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Høytemperatur varmepumper for industrielle anvendelser

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Master i produktutvikling og produksjon
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of Science and Technology

Department of Energy
and Process Engineering

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MASTER THESIS

for

student Gaute Glomlien

Spring 2013

High temperature heat pumps for industrial applications

Høytemperatur varmepumper for industrielle anvendelser

Background and objective

The consumption of primary energy for heating can be decreased by utilizing heat pumps in many industrial applications, as heat pump process can upgrade waste energy at a low temperature level (low quality energy) to energy at a high temperature (high temperature energy) using a fraction of the primary energy. Industrial scale heat pumps have until recently been limited to maximum temperatures of 75-80°C and thereby limiting the implementation of high temperature heat pumps. During the last years new components are available on the market and the use of high temperature heat pumps for waste heat recovery has found its way into the market. A maximum temperature of 100°C is still a limitation for these processes based on the traditional heat pump cycle fluids.

Newer types of heat pumps, like the hybrid heat pump, combine the sorption and the compression based process. This process has water and ammonia as working fluid, which makes it possible to reach temperatures of 110°C with standard industrial refrigeration components. The aim of this project is to increase the operational limits of industrial heat pumps, also including the hybrid process by using the new standard components that are approved for higher pressures and temperatures. By using new components the maximum temperatures can be as high as 180-250°C. This will open up for new markets in the food and processing industry.

The following tasks are to be considered:

1. Literature study of high temperature heat pumps for industrial applications
2. Theoretical aspects of the different processes and selection of the most promising working fluids
3. Make a market survey of available components at higher pressures and temperatures
4. Develop a calculation tool for the process
5. Calculate different possible energy saving potentials for the different promising processes
6. Write a scientific paper with the main results from the thesis
7. Make proposal for future task not solved in this thesis

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Within 14 days of receiving the written text on the master thesis, the candidate shall submit a research plan for his project to the department.

When the thesis is evaluated, emphasis is put on processing of the results, and that they are presented in tabular and/or graphic form in a clear manner, and that they are analyzed carefully.

The thesis should be formulated as a research report with summary both in English and Norwegian, conclusion, literature references, table of contents etc. During the preparation of the text, the candidate should make an effort to produce a well-structured and easily readable report. In order to ease the evaluation of the thesis, it is important that the cross-references are correct. In the making of the report, strong emphasis should be placed on both a thorough discussion of the results and an orderly presentation.

The candidate is requested to initiate and keep close contact with his/her academic supervisor(s) throughout the working period. The candidate must follow the rules and regulations of NTNU as well as passive directions given by the Department of Energy and Process Engineering.

Risk assessment of the candidate's work shall be carried out according to the department's procedures. The risk assessment must be documented and included as part of the final report. Events related to the candidate's work adversely affecting the health, safety or security, must be documented and included as part of the final report. If the documentation on risk assessment represents a large number of pages, the full version is to be submitted electronically to the supervisor and an excerpt is included in the report.

Pursuant to "Regulations concerning the supplementary provisions to the technology study program/Master of Science" at NTNU §20, the Department reserves the permission to utilize all the results and data for teaching and research purposes as well as in future publications.

The final report is to be submitted digitally in DAIM. An executive summary of the thesis including title, student's name, supervisor's name, year, department name, and NTNU's logo and name, shall be submitted to the department as a separate pdf file. The final report in Word and PDF format, sientific paper and all other material and documents should be given to the academic supervisor in digital format on a DVD/CD-rom at the time of submission in DAIM.

☐ Work to be done in lab (Water power lab, Fluids engineering lab, Thermal engineering lab)
☐ Field work

Department of Energy and Process Engineering, January 16^{th} 2013

Prof. Olav Bolland
Department Head

Prof. Trygve M. Eikevik
Academic Supervisor

Preface

This thesis is written to conclude the five year Masters Degree Program I attend at the Department of Energy and Process Engineering at NTNU.

I would like to thank my supervisor professor Trygve M. Eikevik for providing me with an interesting issue, and for his guidance during the course of this thesis. I also wish to express my gratitude towards the industrial participants, EWOS and Tine Meieriet Verdal, for their time and help in creation of relevant case studies.

Gaute Glomlien

Faculty of Engineering Science and Technology
Department of Energy and Process Engineering

Thesis title**: High Temperature Heat Pumps for Industrial Applications**
Name: **Gaute Glomlien**
Date: **2013/06**
Supervisor: **Trygve M. Eikevik**

Abstract:
Industrial heat pumps are successfully integrated as low temperature heat recovery systems, but rarely employed in processes with heat requirements above 100°C. There is, however, a significant amount of waste heat available at these temperatures, and introducing a heat recovery system releasing useful heat at temperatures exceeding 180°C can have a substantial effect on the overall energetic performance of an industrial process. This thesis aims at identifying technological advances which support the development of high temperature, industrial scale heat pumps, and assess their applicability within the confines of two chosen industrial scenarios.

Successful application depends on the working principles of the heat pumps applied, the utilized working fluid and chosen components. Numerous types and technologies are available, making industry contact key to combine a selection thereof with relevant theoretical aspects, and perform a meaningful performance analysis. Drier processes were chosen for consideration, and revealed a peak temperature requirement ranging from 110°C to 185°C.

A calculation tool examining the energy efficiency of various heat pumps is developed, to nominate the most suitable alternatives among the identified heat pump types. The preliminary selection considers each heat pump's match to available thermal reservoirs, and estimates their expected energy efficiency. The most prominent solutions are further investigated. Available research and currently employed heat pump systems are used to develop system descriptions approaching real life applicability. A reexamination of their energetic performance is conducted (with increased accuracy in the calculation tool), to emphasize the limitations inflicted by select working fluids. Strenuous operational parameters are identified at various thermodynamic stages in the heat pump cycle, to examine the applicability of market available components.

The results indicate that mechanical compression heat pumps are most suitable for heat recovery within the identified scenarios. The thermodynamic cycles of a pure ammonia working fluid, and of a binary cascade system using steam-water/ammonia, are examined. Both reveal a significant potential for increasing energy efficiency of the drier processes. The benefits of using a hybrid heat pump are discussed alongside the obtained results.

Compressor(s) are expected to restrict system application. Market available components are operable at the estimated system pressures, but the scenarios bring them close to their upper temperature limits. Thus, further research is necessary to validate/discard the applicability and suitability of different components in high temperature, industrial heat pumps.

A review paper is written (*High Temperature Heat Pumps for Industrial Applications: A Review*) to present the obtained results and utilized approach in an abbreviated form.

NTNU

Faculty of Engineering Science and Technology
Department of Energy and Process Engineering

Oppgavetittel**: Høytemperatur Varmepumper for Industrielle Anvendelser**
Student: **Gaute Glomlien**
Dato: **2013/06**
Veileder: **Trygve M. Eikevik**

Sammendrag:
Industrielle varmepumper er effektivt integrert i varmegjenvinningssystemer ved lav temperatur, men tas sjelden i bruk når varmebehovet i gjenvinningsprosessene overstiger 100°C. Dersom industrielle varmepumper kan utnyttes ved høyere temperaturer, vil man kunne oppnå en betydelig positiv effekt på den samlede energieffektiviteten i en produksjonsprosess. Denne oppgaven sikter følgelig på å identifisere teknologiske utviklinger som støtter bruk av industrielle varmepumper ved økte temperaturnivåer, samt å vurdere deres egnethet og energieffektivitet i to utvalgte industrielle scenarier.

Effektiv varmegjenvinning med et varmepumpesystem avhenger av pumpens arbeidsprinsipp, arbeidsmedium og komponentene den bruker. Et bredt utvalg av tilgjengelige pumper og pumpekomponenter kompliserer prestasjonsanalysene, og understreker viktigheten av å knytte relevante teoretiske aspekter opp mot virkelige industrielle prosesser for å oppnå meningsfulle resultater. Produksjon av varmluft til bruk i tørkeprosesser er høyaktuelt for utnyttelse av varmepumper, og de identifiserte temperaturkravene strekker seg fra 110°C til 185°C.

Et analyseverktøy er utviklet for å undersøke ulike varmepumpers energieffektivitet i de valgte industrielle prosessene. En innledende analyse peker ut fremtredende varmepumpetyper, basert på arbeidsprinsippets match med det identifiserte scenarioet, og en estimert energieffektivitet. En detaljert systembeskrivelse gis av de fremtredende varmepumpetypene, før systemytelsen beregnes pånу (med økt presisjon). Arbeidsmediets termodynamiske parametere identifiseres, for å undersøke anvendbarheten og begrensende faktorer knyttet til markedstilgjengelige komponenter.

Resultatene tilsier at mekanisk drevne varmepumper er best egnet i de identifiserte scenarioene. Et rent ammoniakksystem, og et kaskadesystem med vann og ammoniakk i hver sin termodynamiske syklus, undersøkes, og begge er forventet å øke energieffektiviteten i varmluftsproduksjonen. Fordeler ved bruk av hybride varmepumper er også omtalt i den samme analysen.

Kompressorer er forventet å begrense varmepumpens arbeidsområde. Undersøkelser viser at markedstilgjengelige kompressorer tåler det estimerte systemtrykket, men analysene indikerer også at de bringes tett opp mot den øvre temperaturgrensen. Videre studier er følgelig nødvendig for å validere/ forkaste bruk av disse komponentene.

En vitenskapelig artikkel (*High Temperature Heat Pumps for Industrial Applications: A Review*) er skrevet for presentere oppgavens tilnærming til problemet, samt de oppnådde resultatene.

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1 Introduction

1.1 Motivation

The U.S. Energy Information Administration (EIA) is a statistical and analytical agency within the U.S. Department of Energy. It works to collect, analyze and disseminate independent and impartial information to promote the interaction between energy, economy and the environment, and defines four major sectors of energy consumption: the Residential sector, the Commercial sector, the Transportation sector and the Industrial sector [1].

The Industrial Sector includes manufacturing (defined as all establishments engaged in the mechanical or chemical transformation of materials or substances into new products) and nonmanufacturing (agriculture, construction, mining, and resource extraction) industries, and consumes more energy than any other end-use sector, about half of the world's delivered energy. Long-term projections made by the EIA estimate its average energy consumption to increase on average 1.5% per year through 2035 [1]. The energy consumption in the sector is dominated by five industries, accounting for more than 60% of the energy consumed in the sector put together. These five industries are Chemicals, Iron and steel, Non-metallic minerals, Pulp and paper and Nonferrous metals, which can all be defined as manufacturing industries. Their respective contribution to the energy consumption is represented in figure 1 [1].

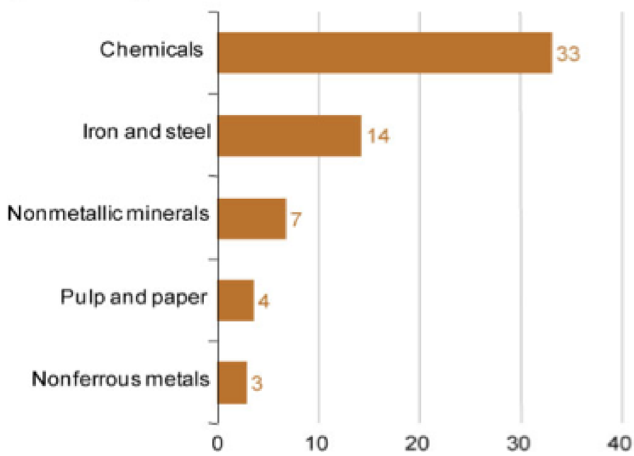


Figure 1: World industrial sector energy consumption by major energy-intensive industry shares, 2008

Common for all is a need for high temperature energy input from various heat sources, accompanied by a high operating cost of feedstock energy. The industries have consequently focused on reducing their energy consumption for years by recovery of excess process energy. Current environmental awareness has also led to increased regulations on emissions, strengthening the appeal of energy recovery systems as they limit the need for non-renewable energy sources in production. Recovery of waste energy and recycling of materials and fuel inputs have consequently become a conventional way of increasing the energy efficiency of an industrial process.

However, the industries still hold a large potential for increased energy efficiency. Current publications from the International Energy Agency (IEA) estimates that manufacturing industries based in OECD countries can improve their energy efficiency by 18-26% compared to 2004 levels, alongside a reduction of CO_2-emissions between 19% and 32%, in terms of primary energy sources alone (energy sources as found in nature). [3]

This is illustrated by the government policy introduced in the Netherlands during the 1990s, aiming at a 33% increase of their energy efficiency between 1990 and 2020. The industrial sector contributed to a third of the total energy consumption, and 85% of its demand was related to heating purposes [4]. Additionally, as seen in figure 2 [4], a significant amount of energy holding a large potential for recovery, went to waste in the refining and chemical industry.

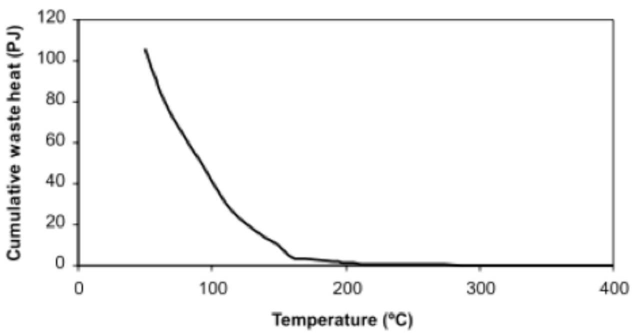


Figure 2: Cumulative waste heat in the refining and chemical industry in the Netherlands, 1999

Numerous solutions have been employed to take advantage of the energy saving potential, related to both energy recovery and recycling of materials and fuel. Industrial application of heat pumps addresses the first challenge. The heat pumps upgrade low temperature excess and waste heat to higher and more useful temperatures, which can be reapplied in processes of interest. Such systems can increase the energy efficiency of an industrial system significantly. An additional advantage is the potential for reduced emissions held by heat pump application, as the required non-renewable heat sources are fully or partially replaced by the heat pumps.

It is important to note that heat pumps only carry a potential for reduction of the primary energy input, not an elimination of it. Heat pumps require work input to lift the excess heat temperature to higher levels. Successful integration of an industrial heat pump system requires that the prospective earnings accompanying its application and utilization be weighed against the pump's capital cost. However, when done correctly, it can deliver substantial benefits.

Consequently, industry scale heat pumps have been implemented with great success in the past. Early editions were generally limited to an upper operating temperature of 75-80°C [5], restricting their implementation in high temperature operations, a significant source to energy consumption and emission in industrial sectors. However, technological advances have found their way to market, which when implemented and/or combined allow generation of pressures and temperatures to deliver energy at maximum temperatures between 180-250°C using industrial heat pumps. [5]

Such combinations and their individual potential are explored in this thesis. Innovative process design and alternate use of existing solutions are in the wind among industry participants, creating cost effective solutions to increase the operational environment of industrial heat pumps. By identifying the key challenges of heat pump application in a set of generic industrial operations, one can hope to illustrate the power held by industrial scale heat pumps, and potentially propose a current best practice solution for high temperature operation.

1.2 Underlying Hypothesis

The underlying problem of this thesis is rooted in the potential for heat recovery in industrial processes, to increase the energy efficiency of utility systems as well as to satisfy the ever-increasing restrictions on operational emissions in the industrial sector.

Industrial scale heat pumps challenge these problems, and are used for refrigeration purposes as well as heat recovery at low temperatures today. Operational limitations have, however, restricted the potential in high temperature industrial applications. But current technological advancements have introduced new components and system designs to market, increasing the scope of heat pump utilization for heat recovery. A presentation of such components and their coupling, alongside their respective potentials for energy savings when applied in industrial processes, should be possible to provide.

This is the basis for the paper's hypothesis. Several heat pump systems compatible with high temperature heat recovery in industrial processes exist, but their utilization is not yet wide spread. By performing a market survey of available components, and estimating the potential for energy savings held by each respective heat pump system, it should be possible to determine and rate their relative applicability at different temperature levels. Emphasizing these aspects is beneficial to spur the implementation of industrial scale heat pumps in high temperature operations, and help lower operational cost in industrial processes as well as reduce their emissions of substances harmful to the environment.

1.3 Main Goal of the Thesis

The main goal of this thesis is to display the potential held by industrial scale heat pumps in high temperature applications, by evaluating the energy efficiency of currently available heat pump systems. Both theoretical and experimental developments are investigated. The evaluations focus on the energy saving potential held by heat pumps, as inclusion of other economical aspects (such as cost of equipment, installation cost, maintenance cost, local cost of energy, etc.) depend on a vast range of factors greatly complicating the analyses, and potentially marginalizing the results due to a high level of uncertainty. A presentation of a market available solution, and its individual components, concludes the thesis.

Secondary goals of the thesis include identification of the key characteristics of a generic set of high temperature industrial processes, and a market survey to identify currently available technologies with satisfactory operating capabilities. A calculation tool is developed to highlight the possible energy savings of each proposed solution.

A scientific paper is also written, to display the review, analysis and conclusions presented in the thesis in a concise manner, and hopefully illuminate the advantages of high temperature heat pump application in the industrial sector.

1.4 Scope of the Thesis

This thesis is limited by market available technologies applicable in high temperature industrial processes. Possible solutions can be based on existing arrangements or on novel developments, and explore effects of different working fluids as well as system components. The technologies must be capable of (*but not limited to*) delivering temperatures ranging between 180-250°C, and hold the potential to replace purchased energy when implemented. Further requirements are introduced through the study sets of generic high temperature industrial processes. Additionally, the evaluated technologies must be deployable in the current market, to ensure usefulness of the obtained results.

A calculation tool assessing the energy efficiency of each heat pump supports the analysis. The calculations are twofold. An initial estimation rates the applicability of each heat pump in the chosen industrial scenarios, based on each system's Coefficient of Performance. As this coefficient only displays a heat pump's maximum theoretical performance in a given setting, further thermodynamic investigations are necessary to compare the energy efficiency of each heat pump to the currently utilized heating applications in sufficient detail. Hence, a thermodynamic analysis estimating the energy consumption in the internally separable heat pump processes is conducted. Based on these results, an introduction to a readily applicable system using on market available components can hopefully be given.

1.5 Structure of the Thesis

The thesis consists of a literature review followed by an evaluation of the proposed technological solutions for high temperature heat pump applications.

Chapter 2 provides an overview of industrial scenarios chosen as a basis for the thesis analyses, and an introduction to high temperature application of industrial heat pumps. The working principles of potential heat pumps are briefly introduced, to aid creation of the calculation tool presented in chapter 3. Applicable working fluids are described, and market available heat pump components suitable for industrial application are identified.

Chapter 3 develops a simplified calculation tool used to evaluate heat pump performance based on a set of given conditions. Prominent system inefficiencies are introduced, and a mathematical relationship is developed to include their effects in the calculations..

Chapter 4 dissects the industrial processes introduced in chapter 2, to identify theoretical aspects of high temperature industrial processes relevant to heat pump application. The heat pumps are introduced as a means for energy recovery, and an unbiased foundation for the performance evaluation requires this initial analysis. A preliminary selection ensues, to determine the most suitable heat pump systems for application in the industrial processes.

Chapter 5 introduces analytical and experimental investigations of actual heat pump systems, based on the most-likely-to-succeed concepts from chapter 4, and the calculation tool is used to evaluate their performance within the identified scenarios. The applicability of market available heat pump components is assessed. Further work necessary to evaluate the real applicability of these heat recovery systems, is introduced to conclude the chapter.

Chapter 6 concludes the thesis, and is followed by a reference list and the necessary appendices.

2 Technical Review

2.1 An Introduction to Challenges Related to the Work

One of several challenges related to the work is identifying a set of industry relevant scenarios. Numerous technological solutions facilitate implementation of heat pump systems in industrial processes, but high temperature application remains untested ground. Detailed knowledge of the theoretical aspects related to different industrial processes is necessary to perform an energy efficiency analysis of sufficient detail to emphasize the advantages of heat pump application. To create a rigorous framework supporting the calculations in the thesis' analyses, industrial participants have been approached to gather technical details from processes where implementation of heat pumps systems may prove beneficial. Basing the calculations on real industrial processes, rather than on imagined scenarios, also introduce the challenges of matching a heat pump system to available process streams to the analyses, as opposed to solely calculating the systems' energetic performance. Obtained information from the industrial producers is presented in section 2.2.

Following the introduction of potential heat pump systems in section 2.3, a calculation tool is developed in chapter 3, to evaluate each system's energy efficiency. These sections are necessary to create fertile analyses (with regards to diversity among the evaluated alternatives and their individual performance), and to propose the most-likely-to-succeed heat pump systems for high temperature industrial application. The calculation tool promotes the favorable features of each heat pump system, alongside the importance of its individual components.

Promising heat pump systems are to be presented with suitable, market available components at the end of this thesis, to ensure the validity and usefulness of the obtained results. This requires insights with component manufacturers keen to protect their innovations and proprietary rights, which may complicate creation of this small database. However, as the scientific paper presents the main findings of the thesis in a concise manner, it should encourage exchange of technical information and relevant experiences with the manufacturers.

2.2 High Temperature Industrial Scenarios

The industrial scenarios studied in this thesis are developed in collaboration with industrial participants, to ensure that the obtained results coincide with the challenges of real world application of high temperature heat pumps. Numerous industries utilize production processes which require heat at high temperature, and excess heat release is a common problem in the sector (refer to figure 2 and its graphical representation of cumulative waste heat in the Dutch refining and chemical industry). The manufacturing industry in general is subject to significant waste heat release, and is hence a suitable basis when creating these scenarios.

Large scale industries hold a potential for considerable energy savings if this heat is recycled, and seem a natural choice for a thesis exploring the energy gain obtainable by employing heat pump systems to recover the excess process heat. However, a large energy consumption and production output generally scale with economical muscle, and production processes are therefore often customized to ensure a maximized utilization of employed equipment. Accordingly, developing scenarios rooted in smaller industrial processes (where the applied heat pumps are based on market available technologies to reduce system cost) is beneficial to increase the relevance of the conducted analyses and applicability of the proposed solutions.

After discussions with and advice from my academic supervisor, Trygve M. Eikevik [29], scenarios where drying processes are an important part of the production line where pursued in this thesis. There are three main reasons for this. Firstly, drying requires high temperature heat addition to the process lines, with temperature levels elevated well above 100°C. This is outside the range of currently applied heat pump systems, rendering such processes highly relevant to the hypothesis of this thesis. Secondly, the process lines are prone to excess heat release due to general system inefficiencies and design, and the heat load requirements match those performed by generic heat pump systems (proven by their current application at lower temperature levels). Lastly, as the processes often use steam and air to produce and deliver the required temperatures, selected heat pumps must deliver heat to thermal reservoirs found in most industrial processes (due to their natural availability). Proposed solutions will, accordingly, emphasize heat pump applicability with typical process streams commonly used temperatures.

