.

5.4 Technical and economic working Domains of HACHP and comparison with conventional vapour compression heat pumps

As discussed by Ommen et al [17], the use of COP and VHC as indicators for the economic viability of a heat pump investment may not always lead to a precise conclusion. This is because components for different refrigerants may vary in cost due to e.g. different material requirements or different heat transfer coefficients and thus, the level of COP needed to attain a viable investment will differ based on the level of investment required for the specific refrigerant. Evaluating the working domain based on a complete economic evaluation of the heat pump installation over the lifetime of the system may result in valuable information not provided by Brunin et al. [8].

The application of a complete economic analysis allows the different heat pump solutions to be compared based on an objective measure. Thus, evaluating the difference in the expected economic gain of the investment and thereby identifying the most relevant technologies.

Ommen et al. [17] showed that the heat sink and heat source temperature differences have a significant influence on both the economic and technical constraints of conventional VCHPs. Ommen et al. [17] investigated four cases with different combinations of heat sink and heat source temperature differences. These are:

- ΔT_{sink} = 10 K, ΔT_{source} = 10 K
- ΔT_{sink} = 20 K, ΔT_{source} = 20 K
- ΔT_{sink} = 20 K, ΔT_{source} = 10 K
- ΔT_{sink} = 40 K, ΔT_{source} = 10 K

This study investigates the working domain of the HACHP for these four cases. For each case the operating conditions, defined by the heat supply temperature, $T_{\text{sink,out}}$, and the temperature lift, ΔT_{lift}, will be varied. The heat supply temperature from 40 °C to 140 °C and the temperature lift from 0 K to 70 K. For each operating condition the ammonia mass fraction and circulation ratio were found by an optimization of the net present value (NPV) under the technical constraints of the studied components.

The working domains are presented graphically for all four cases, showing which operating conditions are possible to supply with a HACHP while respecting both the technical and economic constraints. Finally, the present value (PV) of the HACHP will be compared to the PV of the best available VCHP technology presented by Ommen et al. [17].

Figure 8 and Figure 9 shows the derived working domains of the HACHP under the four investigated sink/source configurations and under the application of both the LP R-717 and HP R-717 components. The grey areas in the figures indicate the operating conditions that comply with all technical and economic constraints. However, it should be noted that the dashed blue line presented in Figure 8 and Figure 9 indicates the point at which the optimal solution attains the maximum allowable pressure. Thus, all solutions to the left of the dashed blue line have pressures below $p_{H,max}$ while all solutions to the right of the dashed blue line have $p_H = p_{H,max}$. The solutions to the right of the dashed blue line are thus feasible solutions but they operate on the pressure boundary to attain the best economy.

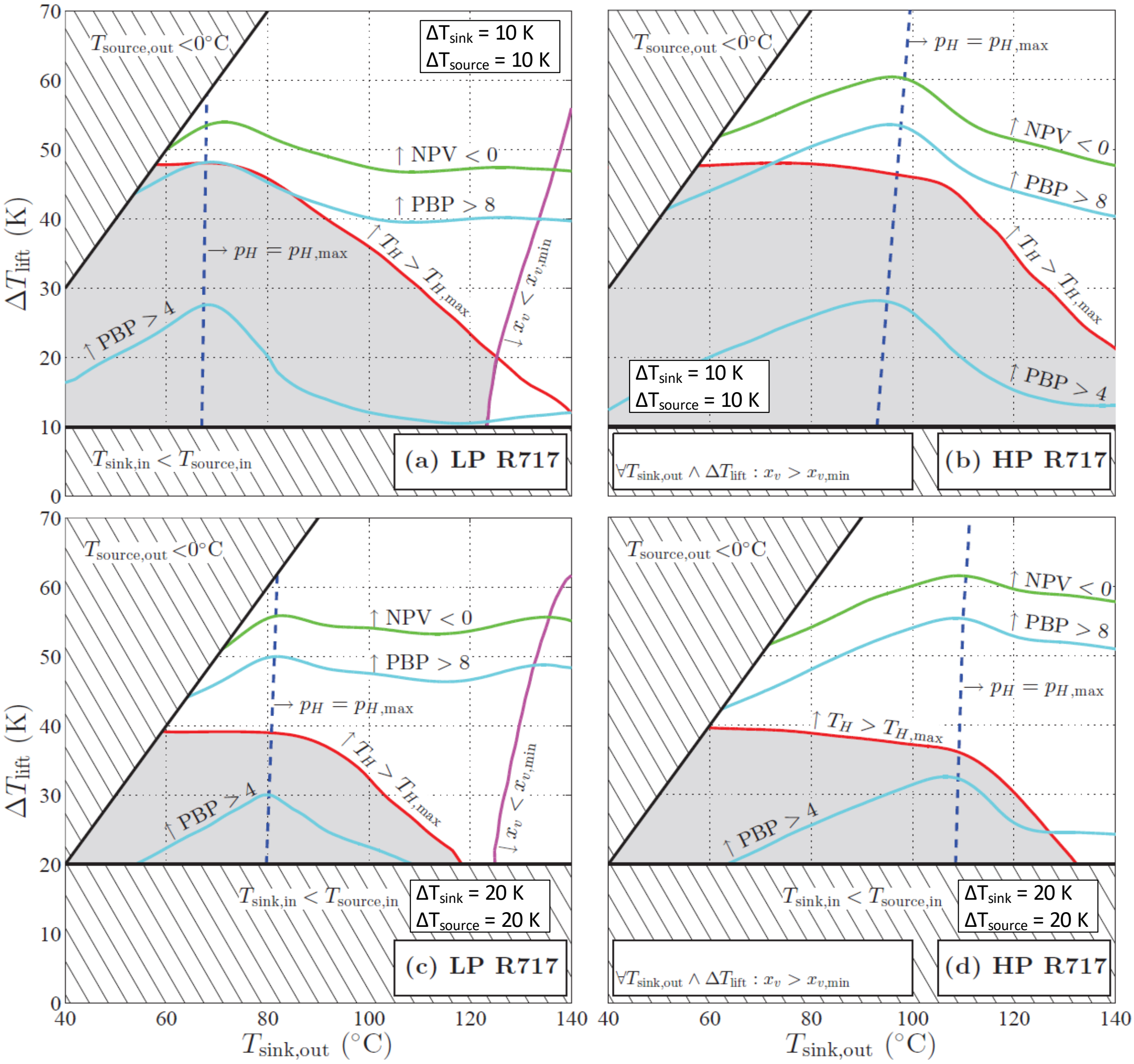

Figure 8 - Working domains for HACHP using LP R-717 and HP R-717 compressors at ΔT_{sink} = 10 K, ΔT_{source} = 10 K (a) and (b), and ΔT_{sink} = 20 K, ΔT_{source} = 20 K (c) and (d) [1].