Several industrial manufacturers were contacted and asked to participate in scenario development. Two of the contacted parties were able to contribute with key process parameters (process flow charts, temperature levels, heat input requirements, etc.) to the thesis, which confirmed a potential for waste heat recovery in their production. The companies participating in the creation of scenarios are EWOS (represented by Geir Aspelund [33] and Mads Speichert [34]), one of the world's largest suppliers of feed and nutrition for farmed fish, and Tine Meierier (represented by Rune Frøseth [35]), Norway's largest producer, distributor and exporter of dairy products. Discussions with the industrial participants, as well as with Trygve M. Eikevik, were conducted to determine the most suitable areas for further investigation and implementation of heat pump systems. Discriptions of the industrial processes are presented in the ensuing sections, 2.2.1 (EWOS) and 2.2.2 (Tine Meierier), followed by section summarizing the available process streams of both scenarios.

2.2.1 EWOS

EWOS is a multinational company, established in 1935, and world leading within production and distribution of fish feed and nutrition for the international aquaculture industry. As of 2011, EWOS' held a 36% market share of the worldwide salmon feed market. [36] Fish feed is made from sustainable raw materials (of marine and vegetable origin), and produced as pellets. The tablet formed pellets are the center of the feed production, providing all the nutritional needs of farmed fish, and at the same time withstanding the strains of transportation (by truck, train and boat), storage (in sheds and silos) and transfer through the feeding systems (both automated and manual). [37] Feed quality (both physical and nutritional) greatly affects both biological and economic performance at fish farms, and account for a major part of the annual farmed salmon production costs. EWOS apply state-of-the-art production technologies to retain their market position, which EWOS themselves divide in to distinct phases: the feed factory; and the fish farm. The focal point of this scenario is the heat generation processes required the former.

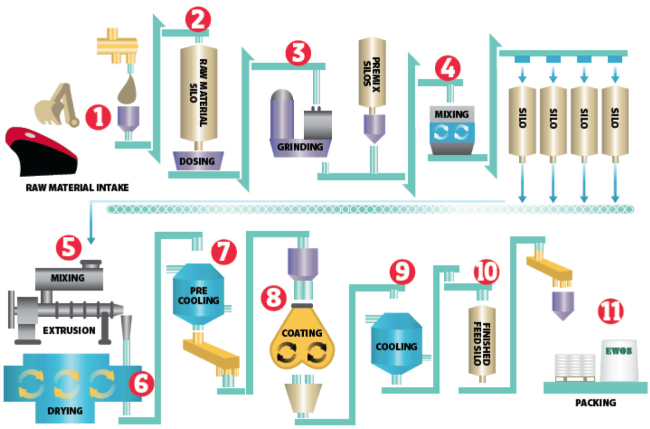


Figure 3: Fish feed production line

Figure 3 illustrates the steps transforming raw materials into the pellets used as fish feed [37]. The raw material is inspected and stored in a carefully controlled environment (step 1-2). Milling, mixing and extrusion of the materials are carried out next (step 3-5), ensuring that the nutrition content is homogeneous and of correct quality. Step 6 is drying of the pellets, a measure both increasing pellet lifetime and controlling the dryness of each pellet. Achieving correct dryness is crucial to regulate the fat uptake of each pellet, and keeping it bound within afterwards. Customer demands require varying pellet characteristics, and the drying process is carefully controlled to create differing compositions. The process requires a high temperature heat supply, and step 6 and the drying process is the focus of high temperature heat pump application in this industrial process. Further treatment of the process ensues this paragraph. Step 8 draws oil into the pellets, and the cooling processes of step 7 and 9 are important to ensure the fat retention and durability of the pellets. The finished product is stored (step 10) and inspected before it is packed (step 11) and shipped of to its recipient.

The drying process of step 6 uses a circulated air flow (initially 8.33 kg/s, but gradually decreasing due to the dehumidification) to heat the pellets [33, 34], evaporating the water entrained in each pellet. The hot air flow enters the drying chamber at ∼120°C, and is heated as presented in figure 4. The remaining temperature lift (from 70°C) is generated by four steam batteries. EWOS creates the steam, which is saturated at a pressure of 10 bar (∼180°C), using either a gas fired or an electric boiler (depending on the current cost of the respective energy sources). Each step of the drying process is carefully controlled, as process irregularities can severely compromise the pellet quality.

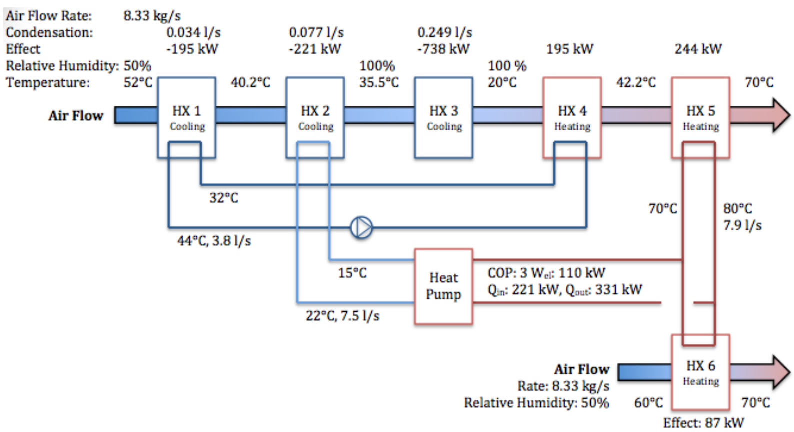


Figure 4: Dehumidifying a high temperature air flow for drying purposes

Figure 4 [33] illustrates how EWOS currently dries and preheats this airflow to 70°C, before supplying the steam heat load to elevate its temperature to dryer levels. (The depicted heat quantities depend on system heat transfer effects, such as cooler surface temperature, heat exchanger design, etc. A thermodynamic investigation, using the chosen calculation tool, is completed to ensure that the performance analyses have an even margin of error throughout the calculations. The alternative temperatures and heat requirements are described in detail alongside the scenario development.) A brief thermodynamic description of a basic dehumidification process is given to explain the depicted process. Warm air enters the dryer, heating the pellets to evaporate their entrained water. The evaporated water is absorbed by the air stream, which exits the dryer at a lower temperature and with increased water content (in this case, at 52°C and with a relative humidity of 50%). Direct discharge of this air to the environment is costly due to several factors (emission control, bad smell requires rinsing, etc.), and it is consequently recycled. This requires reheating of the air stream as well as removal of the entrained water. By lowering the air temperature through heat exchangers 1, 2 and 3 (from 52°C to 20°C), entrained water is condensed and removed from the air stream. A lowering of the air stream's saturation pressure with temperature produces this effect. As the air stream (now completely saturated with water at 20°C) is reheated through heat exchanger 4 and 5 as well as through the boiler, its saturation pressure increases, enabling it to absorb more water and be re-used for drying purposes.

Some of the heat extracted during the dehumidification process is currently being recovered. Heat exchanger 1 and 4 are coupled as a standard heat exchanger system, and recover 195 kW from the cooling process for heating purposes. The second heat exchanger extract 221 kW, which is used as the heat source of a heat pump system, dividing release of 331 kW of useful heat between two air streams (which are both raised to 70°C) through heat exchanger 5 and 6. It should be noted that air flow temperature from these heat exchangers cannot exceed 110°C, as it would require extensive retrofit of the hot air canals utilized in the drying processes. Heat exchanger 3 extracts 718 kW from the air flow using sea water, without any heat recovery.

EWOS' process line comprises two such streams generating hot air flows for drying purposes. They are identical, indicating that all utilities presented in figure 4 are available at twice the capacity.

10 bar saturated stream is produced at a rate of 10 tons/hr, and approximately 1622 kW of gas is required to generate the steam in the investigated process line. It is used for steam driven dryer applications and high-pressure hot water (at 90°C) production, in addition to the final temperature elevation of the hot air streams. 1411 kW of steam is used for these purposes. In addition to the available heat in the dehumidification process, two waste streams release excess heat to the

environment. One is a saturated air stream at 50 °C, the other an air stream with a relative humidity of 40% at 35°C. Both flow at rates of 100 000 m^3/hr.

It is evident that EWOS' pellet production process, and particularly the drying processes, hold the potential for heat recovery using industrial heat pump systems. Both heat sources and heat sinks are available, at several temperature levels. The potential of revising the systems using a high temperature industrial heat pump system to recover excess process heat, decreasing the required heat load from high-pressure steam, should therefore be investigated. A summary of the applicable streams is presented in table 3.

2.2.2 TINE Meieriet Verdal

The basis for modern dairy production was learnt from Swiss cheese makers in the early 1800's. During the late 1800's, cooling systems and separators in production introduced the processes utilized in the industry today. It was, however, the hygienic and technical standards developed during the 1960's and -70's that increased the industry's efficiency to the enterprice it is today. TINE have followed this development for more than 125 years, first established as a dairy farm cooperative in 1881, and is today the prominent producer, distributor and exporter of dairy products in Norway.[38]

TINE Meieriet Verdal was originally built as a cheese factory in 1976, but recently expanded its operation into butter and margarine production (as of 2011), as well as drying of whey (2012).[39] The production plant and equipment have been continuously upgraded to satisfy the requirements of a modern, efficient cheese factory. Cheese, butter and margarine products are all pasteurized, to increase the shelf life of the final products. Pasteurizing requires heat at high temperature, and then immediate cooling. Temperature levels and time spans of the heating and cooling processes depend on the specific product. TINE pasteurizes their raw milk for cheese production at 72°C, using steam and hot water. The cream used to produce butter and margarine is heated to 75°C or 95°C during this prolonging of product shelf life. However, the highest process temperatures are required in whey production. Whey is a by-product of cheese and other dairy production processes, and TINE Meieriet Verdal concentrate and dry it to Whey Protein Concentrate (WPC) and whey permeate powder. It is the spray drying of these products that require the highest temperature heat (air at 185°C and 160°C, respectively). Application of high temperature heat pumps is focused around these spray drier systems, and the temperature lift required by their affiliated drying air streams. Figure 5 and 6 provide schematic illustrations of the two drier processes.

WPC and whey permeate spring from the same sources, separated in a UV-filtration plant during the pasteurization process of whey. WPC is extracted from the filtration plant as a whey concentrate, which is sent directly to pre-drying heat treatment and finally the spray drier. The feed stream in figure 5 illustrates the WPC flow entering the spray drier, where a hot air stream subsequently dries it. The hot inlet air stream temperature is elevated from ambient temperatures using a combination of external energy (from hot water systems and a boiler) and heat recovery. The heat recovery system utilizes excess heat present in the drier's outlet air stream and in the boiler's flue gas, and reduces release of useful heat from the drying process to its surroundings.

Whey permeate is exits the UV-filtration plant in a diluted form, and the entrained water must be extracted before spray drying the permeate. After removal of the excess water, a series of pre-drier treatments is initiated. The whey permeate enters the spray drier as the feed stream in figure 6, where it is dried by a hot air stream. The hot air stream is elevated to the drier's temperature level using a similar system to that described for the WPC drying process. Temperature lift is generated by external energy sources and heat recovery (from the air stream exiting the drier as well as from the boiler flue gas), to increase the drier's energy efficiency.

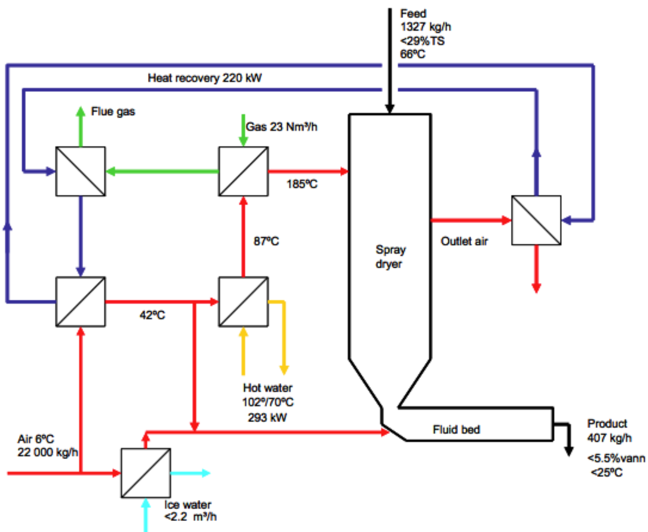


Figure 5: Schematic of the process flows related to drying of WPC

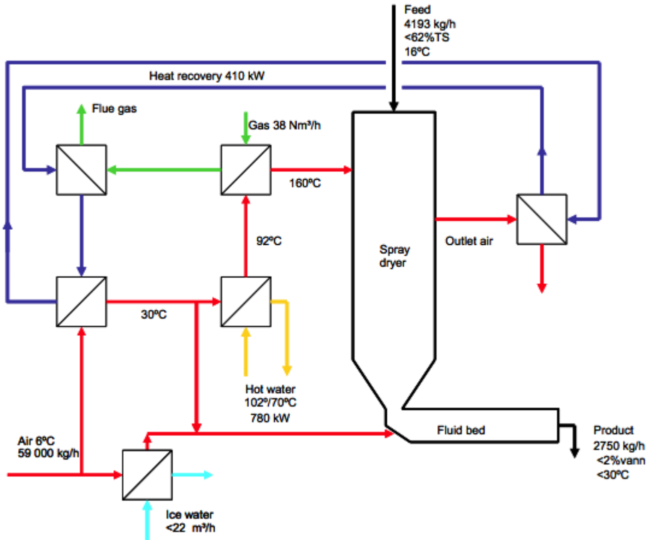


Figure 6: Schematic of the process flows related to drying of permeate

The brief process descriptions, alongside figure 5 and 6, clearly illustrate the similarities between the drier processes. Both utilize air as their drier medium, heated to drier temperatures using a heat recovery system, hot water and finally an externally powered (by gas or electricity, depending on the current energy cost) boiler. However, the WPC feed enters the drier at 66°C and 1327 kg/hr, whereas the permeate feed enters at 16°C and 4193 kg/hr. Thus, the driers cannot operate effectively using identical processes. Differing process characteristics are related to process temperature levels and the heat loads required to generate the respective temperature elevations. These diversities are important when evaluating heat pump application, and are accordingly elaborated on. [35]

The air flow of 22 000 kg/hr used in the WPC spray drier is preheated from 6°C to 42°C by the heat recovery system, using 220 kW of heat extracted from the cold drier outlet air and the flue gas. This lowers the drier's outlet air temperature from ∼70°C to ∼50°C. Hot water, delivering 293 kW to the process, further increases a large part of the air flow (80-85%) temperature to 87°C. 15-20% of the air is assumed sent to the fluid bed, based on the size of the system's air canals and fans. No exact number exists for this stream split. The heat exchange cools the water stream from 102°C to 70°C. The last temperature lift of the inlet air stream is generated by a ∼700 kW heat addition from the system boiler, elevating the air stream to the drier inlet temperature, 185°C.

Whey permeate is dried by a 59 000 kg/hr air flow at 160°C. The air stream preheated from 6°C to 30°C by a 410 kW heat recovery system (utilizing the drier's outlet air flow and the boiler's flue gas). This system cools the drier's outlet air flow from ∼70°C to ∼50°C, the same operational temperatures found at the outlet of the WPC drier. The same hot water heat source used for WPC drying (operating between 102°C and 70°C) elevates the air temperature to 92°C, using 780 kW to heat 80-85% of the initial air flow. 1200 kW is required to lift the air stream to the drier inlet temperature (160°C) using the boiler system.

A complete list of all heating and cooling processes at TINE Meieriet Verdal is provided in appendix 1 [39], and the processes operating at relevant temperature levels for this thesis have been highlighted. The tables express a need for heat at temperatures ≥100°C outside the spray driers as well. High-pressure hot water is used in pasteurization at the cheese factory, lifting the temperatures from 55°C to 122°C. High-pressure hot water at 100°C is also used for automated cleaning systems, and is generated from a heat source at 60°C. There is also waste heat available outside the drier systems. Among them is 70°C cooling water from compressor systems, available at a 25 m^3/day flow rate. Heat pumps are also capable of heat recovery from cooling processes. Several potential heat sources are identifiable from the cooling table in Appendix 1.

2.2.3 Process Overview

Based on the information obtained from the industrial processes at EWOS and Tine Meieriet Verdal, table 1 has been created to provide an overview of the available heat sources and heat sinks, with temperature levels, the accompanying heat load (when possible) and a set of short, descriptive notes.

A prominent feature of the described processes is the operational temperatures mismatch with those suggested by the thesis' introduction. Initial assumptions proposed a need for temperatures between 180°C and 250°C. In reality, the presented processes are bounded upwards at 185°C, utilized in TINE's WPC production process. The physical setup of EWOS' flow line actually restricts the temperature delivered by heat pumps to 110°C. However, as industrial heat pumps are yet to be implemented even at these temperature levels, identified systems capable of recovering heat in the identified scenarios cannot compromise the usefulness of the obtained results. If heat pump application is deemed beneficial at these temperatures, the goal of emphasizing the energy efficiency of heat pumps to industrial participants is met. And as figure 2 illustrates, an abundance of waste heat is available in this lower temperature range in other industrial processes as well.

Accordingly, the boundaries of high temperature heat pump application in industrial processes are pushed to a lesser degree than originally intended, but the potential benefits are illustrated in an operational area with more heat available for recovery.

It is noticeable that both processes are centered around heating of air, which both EWOS and TINE Meieriet Verdal use as the working medium in their drying processes. This is both beneficial and unfortunate for the analyses. The advantages are two-folded. Firstly, as both companies use air in their drying processes, the calculations evaluate the performance of heat pumps combined with a typical industrial heat sink. Secondly, heat exchange with one medium simplifies the calculations of each heat pump's heat transfer effects. The required adaptions to accommodate both scenarios with one calculation tool is minimized, and focus is kept on the heat transfer effects of the heat pump systems. It is, however, unfortunate that the scenarios fail to emphasize the wide applicability of heat pump systems, despite EWOS' operation with a psychrometric mixture and TINE's heating of "dry" (relatively) ambient air. The results cannot, in other words, be applied directly to process streams with a different working medium. But as producing the temperature lifts and matching process requirements and heat pump fluids are expected to be the restrictive forces of the analyses, a limited exposure to different heat transfer effects between the heat pumps and the thermal reservoirs do not compromise the validity of any drawn conclusions.

External heat is also needed to generate steam and high-pressure hot water at both EWOS and TINE, providing heat sinks at temperatures ranging from 100°C and to well above 180°C. A potential scenario could be developed around on these utility requirements.

EWOS		
Dehumidifaction	**Restriction**	**Notes**
Heat Source(s)		
Air Flow	T<52°C	Initially with a relative humidity of 50%. Cooled and reheated to dehumidify the air used to dry pellets. One of two identical air streams.
Waste Stream I	T=50°C	Saturated air stream with a flow rate of 100 000 m^3/hr.
Waste Stream II	T=35°C	Air stream with 40% relative humidity and a flow rate of 100 000 m^3/hr.
Heat Sink(s)		
Air Flow	T≤110°C	Restricted by air canal design near the heating chambers. One of two identical air streams.
High Pressure Steam	T≈180°C	High pressure saturated steam at 10 bar. Created at rates of 10 tons/hr.
High Pressure Hot Water	T=90°C	Created using 1411 kW of steam or 1622 kW of gas.

TINE Meieriet Verdal		
WPC drier	**Restriction**	**Description**
Heat Source(s)		
Drier Outlet Air	T=70°C ∨ 50°C	Available at 70°C prior to the heat recovery system, 50°C after.
Hot Water Heat Exchanger	T=70°C	Water leaving the hot water heat exchanger utilized in the air flow's second heating step.
Heat Sink(s)		
Air Flow	87°C<T<185°C	Lower restriction caused by the air stream temperature after exchanging heat with the hot water, the upper by the drier inlet temperature.
Permeate drier	**Restriction**	**Description**
Heat Source(s)		
Drier Outlet Air	T=70°C ∨ 50°C	Identical to the WPC drier
Hot Water Heat Exchanger	T=70°C	Identical to the WPC drier
Heat Sink(s)		
Air Flow	92°C<T<160°C	Lower restriction caused by the air stream temperature after exchanging heat with the hot water, the upper by the drier inlet temperature.
General	**Restriction**	**Description**
Heat Source(s)		
Cooling Water	T=70°C	From the air compressors, available at a flow rate of 25 m^3/day.
Cooling water	T=56°C	∼81 kW used to cool high temperature pasteurization processes to 50°C.
Cooling Water	T=60°C	∼408 kW used in a falsh cooler system (for whey permeate production). The cooling water's temperature is lowered to 35°C.
Heat Sink(s)		
High Pressure Hot Water	T=122°C	Lifted from 55°C. Used for pasteurization of at the cheese factory. Require ∼104 kW.
High Pressure Hot Water	T=100°C	Lifted from 60°C. Used for automated cleaning. Require ∼340 kW.

Table 1: Process overview

2.3 Heat Pump Technology

The challenges of high temperature heat pump application are established in the succeeding sections, to illustrate the focal points of heat pump selection and performance evaluations. It precedes a section introducing potentially applicable heat pump systems, given to support the development of a calculation tool analyzing the heat pumps' energetic performance. Working fluids relevant for high temperature application are also introduced.

Excess energy (heat) can be recovered passively or actively. Passive heat recovery implies direct energy transfer between two energy levels within a working system. Active heat recovery requires energy transfer from an external source to elevate low-grade energy to a higher level, and facilitates energy savings when conventional passive recovery is not possible.

Heat will spontaneously flow from a higher temperature reservoir to one of lower temperature, unless it is restricted by physical boundaries. Heat pump systems are able to reverse this flow, and transfer energy from lower temperature sources to higher temperature heat. External energy (to the two separate temperature reservoirs) is consumed in the process. Hence, a heat pump is classified as a means for active heat recovery, where excess, low-grade energy (typically from a waste stream) is upgraded to a higher, useful temperature through work input. This relays the fundamentals of all heat pump systems, regardless of their basic working principle. The required work (external energy) input depends on the systems desired temperature lift, and increases with elevated temperature difference between the hot and cold heat source. Costs of running the heat pump system are increased alongside this effect, potentially diminishing the advantage of heat pump implementation. [6, 7]

The idea is for the upgraded heat to replace purchased energy in the industrial system, thereby reducing the overall operational energy costs. Attraction to heat pumps as a cost-saving measure is particularly strong in environments where:

- The pump's heat output can be delivered at a temperature to replace purchased energy (such as boiler steam or gas firing)
- The system's saved cost of energy exceeds the heat pump's operational costs
- The net operating cost savings offset the required capital expenditures of a heat pump within acceptable time limits (typically 2 to 5 years)

2.3.1 High Temperature Application

When determining if industrial processes hold a potential for heat recovery, and whether a heat pump is a prospective solution to cost-effective heat regeneration at high temperatures, a heat pump type should be selected to maximize the benefits of the recovery system. And although several heat pump technologies suitable for industrial use exist, each solution's applicability in high temperature operation must be evaluated. This is not a straightforward selection, as high temperatures impose different challenges compared to low temperature applications (particularly with regards to system pressures). Three factors are key when evaluating the suitability of prospective heat pump types [6]:

1. The heat source
2. The heat sink
3. The required temperature lift

Even though these technical factors are well known, some relevant aspects are often overlooked when heat pumps are evaluated for high temperature applications. Process integration of the heat pump(s) is an example [8]. The temperature level(s) of waste heat is predetermined by the process fundamentals, but the level at which it becomes available for heat regeneration is determined by the utility system design (cooling water, air, etc.). Substantial energy recovery at higher temperatures may be realized by proper investigation of the industrial processes, and pinch analysis is commonly regarded as the most powerful tool for efficient integration.