As may be seen from Figure 8(a), the HACHP using the LP R-717 components at a sink/source configuration of 10 K/10 K can deliver a maximum heat supply temperature of $T_{sink,out}$ = 125 °C and a maximum temperature lift of ΔT_{lift} = 48 K. It can be seen that the maximum heat supply temperature is limited by the vapour ammonia mass fraction, x_v, while the maximum temperature lift is limited by the compressor discharge temperature. Further, it may be seen that all technically feasible solutions attain a positive NPV and almost all attain a simple PBP lower than 8 years. To attain a PBP below 4 years a maximum temperature lift of 28 K can be attained at a heat supply temperature of 65 °C.

Applying the HP R-717 compressors to the 10 K/10 K sink/source configuration, the maximum heat supply temperature is increased to above $T_{sink,out}$ = 140 °C, see Figure 8(b). Further, it can be seen that the maximum lift of 48 K can be retained at higher heat supply temperatures when applying the high pressure compressor. It can be seen that the maximum heat supply temperature and maximum temperature lift are both bound by the compressor discharge temperature. Again, all technically feasible solutions attain a positive NPV.

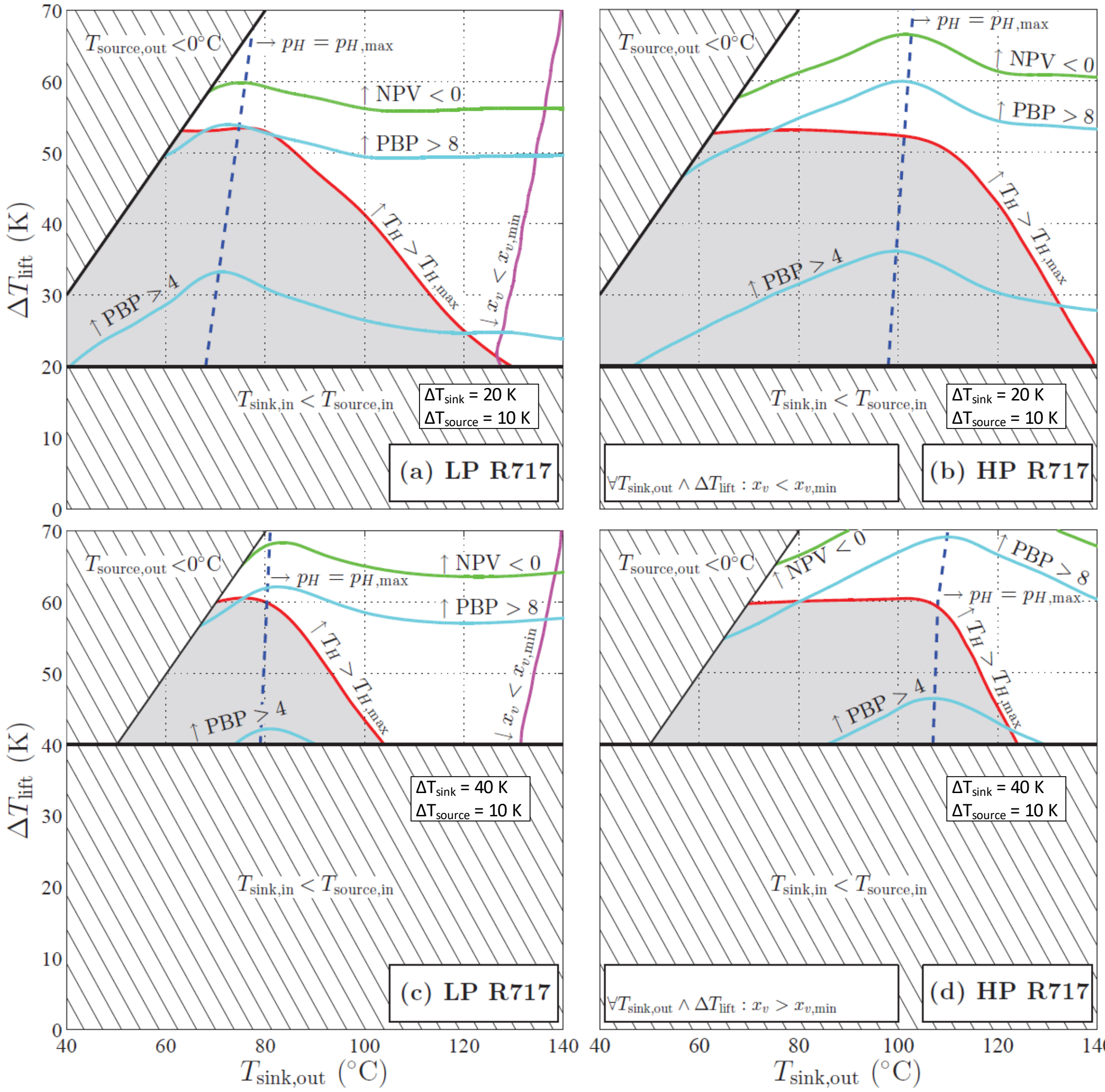

Figure 9 - Working domains for HACHP using LP R-717 and HP R-717 compressors at ΔT_{sink} = 20 K, ΔT_{source} = 10 K (a) and (b), and ΔT_{sink} = 40 K, ΔT_{source} = 10 K (c) and (d) [1].

For the sink/source configuration of 20 K/20 K, Figure 8(c) & Figure 8(d), the maximum temperature lift was 42 K while the maximum heat supply temperature was 119 °C for LP R-717 and 130 °C for HP R-717. For 20 K/10 K, in Figure 9(a) & Figure 9(b), this is reduced to a maximum temperature lift of 54 K and a maximum heat supply temperature of 125 °C for LP R-717 and 135 °C for HP R-717. For sink/source configuration 40 K/10 K, Figure 9(c) & Figure 9(d), the maximum temperature lift is 60 K while the maximum heat supply temperature is 100 °C and 125 °C for LP and HP R-717, respectively.

It can be seen that mainly the working domain for the LP R-717 compressor at a sink/source configuration of 10 K/10 K is bound by the x_v constraint. For 20 K/10 K a minor area is also limited by x_v. All other working domains are limited entirely by the compressor discharge temperature. Further, all technically feasible solutions have a positive NPV and almost all have a PBP below 8 years. In general, when increasing the sink glide the maximum heat supply temperature is reduced while the maximum lift is increased.

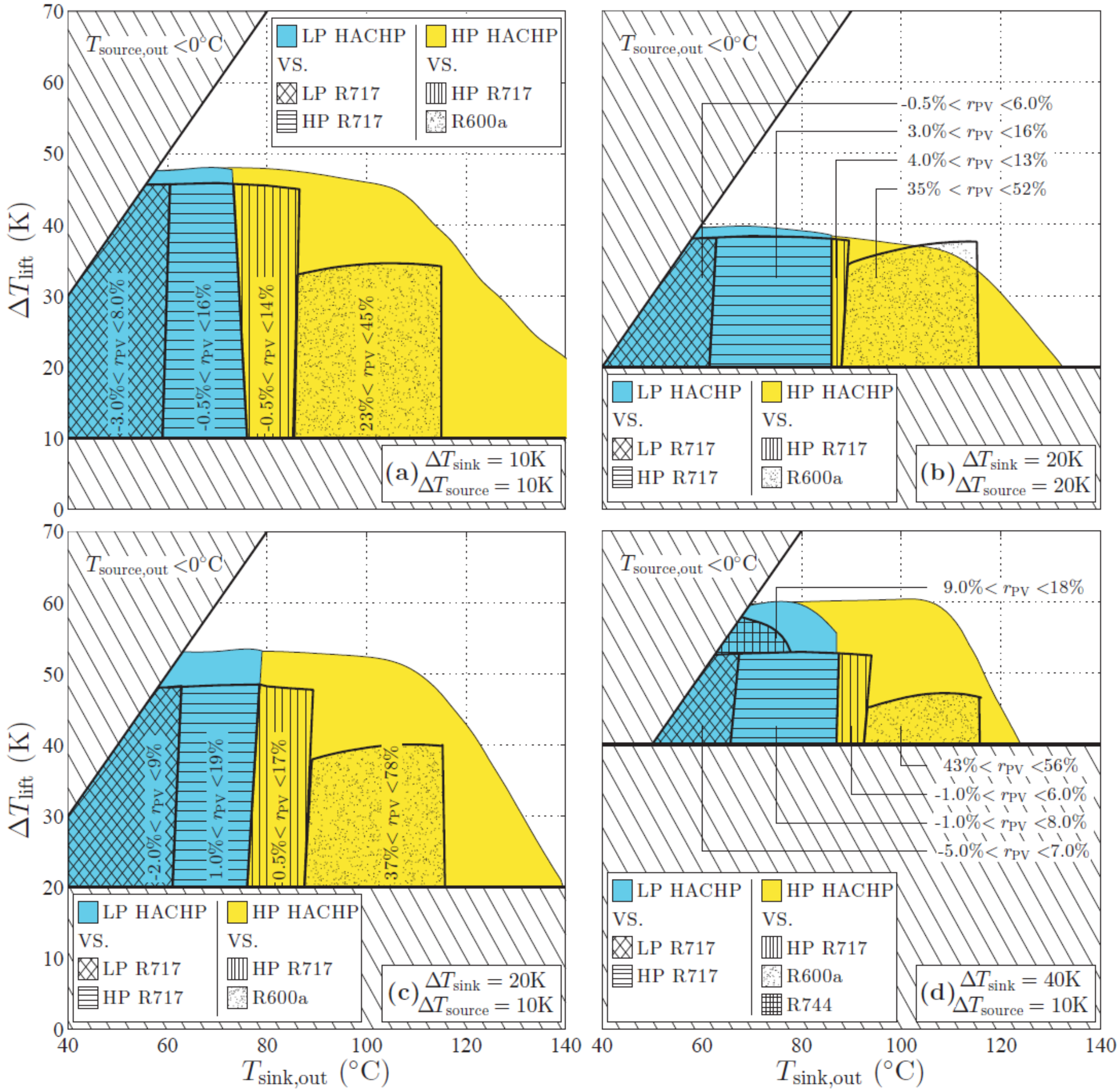

Figure 10 - Most profitable HACHP and comparison with the best available VCHP. The blue area indicates the region in which the NPV of the LP R-717 is higher than the NPV of the HP R-717 solution. The hatched areas indicate which VCHP is the best competing technology[1].

From Figure 8 and Figure 9 it can be seen that the working domains of the LP and HP R-717 components overlap in the low temperature range. It is therefore relevant to evaluate when it is more profitable to choose a high pressure solution over the low pressure solution. Figure 10 shows which component technology yields the highest possible NPV for all four evaluated sink/source configurations. As seen, this results in two regions: a low temperature region (blue) in which the LP R-717 components should be used and a high temperature region (yellow) in which the HP R-717 components should be used. The supply temperature at which the high pressure option becomes favorable is approximately $T_{sink,out}$ = 75 °C for 10 K/10 K, $T_{sink,out}$ = 85 °C for 20 K/20 K, $T_{sink,out}$ = 78 °C for 20 K/10 K and $T_{sink,out}$ = 90 °C for 40 K/10 K. If this is compared to the dashed blue lines in Figure 8 and Figure 9 it may be seen that the switch from LP to HP R-717 happens approximately 5 °C to 8 above the point where the LP R-717 reaches $p_{H,max}$. This is due the retention of the high ammonia mass fraction for the HP R-717 meaning that the compressor volume will be smaller and the heat transfer coefficient will be higher. Hence, the HP R-717 investment becomes smaller than the LP R-717 investment although the investment in the HP R-717 components are higher per unit of displacement volume and heat transfer area.

Further, Figure 10 shows the working domain of the best available VCHP, as concluded from [17]. As seen, the HACHP competes mainly with low and high pressure R-717 and R-600a but also with R-744 for 40 K/10 K. Comparing the working domain of the HACHP to the working domain of the best possible VCHP solution it is clear that the HACHP expands the range of operating conditions for which heat pump application is technically feasible and economically viable. The HACHP allows both higher temperature lifts and higher heat supply temperatures, especially when ΔT_{sink} is large.
Figure 10 also shows the difference in cost between the HACHP and the VCHP. This is represented by the relative difference in present value r_{PV}. As seen, r_{PV} is between -5.0 % and +9 % for the range in which low pressure HACHP competes with low pressure R-717. For low pressure HACHP against high pressure R-717, r_{PV} is between -1.0 % and +19 % while high pressure HACHP versus high pressure R-717 results in r_{PV} is between -1.0 % and +17 %.

For the range where the HACHP competes with R-600a the r_{PV} is between 23 % and 78 %. This is due to the large investment and poor COP associated with R-600a. Hence, for heat supply temperatures above $T_{sink,out}$ = 90 °C where R-717 can no longer be applied, the HACHP seem to be the more profitable solution. Also, for the range where LP HACHP competes with the transcritical R-744 heat pump, the HACHP seem to be the preferred option with r_{PV} between 9 % and 18 %.

5.5 Integration studies at end-users

Several cases were analyzed with respect to the integration of a hybrid heat pump at the end-users under consideration of energy flows and temperature levels of the processes. In addition to this, a general market survey for hybrid heat pumps was conducted. These studies are each summarized in the following sections.

The learnings made from these integration studies were furthermore utilized in the projects EUDP 13-I j.nr. 64013-0164 ”Hybrid varmepumpe til fjernvarme”, and EUDP 14-I j.nr. 64014-0127 ”Experimental development of electric heat pumps in the Greater Copenhagen DH system - Phase 1”.