Process integration is exemplified by the plant utility approach [9], a form of process integration where the upgraded heat is supplied to the plant's utility system to ease integration in existing industrial applications. Heat pumps implemented using this approach do, however, require high temperature lift capabilities (generally around 80°C). This trend is common in high temperature industrial application; the limiting factor when choosing a heat pump type is generally the high temperature lift requirement, from already elevated temperatures. Lift requirements of 100-150°C from heat sources at 80-150°C are common expectations [4], supported by the data illustrated in figure 2. It is, however, difficult to generalize evaluations the high temperature heat pump applicability based on the temperature lift capacity alone, due to large differences between industrial processes. The succeeding section introduces the working principle of potential technological solutions, familiarizing the reader with each heat pump type.

2.3.2 Working Principle

There are several different solutions to heat pump technology, but all are based on the three basic steps of operation described below, and illustrated in figure 7:

1. Recovery of (excess/waste) heat from a heat source; Q_C
2. Increase of temperature by work input; W_{net}
3. Delivery of (useful) heat at elevated temperature to a heat sink; Q_H

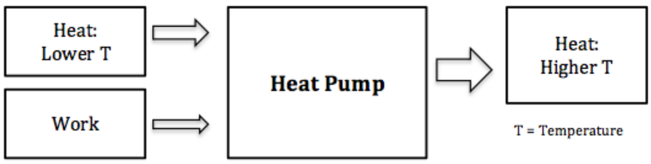


Figure 7: Simplified operating principle of a heat pump

Industrial heat pump design is commonly grouped in two categories, based on mechanical and thermal compression. These are further dividable into two subcategories, based on open- and closed-cycle operation. The four basic operating principles are consequently generalized as; Open-Cycle Mechanical Vapor Compression Heat Pumps, Closed-Cycle Mechanical Heat Pumps, Open-Cycle Thermocompression Heat Pumps and Closed-Cycle Absorption Heat Pumps. Each design is based on mature technological solutions, and all have been implemented in industrial applications previously. Key operating characteristics of the generalized heat pump systems are presented below.

Combinations of the four basic solutions (and other thermodynamic cycles) may also present suitable abilities for industrial application, and cannot be neglected in this analysis. Prominent systems for industrial application, capable of delivering high temperature heat in particular, are introduced in line with the basic systems.

Open-Cycle Mechanical Compression Heat Pumps

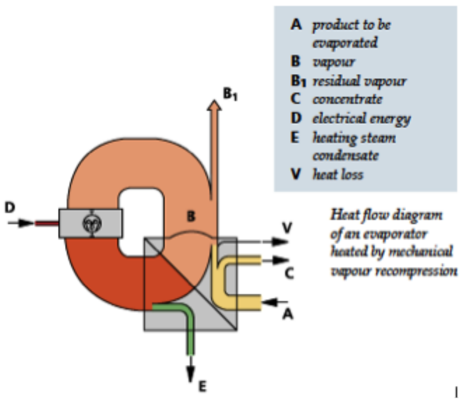


Figure 8: Operating principle of the open-cycle MVC heat pump

Open-cycle mechanical vapor compression (MVC) heat pumps operate with a single compression stage of its working fluid, typically excess steam exiting a process evaporator. Water vapor (the only operable fluid in these systems) is compressed to a higher pressure and temperature, and condenses against its designated heat sink to release useful heat. Condensate leaves the system, and the remaining energy in the heated stream is used to preheat the excess steam entering the compression stage, before it is disposed of as well. These systems are considered to be open as the vapor only goes through one cycle before leaving the heat pump system (i.e. one compression stage and one release of useful heat).[7, 11] A typical system is illustrated in figure 8 [11].

A MVC heat pump's setup eliminates the expansion valves typically found in closed-cycle mechanical heat pumps, as its working fluid is discarded after the condensation stage. (The main purpose of an expansion valve is to depressurize the working fluid for vaporization in the system evaporator, a superfluous operation in open-cycle systems.) And as open-cycle MVC systems are typically implemented in systems where steam production already exists, the need for investments in costly evaporators are eliminated as well. Accordingly, open-cycle MVC heat pumps require a mechanical compressor (operable by common mechanical systems) and a condenser heat exchanger, making it an easily implemented system.

Closed-Cycle Mechanical Heat Pumps

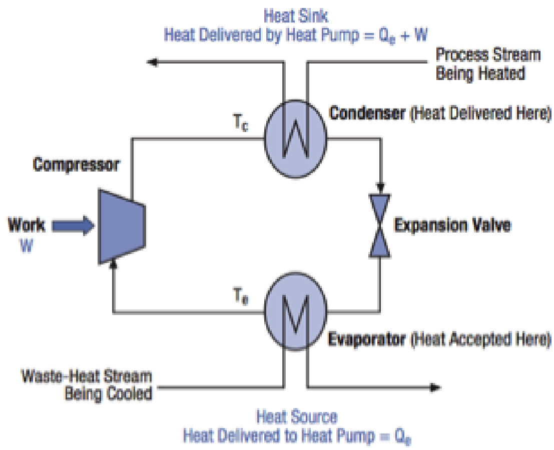


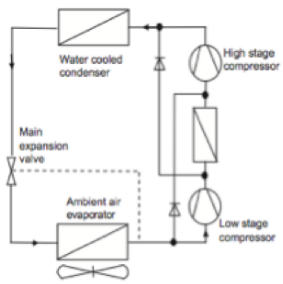
Figure 9: Operating principle of the single-stage closed-cycle mechanical heat pump

The simplest closed-cycle mechanical heat pumps consist of four basic elements: an evaporator, a compressor, a condenser and an expansion valve; as well as a working fluid transferring the heat from the cold to the warm reservoir. Closed-cycle operation refers to recycling of the working fluid, which is a pure component. [6, 7, 10]

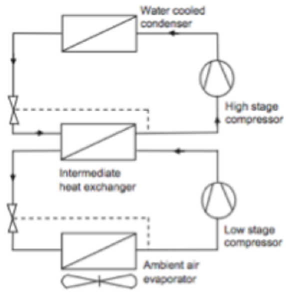
Excess heat is absorbed (isothermally, due to the pure working fluid) from a heat source to vaporize the working fluid in the evaporator, at a low temperature and pressure. A mechanical compressor increases the working fluid's pressure before it enters the condenser, elevating the vapor's condensing temperature. Vapor is condensed to deliver useful heat at the heat sink, fulfilling the purpose of the heat pump. Heat release is either isothermal or with a temperature glide, depending on the working fluid's critical temperature and the condenser temperature (refer to section 2.4 on working fluids and their heat transfer effects). The high-pressure working fluid is finally expanded to the low pressure and temperature required in the evaporator, and the working fluid restart its heat transfer cycle.

Figure 9 [6] illustrates this simplest form of closed-cycle mechanical heat pumps. The mechanical compression can be driven by most common mechanical systems, including electric motors, steam turbines, combustion engines, etc.

Multistage Mechanical Compression Heat Pumps



(a) Subcritical vapor compression



(b) Compression-resorption cycles

Figure 10: Operating principles of the multistage mechanical compression heat pumps

Multistage compression heat pumps are expanded closed-cycle mechanical systems, classifiable as either compound or cascade systems [15]. Schematics of the two cycles are presented in figure 10. The differentiating factor between single and multistage systems is related to compression of the working fluid; condensers, evaporators and expansion valves are implemented with aspects at the heat source and sink as sole considerations.

Compound systems have two or more compressors connected in series, ranging from lower to higher pressures. This decreases the required compression ratios and work input across each compression stage. Compression ratios are often kept approximately equal between compression stages, to maximize the system's energetic performance.

Cascade systems are constructed as a series of single-stage systems, where the condenser in a lower-temperature system transfers heat to the evaporator in a higher-temperature system. Several systems in series facilitate use of several working fluids, chosen to increase system efficiency in each temperature range.

Expanded systems have two main advantages compared to single-stage compression systems; increased energy efficiency over the compression stages (due to lower pressure increase over each compression stage), and potential for higher temperature lift between the heat source and sink. As the compression ratios over each compression stage is decreased, several mechanical solutions are applicable in these systems. Regardless, they must be chosen with care to meet the specific operational requirements of each case.

Hybrid Heat Pumps

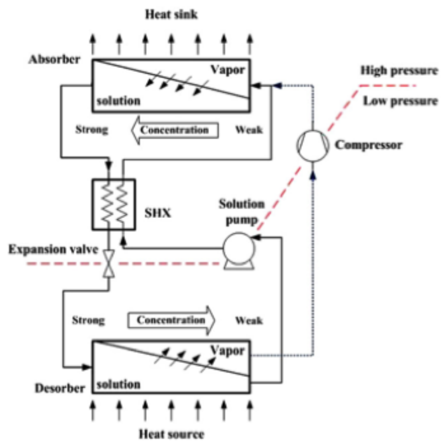


Figure 11: Operating principle of the hybrid heat pump

Hybrid heat pumps are based on the principles of mechanical compression systems, but utilize a two-component working fluid to take advantage of absorption heat transfer effects. [16, 17, 18] A schematic showing one of several hybrid heat pump designs is seen in figure 11 [16]. It illustrates a single-stage vapor compression system. Multistage compression (both as compound and cascade systems) is also applicable, and such systems often employ several internal heat exchangers to reduce external energy requirements.

The illustrated system employs a two-component working fluid, and the components are classified as either working fluid (the most volatile component) or absorbent. A rich/strong solution (working fluid highly solved in the absorbent) is introduced at low pressure to the desorber, where heat addition vaporizes the volatile component. Vapor is then extracted and compressed to a higher pressure. A solution pump works to increase the pressure of the remaining lean solution. The two high-pressure fluids enter the absorber, where they mix and release the generated heat of absorption to a suitable heat sink. The now rich solution is returned to the desorber through an expansion valve reducing its pressure, and the cycle is complete.

A single heat exchanger (SHX) is often applied at the high-pressure side of the system, to transfer heat from the rich solution leaving the absorber to the lean solution entering it. This increases the energy efficiency of the system, by lowering the required energy input from the solution pump.

Both heat transfer processes are operable with temperature glides, due to the boiling trajectories created by varying vapor concentrations in the heat exchangers. Boiling trajectories are controllable by managing the components' flow ratios, and allow highly efficient heat transfer between the heat pump and the process streams in the thermal reservoirs.

Open-Cycle Thermocompression Heat Pumps

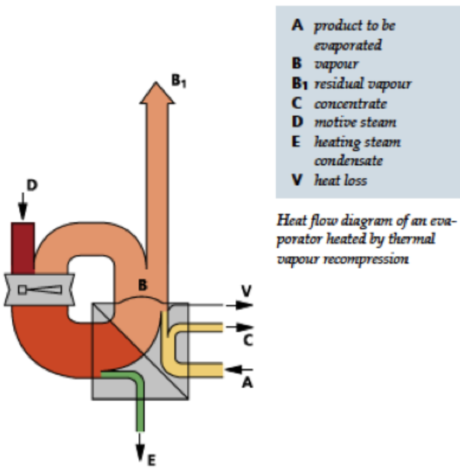


Figure 12: Operating principle of the open-cycle thermocompression heat pump

Open-cycle thermocompression heat pumps use a high-pressure motive stream to increase the energy of excess vapor from an industrial process [11]. Steam jet compressors, illustrated by figure 12 [11] are utilized to increase pressure (and thus the temperature) of excess vapor from an industrial process. As a jet-ejector is designed without moving parts, it constitutes a simple and highly reliable heat pump solution.

The motive stream and excess vapor process stream mix, and the jet ejector compress the resulting stream to a higher pressure and temperature. The warmer mixture stream is sent to a condenser to recover the useful heat created by this system. Surplus energy extracted in the evaporator corresponds to the energy supplied by the motive stream. Heat transfer is considered to be isothermal, as the mixed stream is close to a pure fluid.

As the MVC heat exchanger, expansion valves and evaporators are typically removed from the system, and the working fluid is not circulated.

Closed-Cycle Absorption Heat Pumps

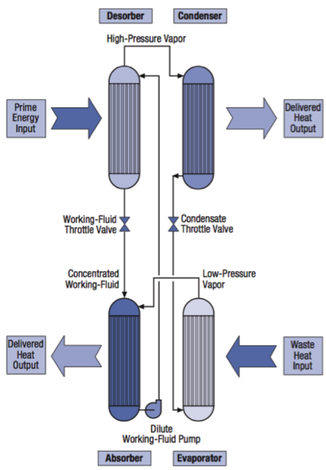


Figure 13: Operating principle of a type I absorption heat pump

Absorption heat pumps exist in two basic forms (type I and II), distinguished by their heat source temperatures (T_D) driving the heat pump, and the hot thermal reservoir (T_H). [14] Type I absorption heat pumps work as a heat amplifiers, and are used when $T_D > T_H$. Type II absorption heat pumps are called heat transformers, and boost the system temperature when $T_D < T_H$ to upgrade recovered waste heat to higher temperature levels. Both operate as thermally driven closed-cycle heat pumps, and utilize a two-component fluid to produce the temperature lift. The fluid pair is separable as a working fluid (the volatile component) and an absorbent.

Absorption heat pumps incorporate four heat exchangers. Heat is added in the generator, to desorb the working fluid from a rich solution of the absorbent. The vaporized working fluid condenses through the condenser, and vaporized in by the an external heat source in the evaporator. Vaporized working fluid enters the absorber, to release useful heat as it is absorbed into the lean absorbate solution returning from the generator. The absorber takes advantage of the heat of absorption and the boiling point elevation created by mixing the two-component fluid.

A type I absorption heat pump is illustrated in figure 13 [6], and the working fluids' thermodynamic cycle operate in system's with a high temperature heat source, and intermediate temperature heat sinks. Mixed fluid enters the generator at increased pressure from the absorber, provided by a solution pump. High-pressure vapor is boiled out of the working fluid by a high temperature external heat source, and subsequently cooled through the condenser to release useful heat. The condensate is the throttled to lower pressure, and recover heat by vaporizing in the evaporator. The vapor enters the absorber, and is absorbed by concentrated working fluid (transported by pressure differences) from the generator. This chemical process generates heat often assumed to be same temperature level as the condenser [24], as both deliver useful heat to the industrial process. Pumping the two-component fluid back to the generator pressure closes the thermodynamic cycle.

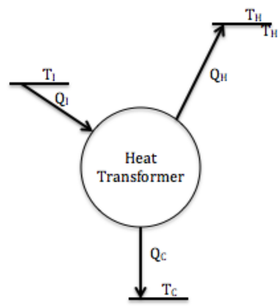


Figure 14: Schematic of a type II absorption heat pump

Type II absorption heat pumps, the heat transformers, operate in systems where $T_D < T_H$. Heat is upgraded with a minimized mechanical energy supply.[14] The thermal reservoirs are often assumed to have three temperature levels; a high temperature heat sink (T_H), two intermediate temperature heat sources (T_I) and a low temperature heat sink (T_C), as illustrated by the schematic in figure 14. [14]

A type II system comprises the same four heat exchangers as the type I heat pump, but run a slightly altered thermodynamic cycle. Type II systems have higher an absorber pressure than generator pressure, implying that the rich solution is pressure driven from the absorber to the generator. Heat at intermediate temperature is used to separate the working fluid from the absorbent in the generator. The low-pressure working fluid condenses at the lowest system temperature, and a pump is employed to increase its pressure pre-vaporization. High-pressure vapor is subsequently absorbed by the lean solution (pumped back to the absorber pressure by the solution pump) as it enters the absorber, and high temperature heat is released.

Depending on the utilized working pairs, heat transfer effects are either isothermal or with a temperature glide. This goes for both type I and type II systems.

Adsorption Heat Pumps

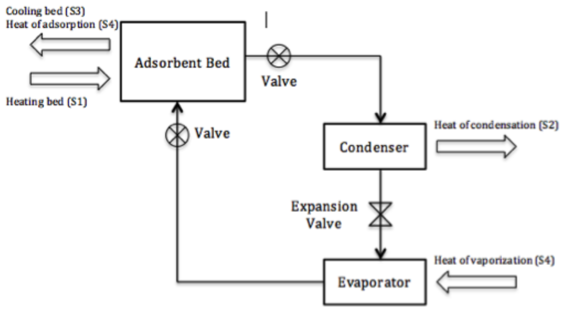


Figure 15: Operating principle of the adsorption heat pump

Adsorption heat pumps are based on solid sorption processes (adsorption is a chemical process where vapor molecules adhere to a solid surface) and run a thermodynamic cycle where heat is released at an intermediate temperature level. A high temperature heat source is used to produce the desired heat lifts. Adsorption heat pumps consist of four main components: an adsorber (containing an adsorbent bed), a condenser, an expansion valve and an evaporator; as well as the working fluid (adsorbate) cycling between them [20]. Thermal effects work to compress and circulate the adsorbate between the components. A simplified representation of the cycle is seen in figure 15.

A complete system cycle is based on two distinct thermal processes: adsorption and desorption of the adsorbate in the adsorbent. Four steps (S.1-S.4) are used to describe the cycle.

S.1 Heat is added to the adsorbent bed, isolated from the condenser and evaporator by two closed valves. No desorption takes place during this temperature increase due as the valves are closed (isosteric heating).

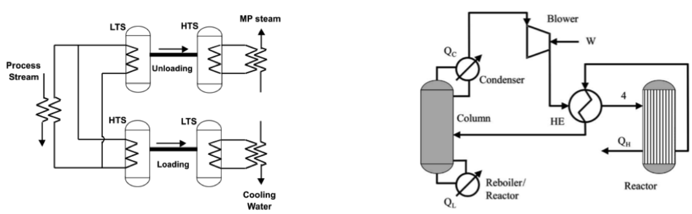
S.2 Heating is continued, and the valve closing off the condenser is opened. This initiates the desorption process, and vaporized adsorbate enters the condenser to release useful heat. The absorbate reaches its maximum temperature level in the cycle.

S.3 The valve is closed again, and the adsorbent bed is cooled to the evaporator's temperature and pressure.

S.4 Vaporization of the adsorbate takes place in the evaporator, and the valve separating it from the adsorbent bed is opened. Vaporized adsorbate enters the bed to be adsorbed, creating more useful heat.

Adsorption of vaporized adsorbate release heat from the adsorbent bed. The subsequent heating process elevates recovered waste heat from a low to an intermediate temperature level, using a high temperature heat source. As desorption begins (under continued heat transfer from the high temperature source), the adsorbate reaches its highest temperature, to condense against the designated thermal reservoirs in the condenser. This summarizes step 1 and 2 (which hold the process' main heat requirements), and implies that heat is taken from a high source (as well as at the low evaporator temperature), and delivered to the heat sink at an intermediate temperature level. Heat transfer is typically assumed to be isothermal, as a adsorbate is a pure fluid. Desorption processes are also typically isothermal.

Chemical Heat Pumps



(a) Inorganic chemical heat pump

(b) Organic chemical heat pump

Figure 16: Operating principles of the chemical heat pumps

Chemical heat pumps use reversible chemical processes to generate and/or store heat, by transforming thermal energy. There are several ways of chemically transforming thermal energy to useful heat, and the working temperatures range from -50°C to more than 1000°C with temperature lifts in excess of 100°C.[19, 30] Chemical heat pump are capable of operation as heat amplifiers (deliver heat at an intermediate temperature level) and as heat transformers (where heat is delivered at high temperature). They are also broadly classified as either inorganic or organic systems.

Inorganic systems typically involve solid-gas, solid-liquid and liquid-liquid reactions [4], and among the most common systems are the aforementioned absorption and adsorption heat pumps. The absorption heat pump has already introduced how reversible chemical reactions (absorption/desorption) are used to produce heat. Adsorption heat pumps illustrated the use of solids in chemical heat production. Figure 16b illustrate a third system, using two salts (a low temperature (LTS) and a high temperature salt (HTS)) to produce useful heat. Heat is released during absorption in the HTS, required in desorption in the LTS. As the salts eventually become saturated with gas/liquid, regeneration is required in to construct a thermodynamic cycle. Regeneration commonly requires high temperature heat, and the system accordingly operates as a heat amplifying system.

Organic systems include hydrocarbons or hydrocarbon derivatives, and take form as liquid-liquid, liquid-gas or gas-gas reactions. Chemical balance (A) generalizes this reaction, where generation of substance C takes place at one temperature level, and regeneration of substances A and B at another.[4]

(A) $\mathrm{A} \longleftrightarrow \mathrm{B} + \mathrm{C}$

At least one endothermic and one exothermic reactor are required in the heat pump systems. Figure 16a [4] shows a process flow diagram for an organic chemical heat pump. Heat (Q_L) is taken isothermally from an excess heat source, and used to decompose molecule A into B and C by an endothermic reaction at low temperature. The two components are separated from any residual component A left in the distillation column by heat addition, and directed to a reactor: A high temperature exothermic reaction takes place to regenerate component A from B and C, and molecule A is returned to the low temperature column to complete the cycle. Useful heat is thereby released isothermally to a designated heat sink.

2.3.3 Generalizations

As mentioned in the section considering high temperature industrial application, it is difficult to generalize the selection of a heat pump system because of the large variety between industrial processes, and due to the potential for customazing the components of a heat pump system. Nevertheless, some guidelines have been provided to create a starting point for further assessments [6].

Single-Stage Closed-Cycle Mechanical Heat Pumps and Open-Cycle Thermo-compression Heat Pumps are eliminated based on operational capability, unless the required temperature lift is lower than $\sim 40°C$.

Open-Cycle Mechanical Vapor Compression Heat Pumps are disregarded unless the heat source and sink is available as a low-pressure steam and a higher-pressure steam header, respectively.

Another notable operational restriction, which cannot be overlooked in this thesis, is the required high temperature heat source necessary in adsorption and several other chemical heat pump systems (including type I absorption heat pumps). This implies that useful heat is delivered at an intermediate temperature level [20]. The reason for implementing a high temperature heat pump in industrial applications often is related to the lack of a high temperature heat source, rendering these systems unfeasible solutions.

An overview of the introduced heat pump systems suitable for high temperature industrial applications is presented in table 2, alongside a comment on their operational capabilities.

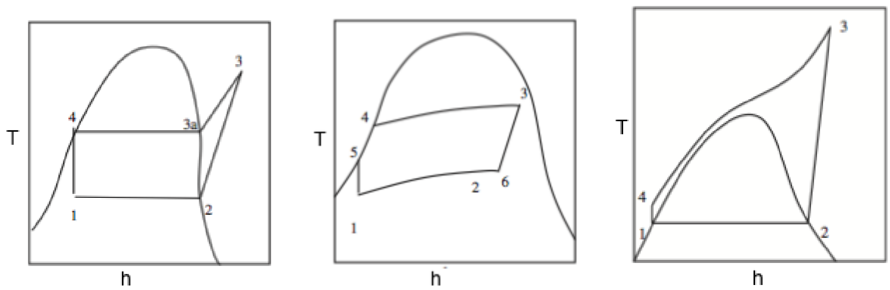
Further investigations are conducted to nominate a favorable technology applicable under variable operating conditions of the identified industrial scenarios. The appointed working fluid(s) of each system requires detailed knowledge of the industrial processes at hand, as a myriad of solutions are available on market. High temperature systems are, additionally, often prone to extreme pressures during operation, imposing severe strain on the heat pump system unless each component is capable of operating under such conditions. Selection of the most promising working fluids, as well as system components, for each scenario is therefore a prominent aspect of cost-effective implementation, and key to successfully reach the goals set forth in this thesis.

Heat Pump Design	Comment
Open-Cycle Mechanical Vapor Compression	Limited working fluids. Only operable with low pressure steam as heat source and high pressure steam as heat sink.
Closed-Cycle Singlestage Mechanical Compression	Incapable of high temperature lifts (>40°C)
Closed-Cycle Multistage Mechanical Compression	Cascade (with 2 or more compressors) or compound (with more than two system heat exchangers) structure.
Hybrid	A combined mechanical compression and absorption heat pump.
Open-Cycle Thermocompression	Limited working fluids. A motive stream mixes with excess steam, and is compressed to vaporize. No moving parts.
Absorption Type I (Heat Amplifier)	Require a high temperature heat source and release useful heat at an intermediate temperature level.
Absorption Type II (Heat Transformer)	Same operating principle as type I, but solution pumps produce a pressure lift elevating the heat sink temperature above the heat source temperature.
Adsorption	Require a high temperature heat source and release useful heat at an intermediate temperature level.
Chemical; Organic and Inorganic	Run as a continuous or discontinuous heat generator, depending on its system configuration. Operable as both heat amplifiers and heat transformer.

Table 2: Guidelines to heat pump selection

2.4 Working Fluids in Industrial Heat Pump Systems

Heat pump working fluids, refrigerants, have a profound effect on system operability and its performance parameters, and a multitude of different chemical compositions have been developed and applied with various degrees of success. Pure fluids and mixtures are both applicable, chosen to maximize the heat transfer efficiency to the heat pump's thermal reservoirs. Sketches of three typical thermodynamic cycles are shown in figure 17a [24], illustrating the temperature and enthalpy-change undergone by the utilized heat pump working fluids.



(a) Subcritical vapor compression

(b) Compression-resorption cycles

(c) Transcritical vapor compression

Figure 17: Temperature-enthalpy diagrams of typical heat pump systems

2.4.1 Pure Working Fluids

Two thermodynamic principles describe the operational basis of pure working fluids; subcritical and transcritical heat transfer.[51] Every fluid has a critical temperature and pressure, above which it must exist in gaseous form. This critical temperature is located at the top point of the bell curves in 17. A gas-liquid mixture develops, as temperature is decreased to enter the expanding bell-curves, with beneficial heat transfer effects attributed to fluid's heat of transformation. Heat release in a heat pump's condenser takes place at temperatures above or below this value, following isobaric pressure lines. Figure 17a and 17c illustrate these cooling processes, from step 3 to 4 in the sketches.

Subcritical thermodynamic cycles operate below the critical temperature, and release heat by condensing the working fluid. A typical system include a gas cooler where superheated vapor release heat with a temperature glide (from step 3 to 3a in 17a), followed by a shell-and-tube heat exchanger where the working fluid condenses in an isothermal process (from step 3a to 4, generally considered the main stage of condensation). A sub-cooler is sometimes used to decrease the fluid temperature below saturated liquid levels.

These cycles are normally considered isothermal, as superheating the gas requires high pressure outputs from the compressors. To minimize compressor costs (which scale with output pressure), superheating processes are limited. However, heat release from the gas cooler should be investigated at gliding temperatures initially, and the effects included if the heat quantities are comparable to those released by condensation in the main condenser stage.

When the condenser temperature is exceeds the working fluid's critical temperature, heat is released with a continuous temperature decrease. This is known as transcritical condensation 17c. Gas coolers are required to release heat to the heat pump's designated heat sink.

2.4.2 Working Fluid Mixtures

Working fluid mixtures have grown to represent an important heat pump component, and is a blend of two or more pure working fluids.[52] Fluid mixtures are often customized to particular process requirements, most often by adjusting the fluid mixture ratios. As pure working fluids, blends are also capable of operating with temperature glides in the thermodynamic cycle heat exchangers.

Depending on their heat exchange characteristics, blends are coined as zeotropic, azeotropic or near-azeotropic. Zeotropic mixtures evaporate and condense with gliding temperatures, whereas heat exchange in azeotropic working fluids is isothermal processes. Near-azeotropic mixtures are often assumed to operate isothermally during performance evaluations, unless its temperature glides matches heat sink/source temperature glides to great extent. The temperature glides are created by varying liquid/vapor concentrations in the heat exchangers, creating by the absorption/desorption processes, and give the fluid mixture a boiling trajectory. Note that the main process heat exchangers, utilizing the working fluid mixture effects, are called absorbers (release useful energy to the heat sink) and desorbers (extract excess energy from the heat source) rather than condensers and evaporators (see section 2.3 on heat pump working principles).

Figure 17b illustrates a compression-resorption cycle, using a near-azeotropic working fluid. Its cycle is fully enclosed by the working fluids bell-curve, indicating operation with a liquid-vapor mixture throughout the heat pump system. If the compressors are capable of wet compression, the mixture is elevated to higher energy levels by the same component. It is, however, more common to separate the mixture working fluids exiting the desorber, to compress saturated vapor and pump the liquid component to the high pressure absorper.

2.4.3 Applicable Working Fluids

Traditional working fluids in early heat pump systems consisted of chlorofluorcarbon (CFC) mixtures. Ever increasing environmental awareness and emission restrictions have, however, restricted their use as the effects of high chlorine contents and CFC's chemical stability have on the global environment has been uncovered. CFCs are known to deplete the ozone layer as well as being a greenhouse gas, and their use in as heat pump working fluids is prohibited in system development.[52] Two factors are introduced to evaluate a working fluid's environmental effects; its ozone depletion potential (ODP) and its global warming potential (GWP). The fluids should also offer the possibility of compact and inexpensive heat pump design, which is greatly affected by the fluid's compression ratios and heat transfer effects.

Due to the beneficial cost and heat transfer effects of CFCs, other working fluids were often developed with similar molecular structures. Hydrochlorofluorocarbons (HCFCs) are used as a heat pump working fluid, and operate with fractional ODPs and GWPs compared to CFC systems. They are nevertheless scheduled to phase out of use alongside the CFCs in industrialized countries by 2020, and entirely by 2040, as more environmentally friendly systems are available. Another fluid, hydrofluorocarbons (HFCs), do not contain chlorine and have zero ODP. Their GWP is, however, still high and special care must be taken when utilized in industrial systems. Fluorinated fluids are in general avoided to great extent in current systems due to their GWPs, and not implemented unless it is strictly necessary.[53]

New heat pump developments have focused on implementing natural working fluids (molecular structures existing in the biosphere), to eliminate and/or minimize system ODP and GWP to near-zero levels.[53] Natural working fluids and application in the industrial scale heat pumps are therefore investigated, both as pure working fluids and as components in mixture fluids. The five most promising natural working fluids are currently:

- Carbon dioxide, CO_2
- Ammonia, NH_3
- Water, H_2O
- Hydrocarbons, short-chained HC-molecules
- Air

A brief introduction to each follows suit. [51, 52, 53]

Carbon Dioxide (CO_2)
CO_2 is a non-toxic, non-flammable refrigerant without major technical barriers restricting implementation in smaller industrial heat pump systems (capacities around 100 kW). It is commonly coupled with the mechanical compression heat pumps, and operates with a transcritical process during heat release. Heat is supplied by the traditional phase change through an evaporator. Potential mediums in the thermal reservoirs rarely restrict heat pump application, and air source/sink systems are in operation.

Application of CO_2 as a working fluid has grown fastest of all systems in the small-scale heat pump segment, and units delivering heat capacities of 100 kW are readily available in the current market.[53] Larger systems have, however, taken longer to develop due to high pressure requirements on the heat pump's high temperature side (maximum system pressures exceeding 100 bar are not uncommon). This severely strains the applicability of market available compressors.

CO_2 has a relatively low critical temperature of 31.1°C [51], a notable feature during heat pump application. For one, the evaporator temperature cannot exceed this temperature if traditional evaporators are utilized. Decreasing CO_2's evaporator inlet temperature further increases the supplied heat quantities without superheating (heating a completely vaporized fluid) the CO_2. This is beneficial to avoid large superheating stages, which causes temperature glides and potentially pinch point problematics with the heat source. Secondly, condenser temperatures above 31.1°C release heat with a temperature glide, and the liquid phase does not form until after the throttling stage.

Ammonia (NH_3)
NH_3 is both flammable and toxic (and highly corrosive to copper materials, which should be avoided in the system components), but its thermodynamic properties are excellently suited for (high temperature) heat pump application. It is utilized extensively in both small and large-scale industrial systems, but requires special attention in design as well as operation. As CO_2, NH_3 is most commonly utilized in mechanical heat pumps, but it is also proposed for implementation in thermally driven heat pumps. (Specifically in absorption heat transformers, as the volatile component. The system proposes another natural refrigerant, water, as the system's working fluid.[54] Salt absorption of NH_3 is also an explored alternative. [64])

Ammonia's critical temperature is 132.4°C, which makes it applicable in high temperature operations. Waste heat is recoverable at relatively high temperatures, and latent heat can be released in relatively large quantities above 100°C. Heat pump developments propose ammonia as a pure working fluid, as well as a component in working fluid mixtures. The hybrid heat pump, operating with a ammonia-water mixture, is of particular interest among researched heat pump systems, due to its potential in high temperature applications and the favorable absorption/desorption-effects in heat transfer.

Water (H_2O)

H_2O is neither flammable nor toxic, and an excellent working fluid for high-temperature industrial heat pumps due to its favorable thermodynamic properties (a boiling point of 100°C at atmospheric pressure, a high heat capacity and a critical temperature at $T_C = 374.1$°C [61]). Water vapor has mainly been used as a working fluid in open and semi-open MVR systems, but closed-cycle mechanical compression heat pumps have also used water as their working fluid. It is also used in thermal compression systems, employed as both the volatile component (combined with salt sorption processes), and as the working fluid (ref. the aforementioned ammonia-water system).

Producing temperature elevations between the heat pump's cold and hot reservoirs are, however, quite challenging in mechanically driven systems. This is mainly due to the imposed strain on the system compressor(s). Water vapor has a low density (decreasing its volumetric heat capacity, despite the high latent energy), which requires a large swept volume to increase the pressure. And although inlet and outlet pressures are relatively low around the heat pump compression stage, the pressure ratio across it may be extreme. This increases both size and cost of the heat pump's compressor(s), or a series of compression stages, if it is to remain operable with H_2O as its working fluid.

Hydrocarbon (HC)

Hydrocarbon working fluids are based on the short-chained HC molecules, which have favorable thermodynamic properties for high temperature application in heat pump systems. Butane and isobutane are particularly interesting, with critical temperatures of 136°C and 151°C, respectively. This enables high operational temperatures with the traditional mechanical compression systems. System pressures are tolerable at these temperatures, rendering the heat pump components readily available.

However, the chosen components must be designed and implemented with care, due to the volatility and flammability of smaller hydrocarbon molecules. Hazardous situations are often prevented by designing machine rooms, isolating the heat pump from other processes, people, etc. in case of system leaks. Implementation of hydrocarbon heat pump systems must therefore be considered carefully, and research of these working fluids are scarce compared to the other natural refrigerants.

Air
Air is not considered for application in the typical thermodynamic cycles, where the working fluid is vaporized and condensed to release useful heat. This restricts application of air in high temperature heat pumps, as the pressure levels on the hot and cold sides are difficult to manipulate. Additionally, the relatively low latent energy in air streams does not promote heat transfer effects with the hot and cold reservoirs.

Other Mediums There is an abundance of other working fluids available on market, and the thermally driven heat pumps use a variety of other refrigerants, both gases, liquids and solids, to complete their thermodynamic cycles. A complete introduction these is omitted from this section, as their performance depend on system integration and design (i.e. chosen components, fluid pairs, etc.).The most prominent working fluids are, however, introduced by name. Further descriptions are provided if it is chosen as a part of a potential heat pump solution presented in chapter 5.

Lithium bromide (LiBr) is a salt typically used in an aqueous solution alongside water in absorption heat transformers, cycled in the pump as a rich/lean solution of LiBr [55]. LiBr and magnesium chloride ($MgCl_2$, a salt) are also used in solid/vapor heat transformers, where ammonia act as the adsorbate and attach to the surface of these salts[56].

Isopropanol ($(CH_3)_2CHOH$) dehydrogenation (removal of hydrogen, (H_2)) into **acetone** ($(CH_3)_2CO$) is common in organic chemical heat pumps, a reversible chemical reaction aided by various catalysts.[4]

2.5 Components

The basic components of the heat pump systems suitable for industrial applications are, as introduced in section 2.3:

- Heat Exchangers.
- Mechanical Compressors.
- Solution Pumps.
- Expansion Valves/Throttling Valves/Restrictors.

A short introduction to these components are given, to avoid repetition of standard system components through the ensuing descriptions of investigated heat pump systems.

2.5.1 Heat Exchangers

Heat exchangers are used in all heat pumps, and transfer heat between the working fluid and the available thermal reservoirs. Mechanical compression heat pumps use heat exchangers for transfer with the process heat source and sink. Heat transformers use heat exchangers for the same purpose, in addition to service the heat supply driving the pumps.

Heat transfer is often assumed to be isobaric processes, and working fluids typically change phase through the heat exchangers. This takes advantage of a working fluid's latent heat, and renders the heat transfer isothermal. Condensation releases heat to the serviced medium at the heat sink, and evaporation requires heat supply from a heat source. However, heat is also transferred from fluids approaching the characteristics of pure gas and pure liquid (i.e. in cooling of superheated gases, transcritical compression systems and sub-cooling saturated liquids), where the working fluid's sensible heat is utilized. Heat exchangers must therefore be capable of gas cooling and liquid cooling as well. Two features are key to successful design, heat transfer area and fluid heat transfer effects.

Heat exchangers are typically chosen as one of two types: shell-and-tube heat exchangers and plate-and-fin heat exchangers. Both systems have prominent features and disadvantages in application, and operate as condensers as well as evaporators. The ensuing section gives a brief introduction to heat exchanger design [58].

Shell-and-tube use is problematic as the working fluid condenses/evaporates within tubes, which affect its heat transfer abilities. A gas-liquid mixture forms during heat exchange, and complicates control of the heat transfer effects and optimize performance. Superheating the evaporating working fluid (to ensure dry compression) is also problematic, and require some margin in operation. Pressure losses in fluid transport through the thin, long tubes may also be significant, and affect overall system performance. The dotted green line in figure 23 illustrates the resulting loss of temperature driving forces in a condenser system, and larger heat exchanger area is necessary to maintain heat transfer effects. This increases system cost. There are several solutions to these challenges [58], but they are not treated in this thesis as pressure losses are neglected in the system heat exchangers to simplify the calculation tool (refer to the assumptions in section 3.2.1).

Plate-and-fin systems have a compact design and are easily adapted to support the heat pump's designed performance. Large surface areas between the two sides of the heat exchanger ensure low temperature driving force requirements. They do, however, experience large pressure losses, which affect their overall efficiency during heat transfer. Superheating is, again, problematic to control, and require a certain margin in operation.

Both systems are in theory applicable in heat transfer with air flow as the thermal reservoirs. Shell-and-tube heat exchangers have, however, been at the center of most research. It is employed in the systems presented in the ensuing sections, and therefore given a further introduction below. Plate-and-fin systems are applicable, and their performance should be investigated further. However, as the focus of this thesis is on evaluation of overall system design and applicability rather than individual components, the known and tested shell-and-tube heat exchangers are described as the sole alternative.

The Shell-and-Tube Heat Exchanger:
Heat pumps generally use shell-and-tube heat exchangers, designed as illustrated by figure 18 [58]. They typically operate with counter-current flow, to facilitate heat transfer with the aforementioned favorable temperature glides. Another common design solution employs tube-side inflow and outflow on only one side of the heat exchanger. But as it prevents counter-current flow, these are often dismissed from heat pump design.

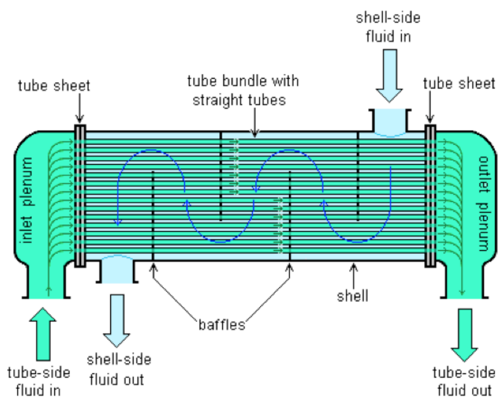


Figure 18: Typical shell-and-tube heat exchanger design

The heat exchangers are designed with either horizontal or vertical heat transfer, typically dependent on the employed working fluids. Pure fluids exchange heat in both systems, but fluid mixtures (typically gas/liquid, as in absorption/desorption heat transfer) generally use a vertical arrangement. Absorption heat exchange use a feeder system, which lets the liquid run along the tube walls, driven by gravity, and the gas is blow upwards in the opposite direction. Varying concentrations drive the heat transfer, and require a minimum amount of external energy to drive the two process flows.

Both the identified scenarios evaluate heat release to air streams, which has a low heat transfer coefficient and poor heat transfer effects. The working fluid is accordingly sent tube-side through the heat exchangers, whereas the air is sent shell-side. This ensures that air runs the long way through the heat exchangers, elongated by baffles. Air crosses path with each tube multiple times to maximize the heat supply, beneficial to reduce the heat transfer lengths, and decrease system cost (which is strongly dependent on the heat exchanger's size).

Liquid Coolers:
The shell-and-tube heat exchangers are also capable of sub-cooling the liquid exiting heat pump condensers, but should be implemented as a separate unit. Working fluids still flow tube-side in the heat exchanger, and experience a temperature drop through the heat exchanger. This simplifies the pinch problematics of heat exchanger design, and increases the overall heat transfer efficiency.

Gas Coolers:
Gas coolers have similar design as the shell-and-tube heat exchangers, and the heat pump working fluid flows tube-side through the heat exchanger. No phase change occurs during heat transfer, these heat exchangers utilize the working fluid's sensible heat. A temperature glide form during heat transfer, and its range depend both on the working fluid and the released heat load. Temperatures are typically much higher than in the heat exchangers utilizing a working fluid's latent heat, and must be considered in system design. Tubes are typically upgraded with fins and interconnected tube serpentines, which increases the overall heat transfer area and sustain high heat transfer effects in what is often gas-gas heat exchange.

An Industrial Manufacturer:
Alfa Laval manufactures heat exchangers, applicable with a wide range of working fluids introduced in the preceeding sections, and introduce a multitude of systems designed for air heating and cooling applications. Other service mediums (gases, liquids and two-phase mixtures) are also applicable. Volumetric capacities exceed the requirements in EWOS' and TINE's production processes, and system components are easily upgraded to satisfy the temperature and pressure levels of demanding industrial use (i.e. by use of acid-proof steel tubes and aluminum fins etc.).

Further information on system design, operational ranges, etc. is found in Alfa Laval's online product catalogue [65]. Traditional systems as well as innovative solutions (such as their plate-and-shell heat exchanger) are presented as readily implementable in industrial use.

2.5.2 Compressor Systems

Compressor systems are used to increase the pressure (and accordingly the temperature) of the working fluid vapor exiting the evaporation stage in a heat pump. Industrial application requires relatively large compressor systems, with high initial costs, due to increased heat load requirements accompanied by larger working fluid flow rates. Open systems (separate compressor and engine) are therefore in preference of closed systems (compressor and engine in the same shell) to simplify maintenance work. Compression is either dry (pure gas) or wet (liquid entrained in the inlet gas). Dry compression is preferable; as it minimizes wear and tear during compression, reduce the system's operational and maintenance costs. Compressors are generally oil-lubricated (to cool, seal and/or lubricate the internal parts), as most working fluids fail to self-lubricate the system during compression. Oil-free systems are available, but costly compared to oil-lubricated compressors. High temperatures tend to affect the pressurized oil negatively, and diminish its lubricating effects. The pressurized oil consequently restricts a compressor's operational temperature range, and must be chosen to complement system pressures and temperature ranges. Oil recovery systems are also a requirement, to reduce operational costs.[18]

Several compressor systems are applicable and should be evaluated in design of industrial heat pumps. Compressors are broadly classified by their operating principle, based on dynamic compression or positive displacement of the compressed medium. Both categories have include several different subsystems, all applicable in industrial processes.[11] Most available systems (and the most prominent in industrial application) are evaluated by the data collected from compressor manufacturers, presented graphically by figure 19[11].

Dynamic Compression:
Dynamic compression systems convert a working fluid's kinetic energy into a pressure increase. Gas enters the compressor, and is accelerated by the impeller blades. A subsequent deceleration through a diffusor increases the working fluid's pressure by lowering its velocity.