SPX-Anhydro

A feasibility study about integrating a high temperature heat pump with sections of a spray drying plant (for example dryer dehumidification combined with air-preheating) were carried out at SPXFLOW Anhydro. This feasibility study is documented in the report “System integration of full scale ultra-high temperature heat pump - a report on development of ultra-high temperature hybrid pump for process application, SPXFLOW-Anhydro, Denmark, 2016”, and is summarized in this section.

Spray drying is the most commonly used industrial process for drying of various products and is the most suitable process for drying of products like milk, whey, maltodextrin, baby food powder etc. Spray drying starts with the atomization of a liquid into a spray of droplets. These droplets are put in contact with hot air in a drying chamber. The spray drying process (overall concept shown in Figure 11) is a multiple-step process consisting of:

1. Dehumidification and heating of incoming process air
2. Atomization of the concentrate into very fine droplets in a hot air stream
3. Water evaporation
4. Separation of the powder from the dryer exhaust air

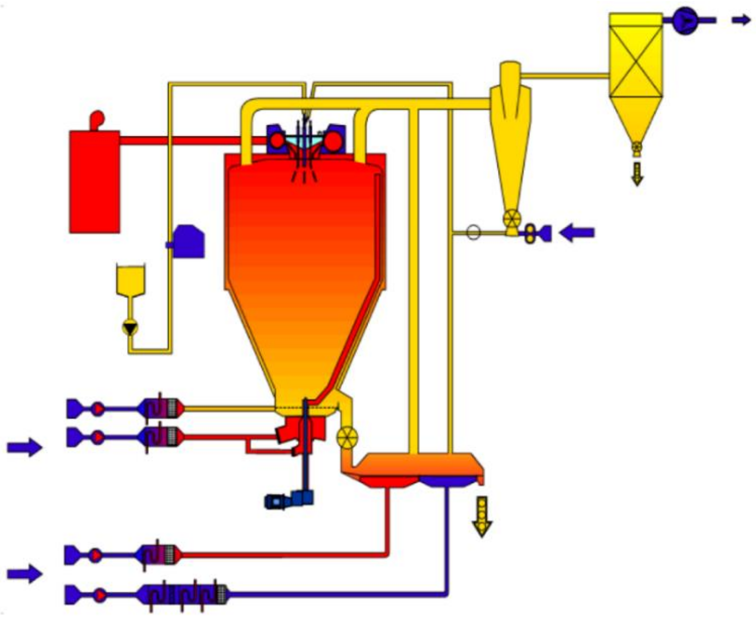

Figure 11 – Typical Anhydro spray drying facility [18].

Industrial spray drying is an energy intensive operation, as it is estimated that a spray drying plant typically needs between 4500 kJ and 11500 kJ to evaporate 1 kg of water. The highest energy loss in a spray drying plant occurs in the exhaust air of the dryer. In addition to the dryer exhaust heat, the low-temperature heat removed during the dehumidification of the main inlet air to the dryer is also significant and if recovered can lead to substantial energy savings. The energy can be recovered in a high temperature heat pump, and hence lead to a reduction in usage of fossils fuels (furnace oil, natural gas etc.). An obvious utilization of the recovered heat is to pre-heat the incoming dryer air.

The dehumidification process can improve the performance of the spray drying plant by maintaining the same low moisture content throughout the year. One way of carrying out the dehumidification is by cooling the air to its dew point. During dehumidification, the heat removed by cooling of incoming drying air is significant, and if not recovered efficiently can lead to loss of energy (to a cooling tower). A heat pump is hence proposed in this study to be integrated as shown in Figure 12, where the heat released in the dehumidification process is recovered, and used to pre-heat the incoming drying air and achieve improved energy economy of the spray drying plant. The cold water generated from heat pump HP-1 is used to cool the incoming drying air to temperature T2. The cooling of drying air in cooler HE-02 is done with the chilled water generated by heat pumps HP-3 and HP-4, which are working in series. The heat from the condenser of heat pump HP-4 is used to pre-heat the dehumidified cold air in first heater HE-03 to a temperature of T4, before the air is preheated further in HE-04 by HP-2, which is the high temperature hybrid heat pump.