Dynamic compressors operate with axial, mixed or centrifugal flow. All present an affinity to high volumetric flow rate compression, but cannot produce large

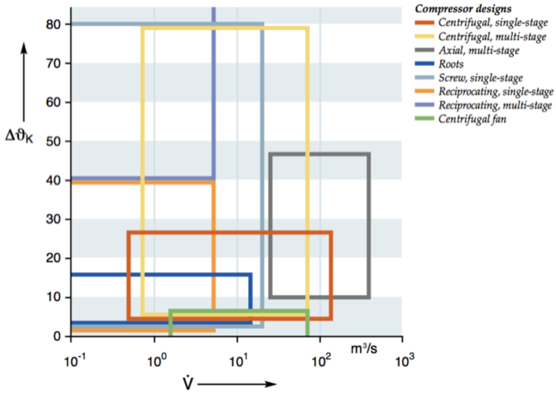
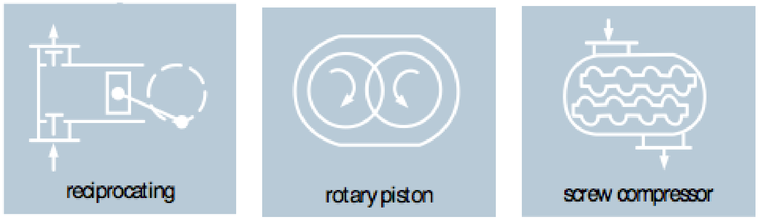


Figure 19: Functional ranges of compressor systems – $\Delta \upsilon_K$ is the increased condensation temperature of water vapor at an initial state of 1 bar, 100°C, $\dot{V}$ the vapor mass flow rate

pressure ratios.[51] This feature reduces their application in high temperature industrial processes, where an often significant temperature elevation (and accordingly large pressure ratios across the compression stage(s)) is the most pressing design condition. This is illustrated by figure 19, where most dynamical compression heat pumps are situated in the lower right corner. Multistage centrifugal compressors are the exception, but the size and cost of these systems escalate quickly as their size and casing require upgrades with increasing system pressures.

Positive Displacement Compression:
Positive displacement systems, and particularly the screw compressors, present wide pressure ratios and volumetric flow rates exceeding 10 m^3/s in figure 19. Multistage reciprocating compressors are also capable of large temperature increases, at slightly smaller volumetric flow rates.

Positive displacement compressors increase working fluid pressure by confinement and reduction of its gas volume. Pressure ratios across the compression stages are accordingly determined by the compressors physical design, rather than by fluid velocity. Current research on compression systems requiring high-pressure ratios typically accordingly focus on reciprocating and screw compressors. Their operational principles are sketched and illustrated by figure 20a and figure 20b,c, respectively.[11]



(a) Reciprocating compressor (b) Screw compressor: Front view (c) Screw compressor: Side view

Figure 20: Applicable compressor systems in high temperature application

Reciprocating compressors are similar to internal combustion engines, and require valves and sealing to operate efficiently. Steam heating of sealing faces, the cylinder shell and the gland pocket is often used to avoid thermal stresses and system failure.

Screw compressors are therefore chosen over dynamical compression and reciprocating compressors, supported by three beneficial characteristics. Firstly, screw compressors feature equal performance over a wide range of heat outputs. As the required heat output in industrial scenarios varies greatly with the heat pump's achieved high temperature level, this feature retains system applicability regardless of an investigated heat sink's temperature span. Partial loads are applicable, without compromising operability significantly. Secondly, screw compressors are operable with the steam-water as internal lubrication, whereas oil is required by other compressor systems used in heat pumps (typically centrifugal compressors, but also axial, reciprocating and other configurations). They are also capable of both dry and wet compression, beneficial if there is superheated water con- taminating the steam exiting the evaporator. Thirdly, the uninstalled cost of screw compressors is significantly lower than that of centrifugal systems at low shaft power outputs (less than 1 MW) [24]. The last statement is based on the results of a research paper, where a propose power output/cost-relations are based on sampled compressors from four manufacturers, and are accurate enough for the purposes of this selection.

Screw Compressors:
Screw compressors raise gas pressure by trapping a fixed gas volume on its suction side, and progressively decreasing the volume through the compressor.[50] Their inlet/outlet gas volume ratio is fixed or adjustable, a configuration which affects system performance as well as cost. Fixed volume ratio compres- sors offer energy efficient systems if their volume ratio is chosen properly, as head pressure floats to achieve efficient system operation. Variable volume ratio machines, on the other

hand, increase the energy performance and the systems' operational flexibility, at increased cost. Both capital and maintenance costs rise, and the system durability decreases compared to the fixed volume counterparts. The increased maintenance costs and diminished reliability are attributed to the additional components needed for volume ratio control.

Screw compressors do not require internal lubrication by pressurized oil. Steam-water compression can use the working fluid for lubrication, and are operable with mechanically loaded pressure differences up to 12 bar. Screw compressors handle 20 bar when oil lubricated. Current systems limit the output gas temperature to 250°C, which restricts system applicability in high temperature industrial application. Screw compressors are also capable of compressing practically all gases.[49]

An Industrial Manufacturer:
GEA Grasso manufactures compressors for refrigeration purposes, and focus on reciprocating and screw compressor systems. Their vast compressor series are designed to operate with most the common working fluids (ammonia (NH_3, carbon dioxide (CO_2), water vapor ($H_2O_{(g)}$), etc.) and work capacities ranging from less than 100 kW to several MW. All product information is available in their online product catalogue [2].

Screw compressors developed by GEA Grasso cover a vast range of operational swept volumes (ranging from 60 m^3/hr to more than 8500 m^3/hr), and are designed to withstand high-pressure operation. Screw compressors in the GEA Grasso SH, GEA Grasso MC and GEA Grasso LT series are designed with standard operational pressures peaking at 28 bar, increased to 63 bar upon request. The screw compressor AC series are designed specifically for high pressure heat pump application with natural refrigerants, and are operable with a designed peak pressure of 130 bar. Its maximum swept volume is 155m^3/hr.

2.5.3 Solution Pumps and Expansion Valves

Solution Pumps:
Solution pumps are operable as positive displacement pumps and dynamical systems (typically centrifugal operation), like the compression systems discussed in the preceedings section. The pumps are used to increase the pressure (and temperature) of liquid working fluids in heat pump application. They are, accordingly, found in system utilizing gas-liquid reactions in heat transfer, which requires both liquid and gas at high temperature levels. However, neither temperature nor pressure levels are expected to restrict application of solution pumps in the heat recovery systems discussed in this thesis. Current pump technologies are capable of compressing pure liquids, and liquid solutions with entrained solids and/or gas

bubbles.

Pumps are estimated to use 20% of the world's electrical power [31], which makes energy efficiency and sustainability key features of heat pump development. Continuous improvements increase the operational ranges of industrial scale pumps, with regards, to temperature, pressure, liquid solutions, etc. Finding market available pump technologies applicable in the introduced heat pump systems and industrial scenarios is accordingly restrictive in system design, and no further introductions are required. Among the prominent pump manufacturers are Dover's Pump Solutions Group (PSG®) and Grundfos, which webpages contain extensive information about pumping solutions and the thermodynamic restrictions of industrial applications.

Expansion Valves:
Expansion valves, throttling valves and restrictors are all components used to decrease the working fluid pressure after heat release to the designated heat sink. Its mechanical configuration is greatly simplified compared to the other components in heat pump systems, as its sole purpose is to restrict the working fluid's flow rate as it enters the evaporation stage.[21] Decreasing the working fluid's pressure to the evaporator's pressure and temperature, facilitates heating of the working fluid by low grade heat from an available process waste stream. If a heat pump system operates with a fixed heat supply (in terms of quality and quantity), the thermodynamic effects produced by the expansion valve extend through the evaporator, and control superheating of the working fluid exiting the evaporator. Control over the expansion process is held internally or externally. The former refers to expansion pressure ratios based on measurements of the evaporator inlet pressure, the latter on measurements of the evaporator outlet pressure.

Expansion valves experience phase change from gas to liquid or liquid to gas, depending on the thermodynamic state of the working fluid exiting the heat exchanger (the former when gas exits after transcritical heat transfer, the latter when saturated liquid exits a condensing system). Ideal, isentropic expansion assume equal working fluid enthalpies at the valve's inlet and outlet, which implies that no energy is transferred to or from the component. This is, of course, not feasible in real applications, and some of the working fluid's energy is lost to its environment. However, it indicates that expansion valves are easily incorporated in heat pump systems, regardless of the heat pump's thermodynamic operational level. Thus, further descriptions of these components are omitted, as they are readily available in the open market.

3 Calculation Tool

The calculation tool is developed to evaluate the energy efficiency of the presented heat pump systems, within the confines of two industrial applications. Accordingly, the obtained results should differentiate the performance of each heat pump based on a set of generic theoretical aspects representing each industrial process.

There are two immediately identifiable sources of energy loss associated with heat pump systems; the first relates to the irreversibilites of heat transfer between the heat pump and the thermal reservoirs; the second to energy dissipation occuring as the required temperature lift is produced. A thermodynamic evaluation of the introduced heat pumps' performance, accounting for these two parameters, is performed to increase the accuracy of the developed calculation tool.

3.1 Coefficient of Performance (COP)

The *Coefficient of Performance* (COP) quantifies the performance of a heat pump operating between two or more thermal reservoirs, based solely on the energy input and output from a process. [21] The COP is defined as the ratio of useful heat release to the net work input required to run the heat pump, and its value is calculated with equation (1).

$$COP_{Heat} = \frac{Q_H}{W_{NET}} \tag{1}$$

Q_H is the heat release from the heat pump to its designated heat sink, and W_{NET} the work requirement elevating the temperature of the energy supplied by the colder heat source, Q_C, to useful levels. Accordingly, $W_{NET} = Q_H - Q_C$ under idealized operating conditions. Equation (1) is used as the vantage point for all COP development in this paper, but it should be simplified to omit the effects of working fluids and component specifications.

Heat transfer is known to take place isothermally (at constant temperature) or with a temperature glide through the heat exchangers. Useful heat is produced mechanically or thermally. These generic system configurations are likely to affect the heat pump performance, and are accordingly included in the simplifications of equation (1).

3.1.1 COP Isothermal Heat Transfer

The simplifications are initiated by an analysis of isothermal heat transfer in mechanical compression systems, and the basic thermodynamic cycle presented in section 2.3. As the aim of the ensuing sections is to simplify equation (1), but maintain its efficiency as a measure of heat pump performance, an understanding of the underlying thermodynamic effects of this process is sought out. This should facilitate an estimation of equation (1) based on a minimal amount of information, easily obtained from an industrial process.

This simplification is based on the power of reversible cycles (and specifically its simplest form, the Carnot Cycle), which states that the ratio of heat transfer between two reservoirs depends only on reservoir temperatures: $(\frac{Q_C}{Q_H})_{rev} = \psi(Q_C, Q_H)$. ($\psi(Q_C, Q_H) = \frac{T_C}{T_H}$ when either the Kelvin or the Rankine temperature scale is employed; T_C represent the cold reservoir temperature, T_H the hot.) As isothermal heat transfer at temperatures T_C and T_H is assumed, equation (1) transforms to equation (2), yielding the maximal theoretical performance of a heat pump system in the given environment.[21] The equation is often referred to as a system's Carnot efficiency, and depicts a heat pump operating without thermal driving forces in its heat exchangers, and without losses through its compression stage(s).

$$COP_{Iso} = \frac{T_H}{T_H - T_C} \tag{2}$$

Equation (2) is simple in application, but holds severe restrictions when the assessments are of real life applications. First of all, it is based on reversible thermodynamic cycles, implying that no energy is lost in the heat pump processes. In reality, this is far from achievable. Secondly, it only measures the maximum energy saving potential held by heat pump systems working between two reservoirs, and do not differentiate between the systems' individual performance. Accordingly, results obtained from equation (2) are not useful when the *concluding* remarks of the thesis are made, unless they are supported by further performance analyses.

However, it points to some important aspects of heat pump application in industrial processes. $COP_{Real} < COP_{Iso}$ for any heat pump system in real life applications, thus providing an upper limit for the energy saving potential held by such a system. A heat pump's COP is also inversely proportional to the temperature difference between the hot (T_H) and the cold (T_C) reservoir, rendering the inherent reluctance related to high temperature lift heat pumps in the industry (mentioned in previous sections) quite clear.

Such aspects, and the intrinsic simplicity of these equations, make the COP a suitable vantage point when developing the calculation tool utilized in this thesis' analyses. And consequently, as the introduced heat pump systems are based on several different working principles, an assortment of equations calculating COPs are necessary to distinguish the performance of the various operating principles, and to appoint those most suitable in the identified industrial processes. COPs for the remaining heat pump working principles are presented in the ensuing sections.

3.1.2 COP Heat Transfer with Temperature Glide

Section 2.3.1 introduced the three key factors to high temperature heat pump application. The first and second relates directly to a system's heat transfer effects: from a given heat source to the heat pump, and from the heat pump to the desired heat sink, respectively. Heat transfer is based on spontaneous mechanisms, where heat flows from the hotter to the colder reservoir in all exchangers. Basic energy losses are readily identified, and relate to the second law of thermodynamics and its effect on exergetic efficiency. It states that increased temperature differences in these heat exchangers inevitably increase heat transfer irreversibilities [22], effectively lowering the energetic efficiency of the entire heat pumps system.
An initial generalization distinguishes the energy efficiency of the introduced heat pump systems. All heat exchangers need a finite temperature difference to transfer energy between a hot and a cold reservoir, and the minimum difference required is generally referred to as the heat exchanger's pinch temperature. A heat pump's energetic efficiency is maximized when the average temperature difference in its heat exchangers approaches this pinch temperature.

Depending on the working fluid utilized in the heat pumps, a *temperature glide* is achievable in certain system heat exchangers. The concept was touched upon in the section on working fluids in chapter 2, and elaborated on in the ensuing paragraphs. Both mechanical compression heat pumps and thermally driven systems are capable of heat transfer with temperature glides, due to its dependability on system working fluid(s) rather than the driving force. The former is used as a basis in derivation of the mathematical relations required to include the effects in estimations of system COP, for three reasons. Basic mechanical compression heat pumps have a simple working principle, with only two thermal reservoirs (heat source/sink) and one mechanical driving force (the compressor system). The effects of temperature glides on heat pump performance are readily illustrated, by comparing the developed equation to equation (2). Long temperature glides are commonly utilized in high temperature mechanical compression systems, in transcritical operation and in the hybrid heat pumps.

A temperature glide refers to the temperature change obtained in heat exchangers during heat transfer processes. [17] It is created by transcritical cooling of a pure fluid, which releases its sensible heat (as described previously, in section 2.4.1), or by using two (or more) substances with different thermal characteristics as the working fluid (typically one more volatile than the other(s)). The volatile component is gradually absorbed into (releasing high temperature heat due to a high system pressure) and evaporated out of (requiring low temperature heat due to the low system pressure) the solution through the heat exchangers (condenser and evaporator respectively). These changes in concentration affect the thermal properties of the working fluid, and accordingly the system's temperature and heat transfer effects.

By opting for these temperature glides, heat exchange can take place with a lower average temperature difference between the heat pump and its adjoining reservoirs, closer to pinch values throughout the heat exchanger. The effect is illustrated by a Lorentz Cycle in figure 21 [17]. An isothermal heat transfer processes (based on the Carnot Cycle) is included as a comparative measure. Excess heat is supplied by a waste stream (in the "desorber" heat exchanger), and released as useful heat at higher temperature to a desired process stream (in the "absorber" heat exchanger). Released heat quantities equal the area under each graph, rendering figure 21 an apt illustration of the efficiency of temperature glides compared to isothermal heat transfer. As maximizing the heat exchanger performance may improve the overall efficiency of a heat pump system significantly, it is included in the overall system evaluation.

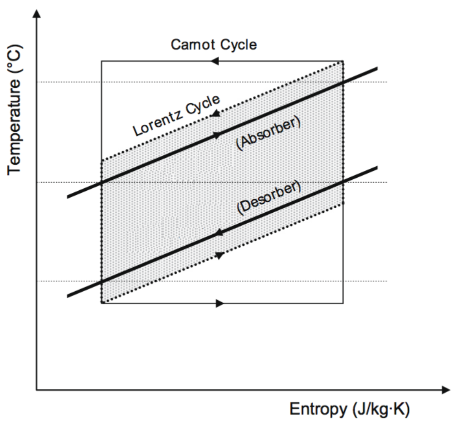


Figure 21: Isothermal (Carnot) and thermally gliding (Lorentz) heat exchange

These heat transfer effects influence the simplifications made to equation (1) when deriving COP_{Iso}, as heat transfer no longer take place under isothermal conditions. The reservoir temperature levels used in equation (2) are accordingly modified to estimate the average operational temperatures ($\overline{T}$) of each heat exchanger, using the following mathematical relation[23]:

$$\overline{T} = \frac{\Delta T_{X,g}}{ln(\frac{T_{X,hot}}{T_{X,cold}})}$$

.

$\Delta T_{X,g}$ is the temperature difference between the hot end ($T_{X,hot}$) and cold end ($T_{X,cold}$) of the heat exchanger, the available temperature glide. Equation (3), address the performance of an ideal Lorentz cycle, and employ said relationship in equation (2) to yield COP_{Glide}. [24]

$$COP_{Glide} = \frac{\overline{T}_H}{\overline{T}_H - \overline{T}_C}$$

$$= \frac{\Delta T_{H,g} \Big/ ln(\frac{T_H}{T_H - \Delta T_{H,g}})}{[\Delta T_{H,g} \Big/ ln(\frac{T_H}{T_H - \Delta T_{H,g}})] - [\Delta T_{C,g} \Big/ ln(\frac{T_C + \Delta T_{C,g}}{T_C})]} \tag{3}$$

T_H and T_C represent the highest heat sink and lowest heat source temperature, respectively, $\Delta T_{H,g}$ and $\Delta T_{C,g}$ the available temperature glides in the hot and cold heat exchanger.

It should be noted that the mathematical relationship giving the average temperature in the heat exchanger only applies when temperature glides are small, or when the driving forces in the exchangers are strong (a large temperature difference between the working fluid and the heat source/sink). Application in other systems may lead to significant errors.[25] This is attributed to the curvature of the condensing and evaporating temperature streams, as they depend on mixture concentration and temperature levels rather than being exponential functions of entropy. Preliminary iterations of the temperature profiles increase the accuracy of COP_{Glide}, but are not included in this calculation tool (as the effects of increased accuracy on the thesis' concluding remarks are too small to defend the required computational developments). As this is an initial evaluation of system performance, the accuracy of equation (3) suffices, if these limitations are kept in mind during application.

3.1.3 COP Thermal Heat Generation

Thermal heat lifts are generated by chemical reactions,and take several different forms depending on the chosen sorption processes. Heat generation is created through absorption (physical mixing of two substances), adsorption (adherence of a substance to the surface of another) or direct molecular bonding. Regardless of the basic operating principle, a characteristic chemical heat pump system includes two or more heat exchangers. Additional heat exchangers are often necessary to generate the thermal temperature lift, as the mechanical energy input created by compressors is eliminated. To calculate the theoretical performance (COP) of a heat pump with temperature elevations produced by thermal energy, certain modifications are required to equation (1). The characteristic change is a redefinition of COP to a ratio of useful heat output to system heat input.

The **absorption heat pump** is the first closed-cycle system presented in section 2.3.2 driven by a thermal heat source. And as it is among the most commonly adopted chemical heat pumps utilized in industrial applications, deriving its COP is a natural vantage point when dealing with the performance of heat pumps with a heat generated temperature lift. Heat exchange is assumed to be isothermal initially, and temperature glides are incorporated after system COPs are derived for all working principles. As emphasized in section 2.3.2, absorption heat pumps are classified as heat amplifiers (type I - releasing heat at an intermediate temperature level) or heat transformers (type II - releasing heat at a high temperature level). Other chemical heat pumps are assumed to operate with similar configurations, and a COP derived for absorption systems is thus applicable for most chemical heat pumps.

In type I adsorption heat pumps, a **heat amplifier**, useful heat is released to the process sink(s) from the absorber (Q_A) and the condenser (Q_C), at their respective temperature levels T_A and T_C. Accordingly, Q_H of equation (1) equals Q_A+Q_C. The work input (W_{NET}) produces thermal lift by a thermally generated pressure increase, and is supplied by an external heat source in the generator, Q_G at T_G. The heat to be upgraded is gained in the evaporator (Q_E at T_E), which serves as a low-grade heat source. Note that this derivation neglects the pump work required to circulate the working fluid, and increase its pressure to generator levels. It is a feasible assumption as the pumps consume negligible amounts of energy compared to the external heat input in the generator. When these relations are employed in equation (1), the COP_{Heat} of an absorption heat amplifier emerges. Although it is derived specifically for type I absorption heat pumps, equation (4) estimates the COP of any thermally driven system operating as a heat amplifier (heat release to intermediate temperatures, driven by a high temperature heat supply).

$$COP_{Amplifier} = \frac{Q_A + Q_C}{Q_G} \tag{4}$$

To develop a COP temperature relation for thermally driven heat pumps (as introduced for mechanical compression systems in the preceding sections), certain assumptions regarding the system configuration are necessary. The basic energy balance of a type I absorption heat pump is $Q_A + Q_C = Q_G + Q_E$. As mentioned in section 2.3.2, type I adsorption heat pumps are assumed to release useful heat to the industrial processes at one temperature level (the absorber and the condenser hold equal temperature: $T_A = T_C = T_I$).[24] T_I represent an intermediate temperature level. If the thermodynamic processes in the heat pump are assumed reversible, the net change of entropy through the cycle is zero. Entropy gain in the evaporator is balanced by a reduction of entropy in the condenser, and the same symmetry exists in the generator (gain) and absorber (reduction). The second law of thermodynamics enables basic temperature-entropy relations to combine, and present the equalities: $\frac{Q_E}{T_E} = \frac{Q_C}{T_C}$ and $\frac{Q_G}{T_G} = \frac{Q_A}{T_A}$. These identities are coupled with the aforementioned heat pump energy balance, and rearranged to solve for $\frac{Q_A+Q_C}{Q_G}$. A relation describing equation (4) as a function of heat exchanger temperatures alone is derived. [26] Equation (5) conveys this relation, and calculates the ideal theoretical performance of a type I absorption heat pump based solely on the operational temperatures of its heat exchangers.

$$COP_{AbsTI} = (\frac{T_G - T_E}{T_G})(\frac{T_A}{T_A - T_E}) \tag{5}$$

The absorber and condenser are known to hold an intermediate temperature level, T_I. Substituting the generator temperature with a generic high temperature indicator, $T_G = T_H$, and the evaporator with a generic low temperature indicator, $T_E = T_L$, a COP estimating the performance of any heat amplifier emerges.

$$COP_{Amplifier} = (\frac{T_H - T_L}{T_H})(\frac{T_I}{T_I - T_L}) \tag{6}$$

If heat release requires a temperature elevation from the system's heat source, all heat amplifying systems are unapplicable. A type II absorption heat pump, or another **heat transformer** system, must be employed. Consequently, equation (5) no longer provides a feasible performance estimation, and yet another performance indicator (COP_{Heat}) must be developed. [14] Useful heat is upgraded by a heat load supplied intermediate temperature level, and released to a designated heat sink by the absorber (Q_A at T_A). Due to its low operational temperature, heat release from the condenser cannot be utilized efficiently by industrial process streams. Heat supply in the evaporator (Q_E) and the generator (Q_G), commonly assumed to operate at equal temperatures ($T_E = T_G$), generates the thermal temperature lift. Accordingly, $W_{NET} = Q_E + Q_G$, to form the $COP_{HeatTransformer}$, also applicable for type II absorption heat pumps. Note that pump work, in heat trasnformers required to lift the working fluid to the absorber pressure level, yet again has been neglected. The underlying reasoning supporting this simplification has not changed.

$$COP_{Transformer} = \frac{Q_A}{Q_E + Q_G} \tag{7}$$

A temperature relation for equation (7) is derived with the same approach as for heat amplifiers. The system energy balance is equal to that of type I heat pumps: $Q_A + Q_C = Q_E + Q_G$. Combined with the entropy balance $\frac{Q_A}{T_A} + \frac{Q_C}{T_C} = \frac{Q_E}{T_E} + \frac{Q_G}{T_G}$, rearranged to yield $\frac{Q_A}{A_E + Q_G}$, equation (8) forms. It evaluates the performance of a type II absorption heat pump, based solely on the heat exchangers' operating temperatures. [14]

$$COP_{AbsTII} = (\frac{T_A}{T_A - T_C})(\frac{T_G - T_C}{T_G}) \tag{8}$$

If the respective heat exchanger temperatures are replaced by indicators of their temperature levels ($T_A = T_H$; $T_G = T_E = T_I$; $T_C = T_L$), a generic COP for heat transformer systems emerges.

$$COP_{Transformer} = (\frac{T_H}{T_H - T_L})(\frac{T_I - T_L}{T_I}) \tag{9}$$

Adsorption heat pumps fitted in industrial applications are generally assumed to operate between three temperature levels. [20] Heat is released at an intermediate temperature level, and adsorption heat pumps are therefore classified as heat amplifying systems. Useful heat is delivered in two stages; by adsorption of the circulated adsorbate in the adsorbent bed (Q_A at T_A), and by condensation of the desorbed absorbate in the condenser (Q_C at T_C). Fluid bed cooling is commonly considered to be a negligible process in overall heat release, or to release heat to the same sink as the adsorption process. T_A and T_C are typically close, to match the temperature of the heat loads released to the industrial process streams. Temperature lifts are produced by a high temperature heat supply (Q_D at T_D), desorbing the absorbate from the absorbent bed. The low temperature heat source (Q_E at T_E) is used to evaporate condensed absorbate, which is transported to and adsorbed in the adsorbent bed to complete a thermodynamic cycle. As the heat generation process operates between the thermal reservoirs know from heat amplifying systems, an identical approach is used to derive COP_{Heat} for an adsorption heat pump. Inserting heat exchanger temperatures into equation (6) ($T_D = T_H$; $T_A = T_C = T_I$; $T_E = T_L$) yields equation (10), the generic performance indicator of adsorption heat pumps.

$$COP_{Ads} = (\frac{T_D - T_E}{T_D})(\frac{T_A}{T_A - T_E}) \tag{10}$$

Despite deriving COPs for both absorption and adsorption heat pump systems, numerous other **chemical heat pump systems** exist. Some are based on operating principles where neither of the tools introduced in equations (2)-(10) provide adequate estimations of the heat pump performance. Regardless, a generic equation estimating their COP, derived from equation (1), should be provided.

Certain chemical heat pumps produce temperature elevations by direct chemical reactions (ref. the organic systems introduced in section 2.3.2) rather than by physical mechanisms (as the systems discussed in the preceding sections do). The basic working principle of the chemical heat pump is based on the reversibility of these chemical reactions, but the numerous molecular reactions and differentiated thermal characteristics among chemical substances complicate creation of a performance coefficient based on external temperature levels. However, if the chemical reactions of a heat pump system are known, an approximation to their ideal efficiency can be made. As chemical reactions facilitate operation characterized as heat amplifying or heat transforming systems, equation (6) and (9) estimate the theoretical efficiencies in the respective cases. The chemical heat pump must be assumed to operate between three temperature levels.[32] And as the carnot efficiency is the highest obtainable COP, regardless of a heat pump's working fluid and specific operating principle, the equations are derived by applying the same basic principles as presented above. Consequently, equation (11) estimates a chemical heat pump's performance.

$$COP_{Chem} = COP_{Amplifier} \vee COP_{Transformer} \tag{11}$$

If the thermally driven heat pumps, which performance is generalized by equations (6) and (9), operate with a temperature glide in either of the heat exchangers, average heat exchanger temperatures ($\overline{T}$) are required to estimate their COP. Approximating $\overline{T}$ with the mathematical relation introduced by the Lorentz cycle rewrites equations (6) and (9) in logarithmic terms. The resulting derivations are long and messy, and have been omitted due to a limited applicability in real systems. They are, however, included in the analytical investigations based on system COPs.

This concludes the development of calculation tools estimating the maximum theoretical performance of heat pump systems, based on the operational temperature levels of the system heat exchangers alone. The nature of these equations does not allow direct performance comparisons between the heat pumps based on different working principles. However, they indicate each system's performance sensitivity when the operating conditions changes, and thus present a useful tool in the thesis' analysis. The developed performance indicators are summarized in table 3, labeled as equation (2)-(11).

Coefficient of Performance of Different Working Principles	
Isothermal Heat Transfer (2) $COP_{Car} = \frac{T_H}{T_H - T_C}$ $T_H > T_C$	**Heat Exchange with Temperature Glide** (3) $COP_{Lor} = \frac{\Delta T_{H,g} \Big/ ln(\frac{T_H}{T_H - \Delta T_{H,g}})}{[\Delta T_{H,g} \Big/ ln(\frac{T_H}{T_H - \Delta T_{H,g}})] - [\Delta T_{C,g} \Big/ ln(\frac{T_C + \Delta T_{C,g}}{T_C})]}$ $T_H > T_C$
Heat Sink at Intermediate Temperature Level (6) $COP_{Amplifier} = (\frac{T_H - T_L}{T_H})(\frac{T_I}{T_I - T_L})$ $T_H > T_I > T_L$ (5) COP_{AbsTI}: $T_H = T_G$; $T_I = T_A = T_C$; $T_L = T_E$ (10) COP_{Ads}: $T_H = T_D$; $T_I = T_A$; $T_L = T_E$	**Heat Sink at High Temperature Level** (9) $COP_{Transformer} = (\frac{T_I - T_L}{T_I})(\frac{T_H}{T_H - T_L})$ $T_H > T_I > T_L$ (8) COP_{AbsTII}: $T_H = T_A$; $T_I = T_G = T_E$; $T_L = T_C$ (11) $COP_{Chem} = COP_{Amplifier} \vee COP_{Transformer}$ *Depending on the operating principle of its working fluid.*

Table 3: Working principle and heat pump COP

3.2 Energetic Efficiency

System COP measures the maximum theoretical performance of each heat pump system in a given environment, but a closer investigation of each heat pump's efficiency is necessary to compare the heat pumps' performance to currently utilized heating utilities. Several sources of efficiency losses in heat pumps are identifiable, and among them are [24]:

- Temperature driving forces.
- Compressor efficiency.
- Pressure drop.
- Heat losses and gains.
- Superheating.
- Throttling losses.
- Mismatch between process and heat pump fluids.

The last bullet point is closely related to the topics discussed in the section 2.3.1, where a basic performance generalization for different heat pump systems is developed. Temperature driving forces are considered in a system's heat exchangers, a topic also touched upon in section 2.3.1. For simplicity, all heat exchangers are usually assumed to operate with equal temperature driving forces. The remaining bullet points affect each heat pump system differently, depending both on their working principle and on the chosen working fluid. Accordingly, the accuracy of a calculation tool estimating the necessary work input is increased if the system working fluids are specified before the investigative calculation tool is selected.

When the system working fluids are identified, a thermodynamic analysis based on mass and energy balances is performed for each component of a heat pump. Typical assumptions made to develop a simplified mathematical model evaluating heat pump performance include [40]:

- Refrigerants and solutions are in steady state and thermodynamic equilibrium conditions at all states.
- Solutions at generator and absorber outlets, and refrigerants at the condenser and evaporator outlets, are all saturated.
- Heat loss/gain to the surroundings is neglected in all system components.
- Pressure loss in piping and heat exchangers is neglected.

- Change in kinetic and potential energy is neglected.
- Mechanical work consumed by pumps is considered negligible, as pressure lifts consume significantly less work when the working fluid's average specific volume increases.
- Intermediate heat sources are considered to operate at identical temperature levels.

These assumptions allow heat capacities of the main system components to be estimated based on the working fluid's change in enthalpy across them. Equation (13) and (14) form the basis for these calculations, derived from the energy balance in equation (12) [44]. Enthalpies (h) are obtained from thermodynamic tables when the working fluid is known. Subscripts i and e are used to label the inflow and outflow of working fluid in each component.

$$\frac{dE_{CV}}{dt} = \dot{Q}_{CV} - \dot{W}_{CV} + \sum_i \dot{m}_i (h_i + \frac{V_i^2}{2} + gz_i) - \sum_e \dot{m}_e (h_e + \frac{V_e^2}{2} + gz_e) \quad (12)$$

Equation (13) quantify the heat release/addition of any system heat exchangers, estimated by the system working fluid's carried energy in and out of the exchangers. As the working fluid exchange heat at the heat sink and source of a heat pump, its energy decreases and increases, respectively. $Q < 0$ represents heat release to a heat sink, $Q > 0$ heat supply by a heat source. Where the temperature lift is produced thermally, $Q > 0$ represents the thermal work input in the generator and evaporator.

$$\dot{Q} = \sum_e \dot{m}_e (h_e) - \sum_i \dot{m}_i (h_i) \quad (13)$$

Equation (14) represents the work input of typical heat pump systems using mechanical compression to generate the temperature lift. As work is added to the systems, $W < 0$.

$$\dot{W} = \sum_i \dot{m}_i (h_i) - \sum_e \dot{m}_e (h_e) \quad (14)$$

Equation (14) estimates the work requirements regardless of the utilized working fluid and heat pump's mechanical operating principle. It is therefore employed whether the pressure increase is generated by compressors (when the working fluid is gaseous) or pumps (when the working fluid is in liquid phase).

When heat exchange is based on endothermic ($Q_{EN} > 0$) and exothermic

($Q_{EX} < 0$) chemical reactions, the component's energy efficiency cannot be estimated by a circulated working fluid's enthalpy. The ideal efficiencies of these reactions are determined by assuming that the transferred heat quantities equal the enthalpies of transformation at the respective reaction temperature levels ($Q_{EN} = \Delta H_R(T_C)$; $Q_{EX} = \Delta H_R(T_H)$).[28] Heat loads from other heat sources or sinks, mechanical compressors or pumps, are estimated by their respective enthalpy changes across the thermodynamic stage at hand (evaporation, condensation, pressure increase, etc.). The energy change is estimated by equations (13) and (14).

$$Q_{EN} = \Delta H_R(T_C) \tag{15}$$

$$Q_{EX} = \Delta H_R(T_H) \tag{16}$$

Equation (13) and (14) are the main equations used for energy analyses in this paper. Other mathematical relations may be utilized for certain calculations, but are not introduced in this section as they are of less importance to the thermodynamic discussions assessing heat pump performance.

3.3 Thermodynamic Inefficiencies

Looking back to section 2.3.1, the third key factor of heat pump systems relates to the pressure increase produced between system heat exchangers, enabling heat transfer at different temperature levels (and essentially run the heat pump). The working principles described in section 2.3.2 introduce compressors, solution pumps, chemical reactions, interstage heat exchangers and combinations of the four as means of increasing the working fluid temperature, which split into a multitude of alternative technical configurations. Energy dissipation therefore originate from a variety of sources, with varying importance and impact on the overall performance of each heat pump system.

Certain assumptions simplify the energy efficiency calculations, and create valid performance indicators supporting reasonable comparison of the heat pump systems' energetic performance. Equation (13) (for thermally driven systems) and (14) (for mechanically driven systems) are at the center of these calculations, estimating the energy input producing the required temperature lift. The obtained results present the maximum performance of each system, and accordingly the minimum work requirements with the given system configurations (chosen working fluid, compressor setup (single stage, multistage, etc.).

Characteristic energy losses and system inefficiencies are, when applicable, introduced alongside the specific systems due to the diversity of losses in between the systems. It facilitates an estimation of each system's capability of heat exchange with the heat source and heat sink (ref. the aforementioned mismatch between the heat transfer effects on the heat pump's working fluid and the process medium), increasing the analyses' accuracy. This generates a generic tool providing results easily comparable to the current energy consumption, which clearly illustrates the potential benefits of heat pump application. Accordingly, the precision of the performance analyses holds a satisfactory level.

Efficiencies of certain standardized components are nevertheless introduced, to approach realistic indications of the component performance. Based on the data in appendix 3 and TINE's base case scenario, the boiler's efficiency can be estimated. Heating the air flow from 87°C to 185°C requires ~516 kW under idealized conditions (ref. eq. (13)), whereas TINE estimates their boiler to use ~700 kW for this operation. Assuming a boiler efficiency (η_B) of 0.75 is accordingly valid. This value is also applied in EWOS' processes, as their boiler's energy consumption is omitted from the obtained process data.

Previous studies have investigated compressor performance and temperature driving forces in heat transfer systems.[24] Compressors are assumed to operate with an isentropic efficiency of 60-80%, easily altered in calculations to illustrate its effect on overall system performance. Temperature driving forces are assumed to be 5-15% of a system's temperature lift (ΔT_{LIFT}), and the heat exchangers use shell-and-tube-heat exchangers typical in heat pump application.

- Heat exchanger temperature driving forces are equal in all heat transfer components.
- Heat exchanger temperature driving forces: $5 - 15\%$ of ΔT_{LIFT}.
- Isentropic compressor efficiency: $\eta_C = 0.6 - 0.8$.
- Boiler efficiency: $\eta_B = 0.75$.

Implementing the proposed heat pump systems will require further investigations to determine these performance parameters accurately, but the energetic efficiency estimated by these manipulated idealized conditions indicates the potential benefits of heat pump application in the selected scenarios.

4 High Temperature Scenarios and a Preliminary Heat Pump Selection

Heat pump application in industrial processes is a complex and costly task, and multiple parameters must be factored in to successfully utilize the process' heat recovery potential. Numerous different technological solutions produce the temperature elevation necessary to upgrade low quality excess heat to high temperature useful heat, but they must be designed to operate within the bounds of highly diverse processes. A preliminary selection nominates a set of feasible heat pump configurations applicable as heat recovery systems in two industrial processes, established in the ensuing sections. Descriptions of the currently available heat pump systems deemed most-likely-to-succeed are subsequently provided, to create a rigorous framework for the section's concluding performance analyses. The calculation tool developed in chapter 3 estimates the energetic efficiency of the specific heat pump systems, within the confines of heat recovery in EWOS' and TINE's production processes.

4.1 Industrial Scenarios

4.1.1 EWOS

Section 2.2 and table 1 introduce three potential heat sources, all applicable to support heat pump operations. Appendix 2 contains the energy consumption and heat release in each step of the dehumidification process, and is referred to when specific heat quantities are discussed.

First is the actual air flow used to dry pellets in the production line. It is introduced to the dehumidification process at 52°C, a suitable temperature when the aim is to release heat at temperatures ranging up to 110°C. However, EWOS already recover the heat extracted by their dehumidifier (HX1/2/3 in figure 4) and use it to reheat the same process stream (with HX4/5) before the boiler is introduced. These systems are common in industrial applications, and highly efficient as simple heat exchange (coupled in HX1/4) recover heat without external energy requirements. The heat pump (HX2/5) is in operation, and left alone to minimize overall system intrusion. Additionally, HX2 extracts ∼150 kW, a small amount compared to the boiler requirement of ∼380 kW elevating the air to 110°C. Heat pump work requirements would therefore significantly outweigh the heat source energy input. (A heat pump would release heat at temperatures ranging from 42.4°C to 110°C, necessitating ∼550 kW of heat input spanning more than 65°C - refer to appendix 2 for exact values.) Thus, this heat source is discarded for further analyses of high temperature heat pump application.

Second is a process waste stream, a saturated moist air flow (air and water vapor) of 100 000 m^3/hr, initially at 50°C. Saturation indicates that the air contains its maximum level of water vapor at the given temperature and pressure. Because water vapor carries more energy per kilogram than dry air (water has higher enthalpy at identical temperature levels), saturated air streams have beneficial heat transfer effects compared to drier air streams. Thus, waste stream I carries more energy than in the air stream described above, due to its higher initial water content. A flow rate thrice the size of the process air flow, further increases its appeal as a heat source. Cooling this stream condenses part of the water, as falling temperatures decrease the air's saturation level, and energy is released to its environment.[41] And as the air is a waste stream, its energy can be utilized to full extent (cooled to ambient conditions) without affecting other processes. A heat pump is therefore able to extract a suitable amount of heat, allowing individual designs of heat pump system with differing working principles.

Waste stream II is the last identified heat source, a waste air stream initially at 35°C and with a relative humidity of 40% (water vapor is entrained to 40% of its saturation level). It is, accordingly, a less energy intensive stream than waste stream I, available at a lower temperature. Its energy is utilized if the heat requirements exceed that released by waste stream I, when cooled from 50°C to 35°C. This increases the heat source's energy potential, decreasing the temperature glide in the heat exchanger to limit its strain on the heat pump (with regards to heat exchange and temperature lift).

Consequently, waste stream I is chosen as the heat source in the scenario based on EWOS' manufacturing process. If deemed necessary in the calculations, waste stream II serve as a potential energy source to increase the heat pump's heat input.

There are three heat sinks in EWOS' production line where heat pump application is considered. First mentioned is the dehumidified air stream used for drying of pellets. At a flow rate of 8.07 kg/s, air and water vapor (with a relative humidity of 10%) is heated from 63.3°C to 110°C. A net heat input of 383.8 kW is needed to elevate the temperature, currently delivered by a steam boiler. Due to the air stream's low relative humidity and moderate flow rate, waste stream I hold enough energy to create the temperature increase. Heat need only be lifted to a higher temperature to be useful.

The second potential heat sink is saturated steam generation at 10 bar, with a production rate of 10 tons/hr. This is an energy intensive operation, which requires a significant temperature lift if a heat pump (when coupled with the available heat sources) is to produce it. It strains the heat pump's components and working fluid severely, as the internal pressure requirements are extreme. Thus, it is not considered for heat pump implementation in this thesis.

High-pressure hot water production is currently driven by high pressure steam. As the hot water hold a temperature of 90°C, it is in the operational range of currently applied CO_2 heat pump systems, exemplified by the Japanese company Alfa Laval's solution [42]. Alfa Laval's system compress gaseous CO_2 to a super-critical pressure of 100 bar. The CO_2 holds approximately 100°C, and is capable of heating water from temperatures of 5°C and higher. Heat delivery takes place with a temperature glide to maximize energy efficiency during heat transfer, and the evaporator can easily operate at ambient temperatures. This renders a high *COP* heat pump, and provide a simple solution to increase the energy efficiency of hot water production. Further accounts of such heat pump systems are found in Julian Milnes article [43] describing the CO_2 heat pumps manufactured by CO_2 equipment specialist Enex. A thorough thermodynamic description of the CO_2 heat pump's operating principle and hot water heating process application is also given by Jørn Stene, as part of his Master Module at the Norwegian University of Science and Technology.[46] Conducting a performance analysis investigating heat pump application in this process is therefore rather redundant, and not pursued in this thesis. The dehumidified air stream is therefore the only heat sink considered in the performance analyses.

Scenario Specifics

The heat source evaluated in the scenario based on EWOS' production line is a waste stream of saturated air, initially at 50°C and a mass flow rate of 100 000 m^3/hr. Equation (13) estimates the stream's maximum potential heat release. It is used by a heat pump delivering higher temperature useful heat to the moist air stream used for dryer applications, which is its designated heat sink. The calculations are based on standard simplifications for psychrometric mixtures, assuming the saturated air flow to consist of dry air and pure water vapor. Mass flow rates of the dry air and water vapor are estimated by the ideal gas equation of state. [45] Humidity ratios subsequently predict the mass flow and condensation rates after heat exchange with the heat pump. The stream's total heat release, specific enthalpies and flow rates are available in the spread sheet enclosed in appendix 2, in the *Heat Source* table.

The process heat sink elevates a dehumidified air flow to 110°C. Equation (13) predicts the potential for heat extraction and heat recovery (both passive and active) in appendix 2. (The heat pump is implemented between HX5/6.) Dehumidification parameters of the air stream were, as suspected in section 3.1 in relation to figure 4, over-estimated by the specified numbers. An energetic evaluation of the dehumidified air at different temperatures estimates the process heat requirements. The spread sheet table *Base Case Scenario* in appendix 2 relays these quantities, and clarify the heat sink's temperature range. The current heat recovery system is only capable of elevating the air flow temperature to 63°C, assuming that the heat pump already implemented operates with a mechanical work input of 23 kW (as specified by EWOS [33]). This increases the heat sink's temperature range to 47°C (between 63°C and 110°C), requiring 383.3 kW of heat delivered from a boiler under idealized operating conditions.

The key elements are summarized in table 4, illustrated by figure 22a, and obtained with data available in table 1 and appendix 2.

4.1.2 Scenario TINE Meieriet Verdal

Section 2.2, table 1 and appendix 1 present a multitude of available heat sources and heat sinks in TINE's production line. Most operates within similar temperature ranges and uses the same working mediums. This limits the advantages of creating more than one scenario based on these process streams, as the performed analyses are easily transferrable between utilities. Heat sources and sinks in both drier processes (WPC and permeate production) are therefore discussed simultaneously below.

One heat source is directly available in both spray driers, and its operating temperature is assumed identical between the two: the spray drier's outlet air carriers heat at ∼70°C. Heat recovery systems utilize some of this heat to preheat the air entering the spray drier systems, and lower the air flow temperature to ∼50°C. As heat recovery systems based on simple heat exchange are the most efficient way of extracting excess process heat, this system remains untouched during the succeeding performance evaluations. Accordingly, the available waste heat holds ∼50°C, and flow at the same rate as the drier inlet air. However, these waste streams are not considered as potential energy sources for a heat pump delivering heat to the same flow at a significantly higher temperature when other heat sources are available, due to the low specific enthalpy of dry air flow (cooling the air to ambient conditions only release ∼200 kW of heat, with a temperature glide which may severely strain potential working fluids).