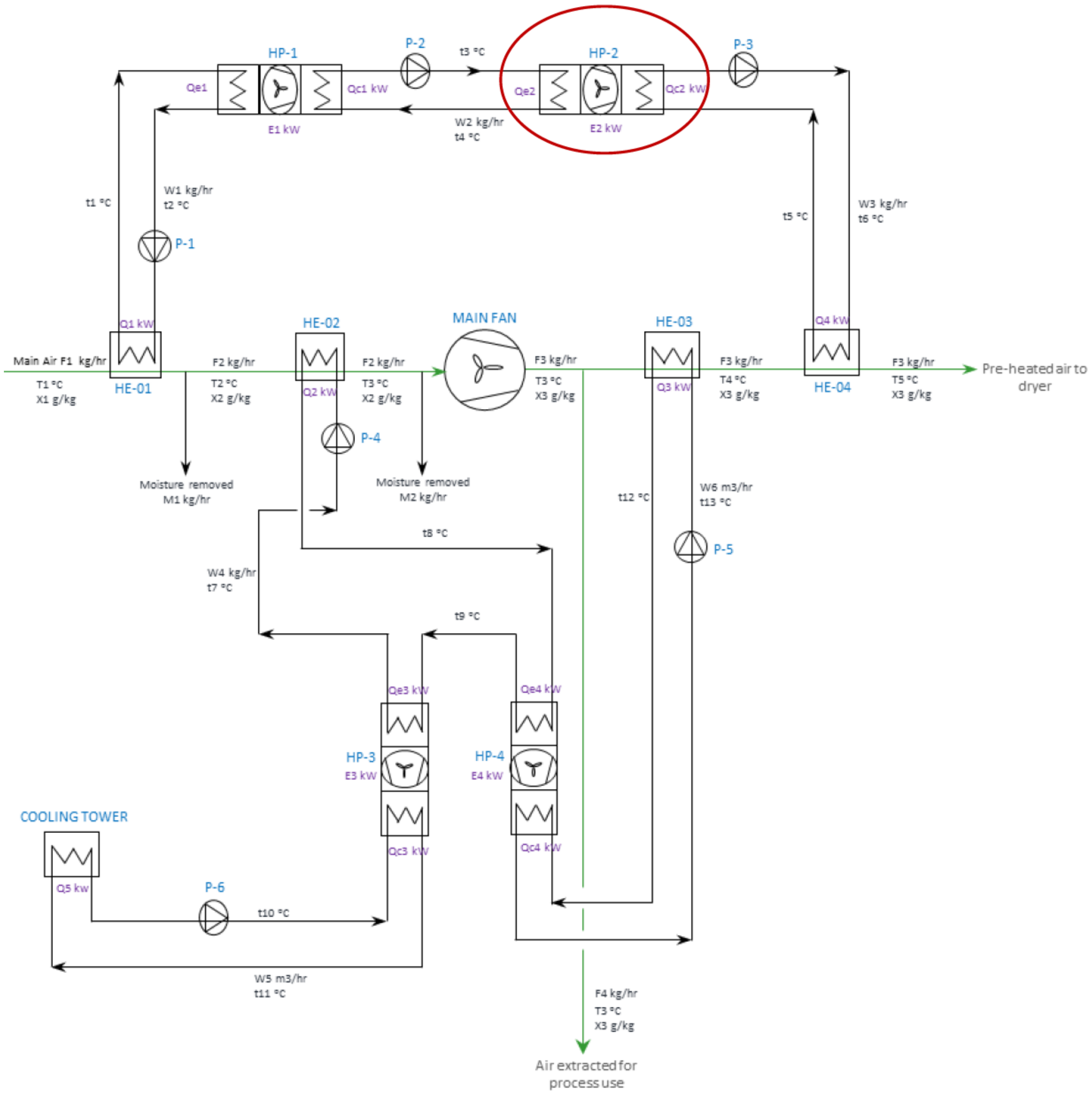

Figure 12 – Process scheme with an integrated hybrid heat pump in a spray dryer plant. High temperature heat pump, HP-2, marked in red circle [18].

The proposed process scheme in Figure 12 is applied to two cases in different regions, to analyze the effect of different ambient conditions and utility costs.

- Case A) High temperature hybrid absorption-compression heat pump integrated in a spray dryer plant located in South America producing non-caking permeate powder with a water evaporation rate at 4332 kg/hr and with fuel oil as a utility in the plant.

- Case B) High temperature hybrid absorption-compression heat pump integrated in spray dryer plant located in Europe also producing non-caking permeate powder with a water evaporation rate at 4152 kg/hr with steam as a utility in the plant.

An excel sheet-based calculation tool was developed, where the mass and energy balances for the process scheme were calculated. In Figure 13 an example of the results from one of the calculations of case A can be seen.

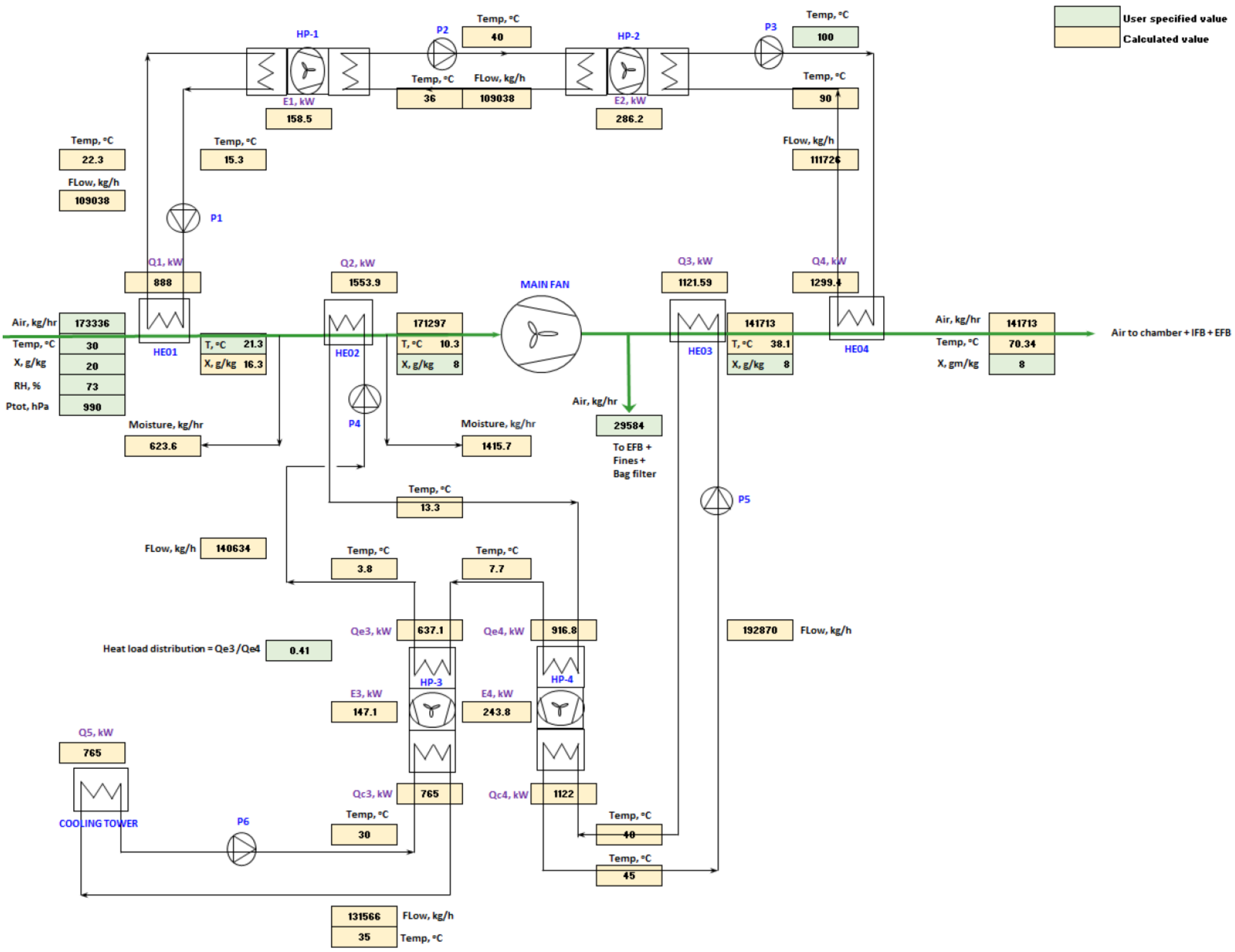

Figure 13 – An Integrated hybrid heat pump in a spray dryer plant in South America [18].

For case A the COP was assumed to be 3.6, together with an assumed outlet temperature of the hybrid heat pump of 100 °C. The monthly average absolute humidity in Buenos Aires was analyzed together with data for the consumption of utilities including power to the heat pump. The savings to justify additional investment in heat pumps based on the assumptions and the analysis were calculated to be 268.475 €/year.

For case B the monthly average absolute humidity in France were analyzed together with the estimated consumption of utilities, yielding the heat pump based process scheme in this case to be 20.760 €/year.

The savings in case B is estimated less attractive compared to case A. This is due to the difference in site conditions, primary the properties of the ambient air and the utility data. Hence, in order to increase the economic viability of the hybrid heat pump a large difference between the cost of electric power and cost of steam is needed. The analysis showed that a COP of at least 3 in these boundary conditions is needed in order to make the implementation of this technology feasible.

Bigadan

Bigadan upgrades biogas to natural gas quality, and an analysis has been made of the energy- and mass balances, including a pinch analysis, in order to investigate the possibilities for integration of a hybrid heat pump in the process, both at the biogas plant in Horsens and in Kalundborg (which at that time was under construction). The investigation defined a suggested size of the heat pump and suitable places for integration of a heat pump, and preliminarily a promising potential for integrating a heat pump in the process was shown. However, more detailed analysis of the technical details and the economic terms were needed in order to make final decisions about building a demonstration plant at this end-user.

ARLA

Various studies were made in relation to the process at ARLA, and one of the conclusions was that the cost target for a large-scale high temperature heat pump at 80 °C to 90 °C was 3.000 kr/kW_{heat}, while it at 110 °C was 4.000 kr/kW_{heat}.

In the initial calculations suitable for the ARLA case, and with assumptions based on the hybrid heat pump already running at Arla Foods Arinco in Videbæk, a temperature on the source side was set from 80 °C to 55 °C, while it on the sink side was set from 90 °C to 110 °C and from 90 °C to 130 °C. In addition to this a screw compressor was assumed to be used to accommodate for the relative high pressure level expected. These calculations show expectable COPs of 4.3 and 3.3, respectively, for the two different temperature glides on the sink side.

Industribejdsning Nord

At Industribejdsning Nord in Aalborg it was investigated if process heating could be made based on a hybrid heat pump where the heat source is district heating water. The process heat at Industribejdsning Nord should in this case supply an evaporator used for treatment of waste water in a pickling process for stainless steel and aluminum.

The calculations made took yearly operating costs at different district heating scenarios, COP, and yearly operating hours into consideration.

Market Survey

A market survey was made in the project to investigate market potential for installing high temperature heat pumps. The focus in this survey were mainly the following 4 sites.

- Nordalim, a company located in Århus which produces urea-formaldehyde. During the production process one step is an exothermic process in a catalysator releasing heat at 650 °C, hence a part of the need for process heat is “free” due to this process, and a part of the surplus heat is therefore distributed to the district heating network. One potential part of the process for a hybrid heat pump could be the evaporator where heat is added at 140 °C.

- A mink food fabrication plant in Holstebro. Here is a cooling need of waste products from chicken and pig slaughterhouses. A heating need for boiling and evaporation is covered with steam at 175

°C, while hot water at 70 °C is needed for cleaning. The analysis showed that a heat source in this setup preferably should be 70 °C to make it economically feasible to install a hybrid heat pump.

- Arla Taulov, which is a cheese dairy plant. In this production process a part of the cooling is made with ice-water, and a part of the heating is made with hot water from a natural gas boiler. In general, a big part of the process is already utilizing heat recovery, however there is still a potential for implementing a heat pump which utilizes the heat from the condenser in the ice-water system to create heat for the production process.

- Koppers, located in Nyborg. Here tar is distilled to a number of products, including additives to diesel oil and dyes. The distillation process occurs in several columns, where heat is added in the bottom at 200 °C to 300 °C. The heat is mostly generated with combustion of leftover distillates, which cannot be sold. The process has a high heat surplus, and a part of this surplus heat is also distributed to the district heating network. But overall, the generated heat is made very cheap due to the “free” access of leftover distillates, hence integration of a heat pump is not attractive in this case.

The integration studies made in this market survey showed that it is not trivial to implement a heat pump in an existing process, and that several issues need to be considered when analyzing the economic feasibility, e.g. if free residue products are available on the site. For the investigated four cases, there were a limited potential to implement a high temperature hybrid heat pump.

Furthermore, a screening was also made as part of this survey. This resulted in more than 400 potential customers to a high temperature hybrid heat pump in Denmark within segments in the pharma, food, oil, gas, and energy industry have been identified.

As a part of the market survey a presentation was made which shows a series of application examples on how a hybrid heat pump can be integrated in the process industry. One of these examples is shown in Figure 14, where the waste heat from an evaporator is utilized by a hybrid heat pump.

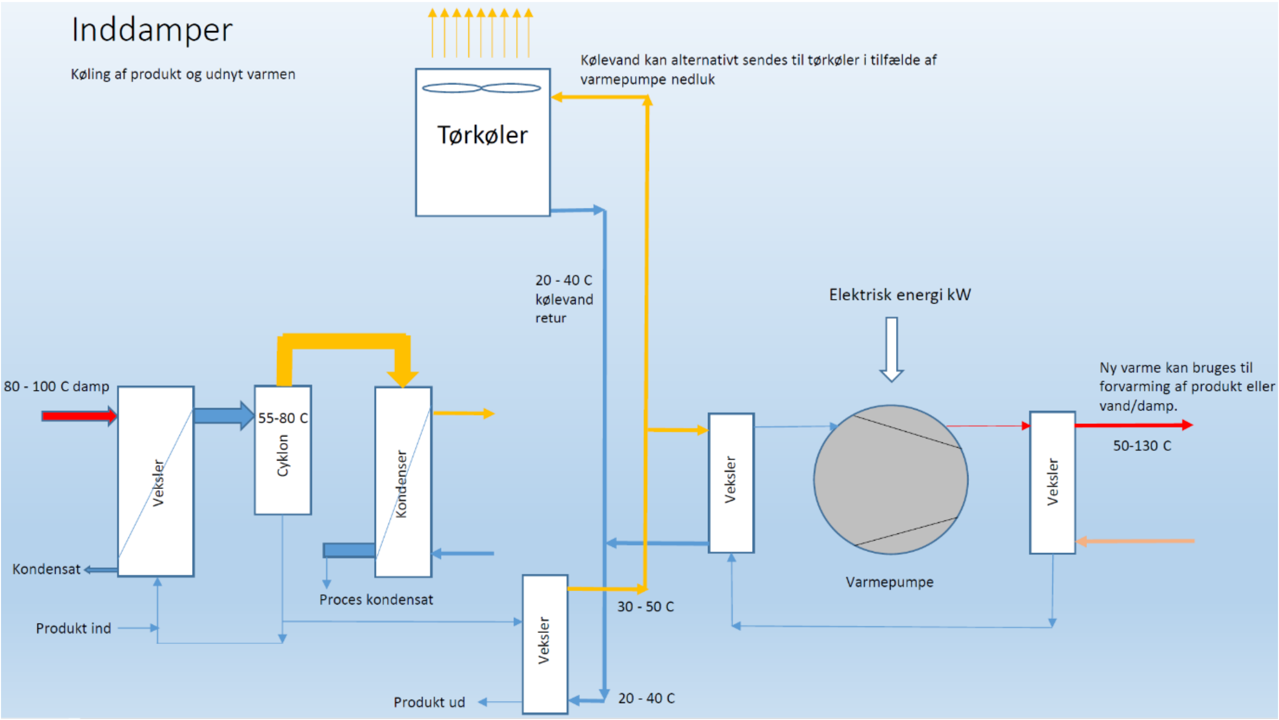

Figure 14 – Application example - Utilization of waste heat from an evaporator [19].