Both process flows systems use hot water district heat at 105°C to heat the drier air flow, designed with a return temperature of 70°C to the district heat exchangers. The water's temperature and heat capacity make it an ideal heat source for the investigations performed in this thesis, capable of delivering a suitable amount of heat to upgrade in a heat pump. However, as it is obtained from a district heat system, TINE likely pays per kilowatt energy extracted, as is customary in heat distribution systems.[29] Thus, the scenario does not consider further exploitation of this heat source for the purpose of heat pump application, to limit the operational energy costs.

Referring to table 3 and appendix 1, several other heat sources are identifiable among TINE's cooling processes. The most energy intensive cooling process is a TVR flash cooler, utilizing 9800 kWh/day or 408.33 kW. The TVR product temperature experiences a drop from 60°C to 35°C, which indicates the temperature bounds of the potential heat exchange. The temperature glide in a heat exchanger coupled with this heat source depends on the applied working fluid, but it is bounded by this temperature range. Two other heat sources are also introduced in table 3, with excess heat available in compressor cooling systems (∼42 kW of heat available if cooled to 35°C) and in pasteurization cooling systems (∼81 kW of heat available). They are not, however, considered comparably rich in energy.

Four heat sinks are identified as suitable for the performance evaluations performed in this thesis. The air flow entering both driers requires a final temperature elevation exceeding 60°C (ideally 98°C and 68°C in WPC and permeate drying, respectively) by boiler systems, delivered at a final temperature transcending 150°C (185°C and 160°C respectively). These temperature ranges are suitable for investigation of high temperature heat pump application. Evaluated systems must be capable of delivering the heating processes required heat loads (specified as ∼700 kW for the WPC drier system, ∼1200 kW for the permeate drier system).

The two remaining heat sinks are needed for high-pressure hot water production, at 122°C and 100°C. Heat utilities of ∼100 kW and ∼340 kW are needed in production, and the temperature is elevated from 55°C and 60°C, respectively. These heat sinks are noticeably similar to the hot air production in EWOS' process line, both with regards to energy consumption and temperature elevation. And as heat transfer effects between the heat pump's heat exchangers and the heat source(s) and sink(s) are of less importance than temperature lifts in the thesis' analyses, a scenario using these heat sinks would resemble that designed from EWOS' processes to a great extent. Heating the driers' inlet air flow, after the district hot water heat exchange, is therefore investigated further in this thesis. However, as the WPC and permeate drier air streams are quite similar, only one is examined in the succeeding calculations. The WPC drier air stream is chosen, as it operates over with a wider temperature range.

Scenario Specifics

The evaluated heat source from TINE's production line is excess heat in a TVR flash cooler, used in whey permeate production. Cooling water currently removes 408.3 kW of heat, with a temperature spanning from 50°C to 35°C. As cooling water is a direct energy loss, a heat pump can utilize this source to produce useful heat at an elevated temperature. And with both the temperature span and the released heat quantity given, no further developments are needed with regards to the heat source.

The process' heat sink is the air flow used in WPC production. Heat is provided as it exits the district hot water heat exchanger, and the temperature ranges from 87°C and up (bounded to 185°C, the drier air inlet temperature). Equation (13) predicts the initial temperature lift generated by the process' heat recovery system (refer to figure 5) and the district hot water heat exchange. The spread sheet *Base Case Scenario* table, found in appendix 3, present the minimum heat requirements of each temperature lift, and the actual energy use (as specified by TINE Meieriet Verdal). When the heat pump is implemented in this system, it replaces some/all of the heat currently delivered by a boiler (ideally ∼516 kW, actually ∼700 kW). This affects heat recovery system, which must be accounted for when the revised system's energetic performance is evaluated.

The calculations assume the air to by dry (its relative humidity is 0), as TINE uses ambience air in their processes. Accounting for entrained water vapor in the air increases the heat requirements, but circumventing it does not render the performance estimations unusable. It is first and foremost the temperature lift and temperature operating range that is investigated in this thesis, and the quantity of delivered heat does not restrict the heat pump application under these circumstances. Seasonal changes will also greatly affect the air's relative humidity, and assuming the air to be dry simplifies the performed calculations without compromising their applicability.

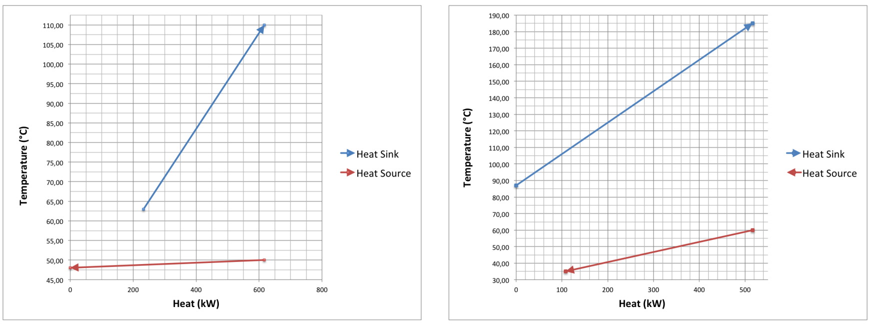
The key elements are summarized in table 4, illustrated by figure 22b, and obtained by data found in table 1 as well as appendix 1 and 3.

4.1.3 Summary

As evident in figure 22a, EWOS has excess quantities of low-grade heat available in their production process, and the amounts exceed those of the high grade heat requirements. TINE has on the other hand, a high-grade heat requirement which exceeds the available heat utilities. The mechanical work and/or external thermal heat input must therefore equal or surpass this energetic deficit, to provide enough energy to the heat sink.

High Temperature Heat Pump Application				
	Heat Source		**Heat Sink**	
Scenario	**Temperature**	**Heat Release**	**Temperature**	**Heat Demand**
EWOS	50°C → Ambient	Not restrictive	63°C → 110°C	≥ 383.8 kW
TINE	60°C → 35°C	≤ 408.33 kW	87°C → 185°C	≥ 516.33 kW

Table 4: Scenarios for high temperature heat pump application



(a) EWOS (b) TINE

Figure 22: Heat quantities and temperature glides of the identified scenarios

4.2 Preliminary Selection

Two factors categorize the heat pumps' applicability through this preliminary selection analysis. One accounts for mismatches between available and required thermal reservoirs in the industrial processes, and the introduced heat pump working principles. The other evaluates ideal heat pump performance and energy requirements, using the COP relations developed in chapter 3.1. Three categories are established, as introduced in table 6, and the general classification of each heat pump is proposed by the preliminary selection analysis. The results nominate heat pump systems for further consideration in the detailed performance analyses conducted in chapter 5.

Preliminary Selection Categories		
1.	Applicable	Heat pump operating principle matches the scenario's heat source(s), sink(s) and temperature lift without revisions.
2.	Manageable	Research and modifications are needed to successfully intergrate the heat recovery system (i.e the plant utility approach).
3.	Difficult	Major compromises restict heat pump application with the available process flows.

Table 5: Preliminary selection categories

Neither category excludes application of a heat pump system, but indicates the expected difficulty and/or energy efficiency related to the various working principles. The preliminary efficiency analyses only consider system from category 1. and 2. This decision is supported by two main arguments. *One*: Heat pumps initially identified as category 3. systems may improve the process' overall energy efficiency. However, the payback period of system implementation is likely unfeasible. *Two*: A multitude of potential solutions exist, and a selection should be made to limit the sampled amount for further analyses. A wide system assortment requires the calculation tool to accommodate an uncertainty analysis to bring a satisfactory level of precision to the results (i.e. economics of purchasing energy from multiple sources to run the heat pumps), which complicates it significantly. Thus, by solely including heat pumps defined as category 1. and 2. systems in the analyses, focus is kept on appointing relatively simple heat pump systems to improve an industrial process' energy efficiency. It should be kept in mind that this is a preliminary selection, and obtained results must be treated accordingly. Systems classified in category 1 are not necessarily the most applicable under realistic circumstances, but those that initially present the best fit and energy efficiency.

4.2.1 Matcing Process Heat and Working Principle

Heat pumps are categorized by their operating principle's compatibility with the simplified industrial scenario descriptions. These descriptions relate closely to the three key factors of a heat pump system, introduced in section 2.2.1. Heat transfer with the available thermal reservoirs haw a particular influence on the chosen heat pump configuration. Working mediums and heat exchanger systems are considered unrestrictive in this analysis, as heat pumps are adaptable to air-to-air operation without major complications. As ∼369 000 heat pump units currently operate with air-to-air thermal reservoirs [47], this is a valid assumption. Temperature elevation is a challenge often restraining heat pump application. But as it is highly dependent on a heat pump's working fluid(s) and internal pressure generation, it is not considered a limiting factor in the preliminary selection.

A characteristic feature of both the industrial processes presented in section 4.1 is the excess heat streams' temperature level. Heat is solely available at temperature levels below those of the desired heat sinks. Current heat utility systems consist of external energy sources, in the form boiler systems (either gas fired or electrically powered) and district hot water heating systems. Direct recovery systems are utilized to heat the streams from their coldest temperatures. Referring back to table 2 in section 2.3.3, certain heat pumps systems (the heat amplifiers) produce the required temperature lift and useful heat release with a high temperature heat source, and release it at an intermediate temperature level. These systems are useful to redistribute heat loads and decrease temperature driving forces in industrial processes with large temperature spans between process streams. However, both EWOS and TINE require external heat utilities to power these heat pumps, which create a redundant system setup. Purchased high temperature heat used to drive a system delivering lower temperature heat increase system inefficiencies, as utility heat exchangers and other heat pump components are bound to experience thermodynamic losses. Heat pump capital costs, operational costs and maintenance cost are additional expenses, and further reduce the appeal of heat amplifiers. Application of type I absorption heat pumps, adsorption heat pumps and chemical heat amplifiers is accordingly difficult to defend, which classify them as category 3 systems.

Table 2 includes two open-cycle heat pumps, which discard their working fluid after releasing the produced useful heat. Thermocompression heat pumps recompress excess vapor, to increase its pressure and generate high-grade energy. Open-cycle mechanical compression heat pumps use a mechanical compressor to increase the vapor pressure and temperature to hot steam, which is condensed to release useful heat. The steam's residual heat (after condensing) is used to used to preheat the vapor stream entering the compression stage, by simple heat exchange. A cold stream entering the heat pumps is therefore required to be low pressure vapor. And as neither scenario heat source is a vapor stream (a saturated air stream is identified among EWOS' processes; cooling effects from a a flash cooler is used for heat recovery in TINE's production; ref. table 1 section 2.2.3 and/or section 4.1), these systems are not investigated further in this thesis.

Thermally driven heat pumps receiving external heat at an intermediate temperature level, the heat transformers, are based on operating principles appropriate for the introduced industrial scenarios. Their energetic performance and heat capacities are, however, strongly dependent on heat input temperature levels and the selected working fluids. Previous research and analyses have shown that differences in generator and evaporator temperature levels have a profound effect on both heat pump COP and the absorber's heat capacity (and accordingly the system's useful heat release).[40] This affects their energetic evaluations, but not their match with the available heat sources. Low temperature heat sinks are also required (cooler than the driving heat supply), and the nature of this heat sink should be considered (directed to a low temperature heat recovery system, or simply lost as waste stream). Both EWOS and TINE dump low grade heat to sea water, cooling water and/or ice water at temperatures close to 0°C (refer to section 2.2 and appendix 1), and low grade heat sinks are readily identified. Heat transformers are accordingly deemed applicable in the identified process scenarios, but their coupling with the available heat utilities demand further investigations to establish effective heat recovery systems. These systems include type II absorption heat pumps and chemical heat transformers.

Table 2 introduces three more heat pump systems: two closed-cycle mechanical compression heat pumps (with single or multiple compression stages) and hybrid heat pumps. Their prominent common features are operation between two separate thermal reservoirs, and compressors producing temperature elevations. Low grade excess heat is extracted from process streams, and a mechanical work input increase its temperature level to release high grade energy to a designated heat sink. Mechanical compression heat pumps use a pure working fluid, and heat is typically transferred isothermally by a working fluid phase change. Temperature glides are attainable (by sensible heat release or transcritical operation), but normally at a trade off with other system design parameters. Hybrid heat pumps operate with a mixture of two working fluids, and manipulating the concentration levels in the absorber and desorber controls heat transfer temperature glides. As both EWOS and TINE have large temperature glides in their heat sinks, ref. figure 22, it is beneficial to design heat pumps accordingly. Their heat sources are (in comparison) practically isothermal, and should be accounted for as well. Proper system design renders these three systems highly applicable, at the assumption that applied heat pump components cope with imposed pressure and temperature levels.

4.2.2 Performance Coefficients

The thesis aims to investigate heat pumps capable of high temperature heat recovery, and compare their energy efficiency to the currently applied heat utilities. This section evaluates ideal efficiency of the introduced heat pump working principles, and their sensitivity to various changes in thermal efficiency and available reservoirs. A comparison to the current system utilities is also performed. The calculations are supported by equations (2)-(11), developed in chapter 3 and summarized by table 3. Performance is based solely on the operational temperature levels in the two scenarios. Restrictions of high temperature elevation are not accounted for, as these depend on a system's applied components as well as on its working fluid(s). Such factors are investigated further in the performance analysis conducted when specific system are identified, supported by the results of this preliminary selection.

EWOS has, as evident from figure 22a, excess quantities of low grade heat available compared to the high grade heat requirements. To maximize heat pump performance, heat extraction takes place at the hot end of the heat source's temperature glide. TINE has, on the other hand, heat requirements which exceed the available waste heat utilities. Added mechanical work and/or thermal energy inputs must therefore equal or surpass this energetic deficit.

Driving forces are introduced as a percentage of the required temperature lift, suggested as an appropriate measurement of system inefficiency at the end of chapter 3. System sensitivity to the heat source temperature is also investigated, in case future process developments, system retrofit, etc. influence the waste streams' temperature levels. The changes are evaluated as a percentage change of the current temperature levels. Work requirements are calculated and compared to the current heat utility input (ideally and with system inefficiencies). Calculations of heat pump COPs are summarized in the tables of appendix 4, alongside reservoir temperatures, heat exchanger driving forces and heat transfer quantities. System performance is considered separately for EWOS and TINE.

The main results obtained in the preliminary performance evaluation are summerized in table 6. Idealized results are based on the heat pumps' carnot efficiency; heat transfer does not require temperature driving forces, and pressure increase is produced without thermodynamic losses. Realistic results refer to system operation with thermal driving forces, and losses in pressure generation. The driving forces are estimated to 15% of ΔT_{LIFT} for the mechanical compression heat pumps, 10% for the thermally driven systems. Compressors are assumed to operate with an isentropic efficiency (η_C) of 0.6. Boiler efficiency (η_B) is 0.75. This represents a worst-case scenario among on the parameters introduced in chapter 3. (Abbreviations M.D. and T.D. in table 6 refer to mechanically driven heat pumps and thermally driven systems, respectively.)

EWOS
Mechanical Compression and Isothermal Heat Transfer: Equation (2) evaluates the heat pump efficiency of isothermal heat transfer. Its carnot efficiency is achieved in operation without losses in neither components, and represents the isothermal heat pump's maximum performance. At 6.2, the heat pump COP indicates that the work input is approximately one sixth of the process heat requirement (62.1 of 383.8 kW), and the system's energy consumption reduces by 321.7 kW compared to the idealized, current utility system. This performance is, however, far from achievable, and further investigations should be conducted, even at the early stage of preliminary selections.

Accounting for thermodynamic inefficiencies in both system heat exchangers, as well as performance losses in the compression stage (throttling losses are not considered), decreases the isothermal heat pump's performance considerably. The lowest system performance is achieved with temperature driving forces at 15% of the thermal reservoir's temperature difference (9.3°C), and an isentropic compressor efficiency (η_C) of 0.6. System COP reduces to 2.9, a ~52.74% reduction of the carnot efficiency. Work requirements of 131.4 kW are nevertheless low compared to the current utilities, and the heat pump achieves an energetic improvement of 252.4 kW compared to the idealized boiler system. Reducing the boiler efficiency increases the heat pump's energy savings to 380.3 kW, and yields a COP of 3.9 when implementation is considered under realistic circumstances.

Implementing a system with a COP of 2.9 is reasonable from an energetic point of view, but an economical evaluation (with regards to payback period, etc.) may stagger the applicability this system. A system performance of 3.89 increases its attractiveness, but economic investigations are still key to successful implementation.

The mechanically driven isothermal heat pump's dependency on its heat source temperature is also investigated, at carnot efficiency. Imposed temperature changes of up to 20% have little effect on overall performance (COP), a mere 2.6% decrease and 2.4% increase when the thermal reservoir is cooled and heated, respectively.

Mechanical Compression and Thermally Gliding Heat Transfer: Equation (3) evaluates heat pump efficiency when heat is transferred with a constant temperature driving force fitted to the process stream's thermal development, and quantifies the importance of thermodynamic efficiency in heat transfer. EWOS has a large glide in their heat sink process stream (47°C), where the beneficial effects of constant driving forces are prominent. The heat source temperature is close to constant (a glide of 2°C release 616.4 kW), and diminishes the advantages of temperature glides compared to isothermal heat exchange. System COP at carnot efficiency increase to 9.71, a 57.1% improvement compared to the ideal isothermal heat pumps. External work input driving the compressor system reduces to 39.5 kW, 10.3% of the ideal performance of the current heat utilities.

Introducing thermodynamic inefficiencies decreases system COP to 6.63 in the worst case scenario (a thermal driving force of 9.3°C and $\eta_C = 0.6$). System performance is, however, still superior to the ideal isothermal heat pumps due to the extensive heat sink temperature glide. 287.2 kW are cut of the current utilities under idealized circumstances, 415.2 kW are saved of the realistic energy consumption.

A stronger dependency on heat source temperatures was uncovered under conditions identical to those introduced for isothermal heat transfer (a 20.6% decrease and 35.0% increase). A lower average heat sink temperature ($\overline{T}_H$), produced by the heat exchanger's thermal glide, explains the fluctuations. Performance is, however, still high, and the results cannot be emphasized as a problematic feature of these systems.

Thermally Driven Heat Pumps: The performance of thermally driven heat pumps differs significantly from the mechanical compression systems. Developed from equation (9), their COPs are lower than 1 as waste heat loads are split between high and low temperature heat sinks, and release useful heat as a fraction of the supplied quantities. Isothermal heat pumps operate with a COP of 0.42 at carnot efficiency, and a COP of 0.55 is achieved if the absorber and generator are operable with temperature glides to match the process streams. The benefits of thermal glides are, accordingly, clearly visible in thermal heat production as well. Waste heat load requirements of 910.5 kW and 694.4 kW, respectively, are imposed on the heat sources, manageable by EWOS' cold thermal reservoir (which releases $\sim$1620 kW if cooled to 45°C).

As solution pumps hold the heat pumps' only external energy requirement, these systems are highly effective when waste heat is available in sufficient quantities. Thermal systems are therefore unparalleled by any other heat utility, if their potential is utilized efficiently. However, challenges arise when the heat source temperature fluctuates, and when working fluids are introduced to the equations.

The latter effect is not explored in this section, but the former is.

Introducing temperature driving forces decrease system performance. COPs reduce to 0.38 and 0.47 and heat requirements increase to 1023.4 kW and 818.1 kW, for isothermal and thermally gliding systems respectively. The effect of changing heat source temperature is even more problematic, and a 20% decrease (9.6°C) of this temperature reduces system COPs to 0.32 and 0.42. EWOS' waste stream heat load is still large enough to cover these heat loads, but the quantities strain the heat exchangers' design (to maintain efficient heat transfer) and cost. Imposing the restraints of temperature driving forces on the heat pumps further decrease their applicability, and diminish the favorable features of minimized external energy requirements.

Manipulation of the cold heat sink temperature is investigated as well, and a profound effected system performance is uncovered. Beneficial features accompany a colder heat sink (due to a decreased low side system pressure), whereas increasing the cold heat sink temperature affects system performance negatively. These effects should be kept in mind during implementation, as proper design of the low temperature heat release can increase system efficiency greatly.

Elevating the heat source temperature is advantageous in terms of heat load requirements, and increased heat pump COP considerably (to 0.52 and 0.67 at a 9.6°C temperature increase). The profitable feature of limiting external energy requirements merits investigation of higher temperature heat sources, potentially delivered at an economical cost, to utilize this effect. EWOS has high-pressure hot water at 90°C readily available in production, which could serve as a heat source and drive a thermal heat pump system. If it is cooled to 80°C, mass flow rates of 12.03 kg/s (isothermal system) and 9.23 kg/s (thermally gliding system) are able to run the heat pumps with carnot efficiencies of 0.76 and 0.99. Thermal load requirements from the heat source reduce by ∼44% (to 505.3 kW and 387.9 kW respectively) in both cases, and two highly interesting systems are introduced. Applying temperature driving forces to the heat exchangers decrease their performance somewhat (and increase the required mass flow rates from the high temperature hot water), but the solutions are still vastly superior to the idealized systems operating with colder heat sources. These results are obtained with heat release at a cold side temperature of 10°C.

An isothermal system increases the external hot utility requirements by 121.5 kW compared to current heat utilities under idealized operating conditions, and with 37.4 kW under realistic circumstances. A thermally gliding system uses 4.1 kW more heat than the current hot utility when thermal inefficiencies are neglected, but 75.9 kW less temperature driving forces and a boiler efficiency are included in the performance analysis. Thermally driven systems may therefore be beneficial from a cost perspective, as the temperature requirements are decreased significantly (at least 20°C, from hot utility temperatures exceeding 110°C). A in depth performance analysis and an economic evaluation should accordingly be conducted to validate or disprove the applicability of these systems.

Tine Meieriet
Mechanical Compression and Isothermal Heat Transfer: An idealized COP of 3.05 is obtained for heat pumps with isothermal heat transfer, imposing an external work requirement of 169.1 kW on the compressor system driving the heat pump. Idealized performance with the current heat utilities necessitate 516.3 kW of process heat, rendering the heat pump an effective energetic substitute (potentially lowering the external energy supply by 67.25%).

Introducing system inefficiencies decreases the heat pump performance to a COP of 2.46 and increase the net work requirement to 349.2 kW, as a worst case scenario. It is still a preferable system, in terms of energy consumption, compared to the currently employed heat utilities (with a margin of 166.1 kW in the idealized case). If the heat utility approaches realistic consumption levels, the boiler efficiency (η_B) decreases to 0.75, and the energetic benefits of heat pump application increases (and reduce the system's total energy consumption by 339.2 kW).

Changing the heat source temperature has little effect on overall heat pump performance due to the large temperature lift requirements between TINE's available heat source and desired heat sink. Evaluating larger fluctuations are neither necessary nor productive, supported by to two main arguments. There is little waste heat available at higher temperatures, and the evaluated heat source has a limited amount of energy to release. Heat exchange with multiple sources is accordingly not an option.