6. Utilisation of project results

The project has expanded the knowledge of the hybrid heat pump process and supported the work to commercialize the technology for both Innoterm and Hybrid Energy.

The project has carried out investigations both economically and technically to identify possibilities for extending the operating area for the technology, e.g. for a single-stage piston compressor cycle, and also to identify cost reductions. Cost reductions are as an example investigated on the heat exchangers and the compressor type. For the heat exchanger it has been investigated to design it as one heat exchanger for both the ab- and desorber part, instead of designing it as two heat exchangers.

Also, the operating strategy has been investigated. An example of this is a new strategy for achieving higher COP based on sensitivity analysis of the process with focus on the liquid flow and the system pressure was made. This procedure made is now a possibility in the control system of the two-stage hybrid heat pumps, which HE manufactures.

Furthermore, a number of end-users has increased their knowledge about hybrid heat pumps, and the possibilities for integrating a heat pump in their energy system. In general, the project has led to several inquiries and investigations about integration of heat pumps in existing processes, where the opportunity for a high temperature heat pump has been the starting point.

Additionally, the project has increased the awareness of the hybrid heat pump process, which is reflected in the list of dissemination activities (please see chapter 8).

During the project period Hybrid Energy has sold a number of hybrid heat pumps, e.g.:

- A two stage heat pump to Løgumkloster District Heating, which is related to EUDP project 13-I j.nr. 64013-0164 ”Hybrid varmepumpe til fjernvarme”. The capacity is 1100 kW and it is designed for a sink temperature of 85 °C.

- A two-stage hybrid heat pump for a waste water treatment plant in Oslo with a capacity of 800 kW.

- A hybrid heat pump at TINE Dairy, Norway. The application here is waste energy recuperation and hot water production with a temperature of 95 °C. The capacity is 900 kW and it has a $COP_{heating}$ (seasonal) of 5.4.

The work in the project has provided a considerable impact on later research in the field based on the use of zeotropic mixtures as working fluids in heat pump systems. In particular the work has provided inspiration for further development in these projects:

- THERMCYC - Advanced thermodynamic cycles utilising low-temperature heat sources (*DSF 1305-00036B*). The project aimed at solutions for thermal plants, for power generation, heat pumping and cooling by use of low value sources as waste heat and renewable sources at high efficiency. The THERMCYC project included five PhD projects, which all included focus on zeotropic mixtures as working fluids – in power cycles, in heat pumps, in heat exchangers and in compression and expansion machines. In addition, performance of zeotropic mixtures was tested in plate exchangers leading to new correlations for calculation of heat transfer.
- MIREHP - Mixed refrigerant heat pumps/cooling systems (*EUDP 64016-0045*). The project included demonstration of zeotropic mixtures using Propane and CO_2. The research part of the project included modelling of performance for a large range of zeotropic mixtures and screening of their performance.

In teaching the results of this project have been used in the course 41416 Energy Systems – Analysis, Design and Optimization for Master's degree students at DTU. The course focuses on the integration of thermal systems into the energy system, and the results from the project have been used as part of the content related to exergy analysis. Three student projects have been completed in relation to the project:

- Thomsen, Patrick Durup. Integration of Hybrid Heat Pump for Utilization of Industrial Waste Heat, Master's thesis, 2013.
- Kantsø, Jacob Agerbæk. Kompression af Vand-Ammoniakblandinger. Bachelor of Engineering thesis, 2013.
- Poulø, Philip Bolbroe. Modellering af Hybridvarmepumpe. Bachelor of Engineering thesis, 2014.

7. Project conclusion and perspective

The project has successfully expanded the knowledge of the hybrid heat pump process and hence supported the work to commercialize the technology. The theoretical work included fundamental research and development with the hybrid heat pump process, where advanced exergy analyses were connected with economic analysis in order to investigate technical optimization possibilities and economic consequences.

Studies about the integration possibilities for a hybrid heat pump at the end-users have been made, and additional application case studies were completed. A one-stage pilot demonstration plant was also designed, however even though the technology has a high potential and has gained a lot of interest from the industry, it was not possible to build a demonstration plant, as the scope for the project partners changed during the project.

Demonstrating the hybrid heat pump within the project was found to be more challenging than anticipated, which resulted mainly from the required commitment from end-users associated with the investment cost in the demonstration plant under consideration of the project conditions. In addition, alternative technologies with competitive performances and life-time cost approached the market, resulting in an increased competition for the hybrid heat pump technology.

In general, the interest for high temperature heat pumps are increasing, hence the potential for the hybrid heat pump also increases. A number of commercial hybrid heat pump plant are today already operating in the industry, where the technology is particularly suitable for applications with large temperature glides, both on the sink and source side. In one of these heat pumps, the nominal operating temperature is 95 °C.

With the increasing focus on high temperature heat pumps, the number of suppliers that can supply alternative types of high temperature heat pump technology also increases, while only a limited number of potential suppliers for high temperature heat pumps existed by the beginning of the project. Hence, in order to further successful expanded the technology more cost reductions are expected to be necessary, in order to be cost competitive with other new types of high temperature heat pump technology, that are able to provide similar high temperatures.

Some further perspectives for the hybrid heat pump technology, including cost reduction possibilities are given below and are based on [20]:

- To minimize limitations for the compressor as a constraining component a solution could be to use multi-stage compression with intercooling or liquid injection, hence higher temperature lifts at lower discharge temperatures should be possible.

- Regarding design and operation of the absorber and desorber the trend is to have vertically arranged plate heat exchangers, however there are still challenges associated with the optimal design to achieve best possible liquid–vapor distribution, and hence there is an optimization potential for this.

- When operating the solution pump, a focus needs to be on avoiding cavitation occurring at low net positive suction head (NPSH), which there also is a potential to cost optimize.

The project has been disseminated widely and has increased the awareness of the hybrid heat pumps as several journal publications, peer-review conference papers, and presentations have been made.

8. Appendices

The main publications, presentations, reports and list of dissemination activities for this project are listed in this chapter.

Project reports:

- System integration of full scale ultra-high temperature heat pump - a report on development of ultra-high temperature hybrid pump for process application, Process-Research & Development, Global Design Center, SPXFLOW-Anhydro, Denmark, 2016.

PhD Thesis:

- Jensen, J. K., Industrial heat pumps for high temperature process applications. A numerical study of the ammonia-water hybrid absorption-compression heat pump, PhD Thesis, DTU, 2015.

Journal Publications:

- Jensen, J. K., Markussen, W. B., Reinholdt, L., Elmegaard, B., Exergoeconomic optimization of an ammonia-water hybrid absorption–compression heat pump for heat supply in a spraydrying facility, International Journal of Energy and Environmental Engineering 2015; 62, 195-211, doi:10.1007/s40095-015-0166-0.

- Jensen, J. K., Markussen, W. B., Reinholdt, L., Elmegaard, B., On the development of high temperature ammonia-water hybrid absorption compression heat pumps, International Journal of Refrigeration 2015, doi:10.1016/j.ijrefrig.2015.06.006

- Ommen, T., Jensen, J.K., Markussen, W.B., Reinholdt, L., Elmegaard, B.. Technical and economic working domains of industrial heat pumps: Part 1 - single stage vapour compression heat pumps. International Journal of Refrigeration 2015; 55: 168–182. doi:10.1016/j.ijrefrig.2015.02.012.

- Jensen, J.K., Ommen, T., Markussen, W.B., Reinholdt, L., Elmegaard, B.. Technical and economic working domains of industrial heat pumps: Part 2 - ammonia –water hybrid absorption-compression heat pumps. International Journal of Refrigeration 2015; 55: 183-200. doi:10.1016/j.ijrefrig.2015.02.011.

- Kærn, M. R., Modi, A., Jensen, J.K., Haglind, F., An Assessment of Transport Property Estimation Methods for Ammonia–Water Mixtures and Their Influence on Heat Exchanger Size, International Journal of Thermophysics 2015; 36 6: 1468-1497, doi:10.1007/s10765-015-1857-8

- Kærn, M. R., Modi, A., Jensen, J.K., Andreasen, J.G, Haglind, F., An assessment of in-tube flow boiling correlations for ammonia-water mixtures and their influence on heat exchanger size, Applied Thermal Engineering, 93: 623-638, doi:10.1016/j.applthermaleng.2015.09.106

Peer-reviewed conference papers:

- Jensen, J. K., Reinholdt, L., Markussen, W. B., Elmegaard, B., Investigation of ammonia/water hybrid absorption/compression heat pumps for heat supply temperatures above 100 °C, Proceedings of ISHPC 2014 - International Sorption Heat Pump Conference, University of Maryland.

- Jensen, J. K., Markussen, W. B., Reinholdt, L., Elmegaard, B., Exergoeconomic optimization of an ammonia-water hybrid heat pump for heat supply in a spray drying facility, Proceedings of ECOS 2014 - The 27th International Conference on Efficiency, Cost, Optimization, Simulation and environmental Impact of Energy Systems, Åbo Akademi.

- Ommen, T., Jensen, J.K., Markussen, W.B., Reinholdt, L., Elmegaard, B., Technical and economic working domains of industrial heat pumps: Part 1 - vapour compression heat pumps. 11th IIR - Gustav Lorentzen Conference on Natural Refrigerants - GL 2014. Chinese Association of Refrigeration.

- Jensen, J.K., Ommen, T., Markussen, W.B., Reinholdt, L., Elmegaard, B., Technical and economic working domains of industrial heat pumps: Part 2 - ammonia–water hybrid absorption–compression heat pumps. 11th IIR - Gustav Lorentzen Conference on Natural Refrigerants – GL 2014. Chinese Association of Refrigeration.

- Jensen, J. K., Markussen, W. B., Reinholdt, L., Elmegaard, B., Conventional and advanced exergoenvironmental analysis of an ammonia-water hybrid absorption –compression heat pump , Proceedings of ECOS 2015 - The 28th International Conference on Efficiency, Cost, Optimization, Simulation and environmental Impact of Energy Systems, ENSGTI.

- Ommen, T., Jensen, J.K., Markussen, W.B., Elmegaard, B., Enhanced technical and economic working domains of heat pumps operated on series, Proceedings of ICR 2015 - 24th IIR International Congress of Refrigeration, Japan Society of Refrigerating and Air Conditioning Engineers.

- Jensen, J.K., Kærn, M.R, Ommen, T., Markussen, W.B., Reinholdt, L., Elmegaard, B., Effect of liquid/vapour maldistribution on performance of plate heat exchanger evaporators, Proceedings of ICR 2015 - 24th IIR International Congress of Refrigeration, Japan Society of Refrigerating and Air Conditioning Engineers.

- Reinholdt, L., Kristofferson, J., Zühlsdorf, B., Elmegaard, B., Jensen, J. K., Ommen, T., Jørgensen, P. H., Heat Pump COP, part 1, Generalized method for screening of system integration potentials, Gustav Lorentzen Conference in Valencia 2018, DOI: 10.18462/iir.gl.2018.1380

- Mancini, R., Jensen, J. K., Reinholdt, L., Markussen, W. B., & Elmegaard, B. (2019). Design optimization of plate heat exchanger absorbers and desorbers for hybrid absorption compression heat pumps. In Proceedings of ECOS 2019: 32nd International Conference on Efficiency, Cost, Optimization, Simulation and Environmental Impact of Energy Systems.

- Zühlsdorf, B., Jensen, J.K., Reinholdt, L., Elmegaard, B., Comparison of working fluid mixtures in different heat pump cycles, ICR Conference 2019 in Montreal, DOI: 10.18462/iir.icr.2019.1166

Student thesis (DTU):

- Master thesis: Integration of Hybrid Heat Pump for Utilization of Industrial Waste Heat (Patrick Durup Thomsen, DTU, 2013)
- Bachelor thesis: Kompression af Vand-Ammoniakblandinger (Jacob Agerbæk Kantsø, DTU, 2013)
- Bachelor thesis: Modellering af Hybridvarmepumpe (Philip Bolbroe Poulø, DTU 2014)

Presentations:

- 11/2012: Presentation made at conference: "Energieffektivisering i industrien". Conference made at DTI with around 120 participants, and 30 presenters.
- 09/2013: Project presented at the "DTU-international Energy Conference".
- 10/2017: L. Reinholdt, "Higher HP COP through better temperature match", European Heat Pump Summit.
- 10/2017: B. Horntvedt, "16 years with high-temperature hybrid heat pumps" at the 1st International Symposium about High-Temperature Heat Pumps in Copenhagen.
- 11/2019: S. R. Nordtvedt, "Combined heating and cooling: Integrated ammonia-water heat pump in modern dairy production" at the 2nd International Symposium about High-Temperature Heat Pumps in Copenhagen.
- 02/2019: Torben Lehmann, Process Engineeing "Hybrid varmepumpe - Process flow diagram – Principper og steder varmepumpe kan bruges"

Articles and student thesis (NTNU):

- Bergland, M., Eikevik, T. M., Tolstorebrov, I., Optimizing the compression/absorption heat pump system at high temperatures, Proceedings of ICR 2015, Japan.
- Development of the Hybrid Absorption Heat Pump Process at High Temperature Operation (Anders Borgås, 2014)
- Optimizing the Compression/Absorption Heat Pump System at High Temperatures (Martin Glosli Bergland, 2015)

Software:

- Thermodynamic calculations of a 1-stage hybrid heat pump cycle made with EES calculation software, including a user interface. A request for a copy of the model can be made by contacting Jonas K. Jensen, DTU, email: jkjje@mek.dtu.dk.

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