Mechanical Compression and Thermally Gliding Heat Transfer: The extended temperature glide of 97°C between the inlet and outlet temperatures of TINE's desired heat sink favor heat pumps capable of heat exchange with changing temperatures. The heat source has a temperature glide of 25°C, in contrast to the thermal reservoir available in EWOS' production process. This favors use of temperature glides in the heat exchangers to limit heat transfer inefficiencies. An ideal COP of 4.69 demand a work input 110.0 kW, 35% less than isothermal heat transfer systems and 78.7% less than the idealized boiler system. And as the cold reservoir releases a maximum of 408.3 kW, process integration minimizes waste heat release from these systems (a heat surplus of 2 kW remains at a temperature close to 35°C) and accordingly its cooling utility requirements.

The system's COP is decreased to 3.26 by imposing a temperature driving force (15% of T_{LIFT}) on the heat exchangers, and the compressor work input increases to 263.8 kW when an isentropic compressor efficiency (η_C) of 0.6 is introduced. However, performance remains high compared to other realistic system. It consumes 24.5% less energy in compression than systems with isothermal heat transfer, and 61.7% less energy than a boiler with an efficiency of 0.75.

The influence of fluctuating heat source temperatures is noticeable, but neglected from further analysis. Performance remains high despite a 20% temperature decrease.

Thermally Driven Heat Pumps: Thermally driven heat pumps encounter a severe challenge in heat recovery in TINE's production. High output temperatures (peaks at 187°C) and a relatively cold intermediate heat source (range from 50°C to 35°C) yield low system COPs. This imposes a high quantity heat input requirement, which cannot be covered by the identified process heat source. The problem is solvable by identification of other available waste streams among TINE's production processes, but these generally hold a lower temperature. This decreases the already low system COPs, ranging 0.16-0.27 for isothermal heat pumps and 0.32-0.44 for thermally gliding heat pumps. Carnot efficiencies of 0.21 (heat load requirement of 2430.6 kW) and 0.38 (heat load requirement of 1349.9 kW), respectively, cannot defend system applicability with reasonable payback periods.

However, TINE has access to a district hot water heating system at 102°C, already utilized to preheat the dryer air stream. Increasing the amount of purchased heat from this source facilitates use of the thermally driven heat pumps. Mass flow rates of 10.75 kg/s (isothermal operation) and 7.65 kg/s (thermally gliding systems) introduce systems with carnot efficiencies of 0.52 and 0.73, respectively. The hot water returns to the district heat system at 80°C. Thermal driving forces in the heat exchangers increase the required mass flow rates, but should not restrict system development, as it is achievable with standard components.

Temperature driving forces at 10% of ΔT_{LIFT} (98°C) requires 1080.5 kW and 812.8 kW of heat supplied to the heat pumps at the intermediate temperature, large quantities of purchased external heat. Isothermal heat pumps require an excess of 392.1 kW compared to the current boiler systems, under the constraints of realistic operation, and heat pumps with temperature glides require 124.4 kW more than the current boiler system. There is, however, a beneficial feature of applying thermally driven heat pumps. The purchased heat is supplied at 102°C, a full 78°C less than the boiler temperature (without considering the boiler's required temperature driving forces). A significant reduction of energy cost is possible, and the heat pump systems should be evaluated further (with respect to energy efficiency and economic viability) before their applicability can be validated or disproved with reasonable amount of certainty.

4.2.3 Results

Three of the introduced heat pump systems (closed-cycle mechanical compression heat pumps with isothermal heat exchange, closed-cycle mechanical compression heat pumps with thermally gliding heat exchange and closed-cycle thermally driven heat transformers) have been evaluated with regards to their basic energetic performance, within the confines of EWOS' and TINE's production processes. The remaining heat pumps were deemed too difficult/expensive to apply, due to a significant mismatch between their operating principle and the available thermal reservoirs, and omitted from the preliminary energetic performance analysis (the obtained results would be useless in nomination of a set of best-practice solutions). Table 6 summarizes the main results of section 4.2.2, and is used in combination with section 4.2.1 to propose a categorization of the all the introduced heat pumps.

Closed-cycle mechanical compression systems are superior to the current heat utilities as well as the thermally driven heat pumps with regards to operational energy efficiency, supported by the results of the preliminary performance analysis. Direct application between the available thermal reservoirs in both scenarios, as described in section 4.2.1, further increases their attraction.

Temperature glides are confirmed as a beneficial feature to be pursued in heat pump development. (However, the numerical results in table 6 overestimate the performance of these systems.) Their performance is largely explained by the assumption of constant temperature driving forces. Developing working fluids capable of following the thermal progress of any heated process stream is not possible, and the applied heat transfer efficiency should be reduced to estimate a realistic system performance. Predictions of system consumption are accordingly not to be used as more than indicators of performance.

Isothermal heat transfer also introduces an energy efficient heat recovery system compared to the current boiler systems. The simplified performance evaluations estimate that these mechanically driven systems halve the external energy requirements in both scenarios.

Preliminary Performance Evaluations				
	EWOS		**TINE**	
System	Idealized	Realistic	Idealized	Realistic
Boiler	383.8 kW	511.7 kW	516.3 kW	688.4 kW
M.D. Isothermal H.P.	Heat Source: Original			
Energy Consumption	62.1 kW	131.4 kW	169.1 kW	349.2 kW
Potential Reduction	*321.7 kW*	*380.3 kW*	*347.2 kW*	*339.2 kW*
M.D. Gliding H.P.	Heat Source: Original			
Energy Consumption	39.5 kW	96.5 kW	110.0 kW	263.8 kW
Potential Reduction	*344.3 kW*	*415.2 kW*	*406.3 kW*	*424.6 kW*
T.D. Isothermal H.P.	Heat Source: Alternative			
Energy Consumption	505.3 kW	549.1 kW	994.9 kW	1080.5 kW
Potential Reduction	*−121.5 kW*	*−37.4 kW*	*−478.6 kW*	*−392.1 kW*
T.D. Gliding H.P	Heat Source: Alternative			
Energy Consumption	387.9 kW	435.8 kW	707.8 kW	812.8 kW
Potential Reduction	*−4.1 kW*	*75.9 kW*	*−191.5 kW*	*−124.4 kW*

Table 6: Main results of the preliminary performance evaluations

All closed-cycle mechanical compression heat pumps are accordingly categorized as applicable systems (category 1). This includes single- and multistage compression systems, as well as hybrid heat pumps.

Thermally driven systems appear to have an inferior energy efficiency compared to the current heat utilities, but the energy deficit seen in table 6 is present due to a significant temperature decrease of the purchased energy. It should, accordingly, not be considered a decisive system limitation unless further investigations (of both energy efficiency and economic potential) are pursued. No mechanical work input is required (as the solution pumps are neglected), but high temperature hot water must be purchased to drive the heat pumps thermally. These heat load requirements hinder direct applicability between the available thermal reservoirs, which alongside the increased energy requirements classify the heat pumps in category 2, as they necessitate certain modifications. The categorization includes all available heat transformer systems.

Heat amplifiers and open-cycle systems are deemed difficult and costly to implement based solely on their mismatch with the process requirements. As clear category 3 systems, they are discarded from further analysis.

This concludes the preliminary selection, and the nominated system applicability is summarized in table 7. Category 3 systems are disregarded from future performance analysis, due to their variable limitations described above. The working principles of closed-cycle mechanical compression are deemed most-likely-to-succeed in application with the identified industrial scenarios, and selected for an extended performance evaluation. Thermally driven heat transformers also present energetic and economic benefits compared to the current heat utilities. However, the overall expectancy of increased energy consumption render them less interesting within the scope of this thesis.

Prominent investigations (analytically and/or experimentally) of mechanically driven heat pump designs are accordingly introduced in chapter 5, and undergo a closer examination of certain key performance parameters. This includes an introduction of suitable working fluids, and evaluation of applicable system components. Equations developed in section 3.2 are used to quantify the various heat pumps' expected performance.

Preliminary Selection		
1.	Applicable	Closed-cycle mechanical compression heat pumps Hybrid heat pumps
2.	Manageable	Type II absorption heat pumps Type II chemical heat pumps.
3.	Difficult	Type I absorption heat pumps Type I chemical heat pumps Adsorption heat pumps Open-cycle thermocompression heat pumps Open-cycle mechanical heat pumps

Table 7: Preliminary selection

5 Applicable Heat Pumps and a Performance Analysis

Research of (industrial) heat pumps operating in the investigated temperature ranges is scarce, and available results from numerical and experimental performance evaluations are limited. Three basic systems are accordingly described and investigated analytically in the succeeding sections, based on the working principles nominated most-likely-to-succeed by the preliminary selection.

Applicable working fluids must be appointed, and the selection is supported by prominent systems identified in the technical review. The performance evaluations aim to disclose the thermodynamical states required for successful operation as heat recovery systems in EWOS' and Tine's production processes. Pressure, temperature, enthalpy and mass flow rates are important factors in heat pump design (ref. section 2.3-2.5), and CoolPack simulations are performed to identify their cyclic values. Overall energy consumptions are estimated with these parameters, with equations introduced in chapter 3.2 at the center of the calculations.

Certain systems require extensive numerical analyses to determine their performance accurately, and this limitation is particularly prominent in heat transfer to/from fluid mixtures. The thermodynamics of sorption processes are highly complex, and require extensive simulations to be evaluated with satisfactory accuracy. Simplifying measures are therefore introduced alongside these complexities, to help characterize the effects' influence on overall system performance and applicability of market available system components. The issues are most prominent in evaluations of hybrid heat pumps.

Current research of two thermally driven systems, deemed with manageable applicability in the preceding sections, are introduced to emphasize their potential in future applications. The heat pumps are unable to reduce overall energy consumption, but run with heat input at lower temperatures than the current heat utilities. Long-term economical analyses are therefore necessary to verify/disregard implementation, but fall outside the scope of this thesis. The identified research also fails to address the main challenge of this thesis (appointing the potential of heat pump energy recovery systems at high temperature, with currently market available components), and further analyses of these heat pumps are consequently omitted. Extensive technical research must continue to create readily applicable systems, but is, unfortunately, not commonly performed due to the inherent complexity sorption processes (compared to mechanically driven system research).

A discussion of the obtained results, and a proposal of tasks for further consideration, finalizes the chapter.

5.1 A Singlestage Mechanical Compression Heat Pump

5.1.1 Introduction

The simplest system design considered for application is a singlestage, closed-cycle mechanical compression heat pump, operating with a pure working fluid. A complete heat transfer cycle requires the working fluid to undergo four thermodynamic processes: compression, condensation, throttling and evaporation. Figure 9 of section 2.3 provides an apt system illustration.

These heat pumps have been thoroughly investigated, but their application in high-temperature industrial processes is still limited. A high compressor outlet pressure, required by the working fluid to produce the extended temperature elevations and release high temperature heat, mainly staggers development. This limits the selection market available components. Appointing working fluids capable of high temperature heat release and with a manageable compressor outlet pressure, is therefore key to successful development of these heat pumps.

Working Fluid

Natural refrigerants are a focal point in heat pump development, to accommodate emission restrictions and increased environmental awareness. Numerous solutions have sprung from these investigations. CO_2 is, as mentioned briefly in section 4.1, successfully implemented as a working fluid in hot water production at temperatures around 100°C, using singlestage mechanical compression heat pumps. The low critical temperature of CO_2 facilitates heat recovery from sources at ambient temperatures, and illustrates currently available technologies ability to operate with the high temperature elevations required in the investigated scenarios.

However, an outlet temperatures of approximately 100°C and a low critical temperature complicates application in the identified scenarios. The available heat source temperatures initially hold $50 - 60$°C, and deliver substantial amounts of energy close to this temperature (ref. figure 22). Thus, evaporator heat transfer is either transcritical, or operating with unreasonably high temperature driving forces, if CO_2 is employed as the working fluid.

The high heat sink inlet temperatures in both EWOS' and TINE's processes (63°C and 87°C, respectively) diminish CO_2's heating potential, as transferable energy is available in a limited temperature span of 20-30°C. Increased system mass flow rates, accompanied by higher system costs due to larger equipment, are accordingly required to release the desired heat quantities. (Enthalpy changes are approximated by $dh = C_p dT$ if ideal gas behavior is assumed. [21] This transforms equation (13) to $Q = \dot{m}C_p(T_e - T_i)$ and supports the statement.)

High pressure requirements are also a restrictive factor to further research of CO_2's application in system with output temperatures exceeding ~100°C. Extended temperature elevations impose high system pressures, and limits the applicability of current compression technology. Investigation of CO_2 implementation at higher operational temperatures is therefore limited, and other fluids are the center of further developments.

Extensive research on hydrocarbon working fluids has not been identified in the performed literary review, indicating a fondness for other working fluids despite the attractive thermophysical properties of short hydrocarbon chains. The hazardous operational characteristics of volatile, highly flammable gases probably explain this development. Investigations of water as a pure, high temperature lift working fluid is also limited, most likely due to the large pressure ratios and specific volume changes across the compression stage (requires large swept volumes which increases compressor size). This strains the applicability of currently available compressors. Research on air is also scarce, because of the elevated pressures required to accommodate high temperature heat lifts. This last feature is also a reason not to implement compressors directly in the air streams. All these features are easily extracted from pressure-enthalpy diagrams, available in digital format in the CoolPack Computational Tool.[57]

Research on pure ammonia systems is prominent. They are particularly attractive in industrial applications with heat sinks exceeding 100°C, as alternative systems to CO_2 heat pumps.[51, 58] NH_3's elevated critical temperature (132.4°C) enable subcritical heat transfer at these temperatures, a beneficial feature if heat is exchanged with higher thermal capacity fluids (as the isothermal latent heat release sustain temperature driving forces in the heat exchangers). Elevation of these operational limits should is explored, as NH_3 can be extended to higher temperatures than CO_2 at similar pressure levels (i.e. ~190°C versus ~118°C at a maximum heat pump pressure of 100 bar). Transcritical operation is, however, not yet achieved with standard components due to the extreme system pressure requirements.

The high critical temperature facilitates a subcritical compression cycle, illustrated by figure 17a. The working fluid is evaporated to a saturated gas (or slightly superheated) state by a low grade heat source, pressurized by a dry compression stage, and cooled to release high grade useful heat. An expansion valve depressurize the condensed fluid back to the its evaporator state. A brief description of the thermal processes is given below.

Heat Exchange
Evaporating the cold-side gas-liquid ammonia mixture to a saturated gas is completed by a single heat transfer stage, a straightforward process requiring no further introduction.

Ammonia release both sensible and latent heat, as it is run in a subcritical thermodynamic cycle. Superheated ammonia's peak temperature is utilized to elevate the air streams to their ideal outlet temperature levels (EWOS: 110°C, TINE: 185°C). Latent heat release from condensing ammonia is expected to cover the initial heating demand, as air has a low specific heat capacity and experiences wide temperature glides with moderate heat inputs. This excludes sub-cooling of condensed ammonia from the analyses, which is normally included due to the beneficial effects minimized thermal driving forces and maximized useful heat release from the working fluid.

Useful heat release is accordingly completed in two stages, to maximize heat transfer effects (as gas-liquid mixing during heat exchange is minimized) and avoid pinch point problematics. The air streams enter the system condenser, and heated to a system pinch point by a latent heat release, condensing NH_3 gas from a saturated state. It subsequently enters a gas cooler, where superheated ammonia is cooled from the compressor's outlet temperature. This sensible heat release elevates the air stream to its peak temperature.

Figure 23 [51] illustrates the heat release of both processes. The sub-cooling section is neglected. Superheated ammonia (red line) is discharged from the compressor at temperature T_{WFD}, and condenses at T_{WFC}. This elevates the air streams (blue line) to its outlet temperature, T_{PO}, peaking at a higher temperature than NH_3's condensation temperature (enabled by the initial temperature glide). Heat quantities released by the superheated ammonia depend on compressor outlet temperature and pressure. Condensation is assumed to heat the air streams from their inlet temperatures (T_{PI}).

Process pinch points are encircled in green, and dictate the required heat pump pressure. Working fluids often experience a pressure loss through during heat exchange, and the resulting loss of temperature driving forces is illustrated by the dotted green line in figure 23, and larger heat transfer areas are necessary to sustain heat transfer effects. Similar effects are experienced in the evaporator. This increases the component's size and accordingly cost.

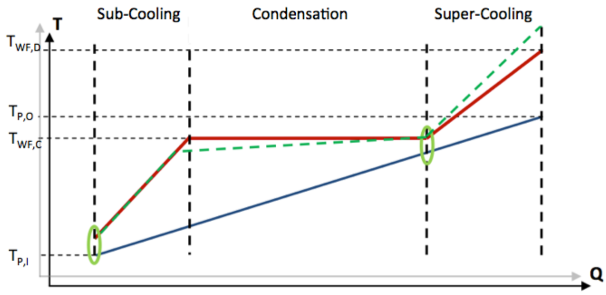


Figure 23: condensing NH_3 from a superheated temperature

Shell-and-tube heat exchangers are chosen in the condenser and evaporator. The evaporator is assumed to be a direct expansion evaporator. Gas coolers are used to cool the superheated ammonia. Ammonia flows tube side counter-current to the heated/cooled medium in all heat exchangers. The aforementioned pressure losses are often attributed to flow in the thin, long tubes. This minimizes temperature driving forces in the system, and maximize the heat transfer potential of available temperature glides. [29, 51] Current systems accordingly propose shell-side air flow, where baffles increase the flow's path length through the heat exchanger (ref. figure 18).

Compression

The compression stage increase the working fluid's pressure after it exits the evaporator. Dry compression is preferred (which requires a superheated fluid), but small amount of entrained liquid is manageable. Multiple compressor systems are applicable and evaluated in current research. Centrifugal compressors (and other dynamic compressors, which increase pressure by converting its working fluid's velocity) are, however, commonly disregarded from high temperature application. Both analytical and experimental research investigates the performance of positive displacement compressors, specifically reciprocating and screw compressors. Open systems are preferred, to simplify maintenance work.

NH_3 is not able to self-lubricate the systems during compression, and require oil-lubrication to cool, seal and lubricate the compressor internals. The oil lubrication is assumed to be perfect, and hinders leakages through the compression stage. This indicates an ideal volumetric efficiency across the compression stage.

5.1.2 Performance Analysis EWOS

Figure 22a skecth the temperature and heat loads of EWOS' available waste heat utility, as well as of the desired heat sink. A heat requirement of 383.8 kW elevates the moist air flow's (mass flow: 8.33 kg/s, initial relative humidity: 10%, atmospheric pressure) temperature from 63.3°C to 110.0°C, and is key to successful heat pump application. The low grade heat is available in excess quantities from a different moist air stream, at temperatures below 50.0°C.

Temperature driving forces of 7.5°C are assumed in both heat exchangers. The relatively large driving forces ($\sim$ 12% of the heat pump's internal temperature elevation) increase validity of the obtained results compared to experimental research and real life applications. Reluctancies to low driving forces are rooted in three statements. *One*: (Moist) air has a low heat capacity compared to ammonia, and tis flow a low heat transfer coefficient. Increasing temperature driving forces reduce the possibility of incomplete heat transfer. *Two*: Large driving forces alleviate the strain imposed by working fluids with varying heat capacities (neither moist air nor ammonia can be expected to heat/cool linearly). Assuming large driving forces reduce the possibility of negative driving forces in real applications, and simplify calculations significantly (ref. section 3.1, and the discussion of Lorentz cycles). *Three*: Research of high temperature heat transfer between industrial scale heat pumps and air is scarce, and a best-practice heat exchanger design is not yet been established.

Ideal Operation

Ammonia is proposed to condense at 102.5°C at a flow rate of 0.393 kg/s. The relatively high condenser temperature is chosen to limit the superheated ammonia requirements (which rapidly increases the compressor's outlet temperature), and maintain a manageable maximum system pressure. Isothermal condensation (to a saturated liquid) increases the moist air's temperature from 63.3°C to 97.8°C, slightly above pinch requirements at the condenser's exit, but with a minimized heat surplus.

Moist air enters a gas cooler at 97.8°C, where it is heated to 110.0°C by the superheated working fluid. The remaining sensible heat load of 100.4 kW is released as ammonia cools from 158.4°C (its maximum cycle temperature at the compressor outlet/gas cooler inlet) to 102.5°C. The cooled ammonia experiences a significantly wider temperature glide than the heated air stream, imposed on the system by vastly different mass flow rates ($(\dot{m}C_p)_{\mathrm{air}} >> (\dot{m}C_p)_{\mathrm{NH_3}}$). Further pinch point problematics are accordingly avoided. Air exits the gas cooler at its peak temperature, dried to a relative humidity of $\sim$ 1.6%, after a heat treatment of 383.8 kW supplied the excess energy recovery system.

Ammonia is assumed to vaporize from a gas-liquid mixture to a saturated gas state at a low system pressure of 16.2 bar (41.5°C). Pressurizing ammonia requires a net work input of 82.5 kW, which implies that 301.3 kW is recovered from the available waste stream. This heat extraction cools the moist air waste source to approximately 49.0°C, a readily available heat load which meets the imposed pinch point requirements.

Ammonia has a specific volume of 0.080 m^3/kg as it enters the compression stage. Isentropic compression increase the system pressure to 65.65 bar, and reduce the specific volume to 0.0262 m^3/kg. The compressor's capacity must accordingly exceed a swept volume of 76.3 m^3/hr.

Approximation to Realistic Conditions

Thermal driving forces are accounted for in the idealized analysis, but ideal compression is not attained in real applications. System inefficiencies are therefore approximated by the isentropic efficiencies introduced in section 3.3, to approach the performance real compressor designs.

Compressor inefficiencies affect the thermodynamic states at the heat pump's high pressure. Understanding the effects of these losses can, however, be used to counterbalance their negative influence on overall system performance, and they are accordingly examined further. The condenser temperature is initially sustained at 102.5°C and 65.65, to reveal the compression stage's impact on overall thermal efficiency. Total heat output is also kept constant, at 383.8 kW, to avoid superfluous heat production. Figure 24 sketches the thermodynamic cycles with various isentropic efficiencies (in a pressure-enthalpy diagram), and illustrates clearly how the compressor outlet temperature increases as its efficiency decreases. (Temperatures are indicated by the red lines, measured in °C.)

Ammonia's increased outlet temperature from the compressor has two prominent effects on the heat pump's operational regime:

One: Its fraction of sensible heat (available as superheated vapor on the right side of the two-phase dome in figure 24) increases, and the latent heat release from condensation is accordingly reduced. This implies that air exiting the condenser gets progressively colder, and the air's remaining temperature elevation must be provided through the gas cooler. This increases the heat exchanger's thermal driving forces, and with it the heat transfer irreversibilities. An isentropic efficiency at $\eta_C = 0.6$, lowest among the evaluated compressor performances, fail to heat the air stream past 93.5°C, and impose driving forces of 9.0°C or more

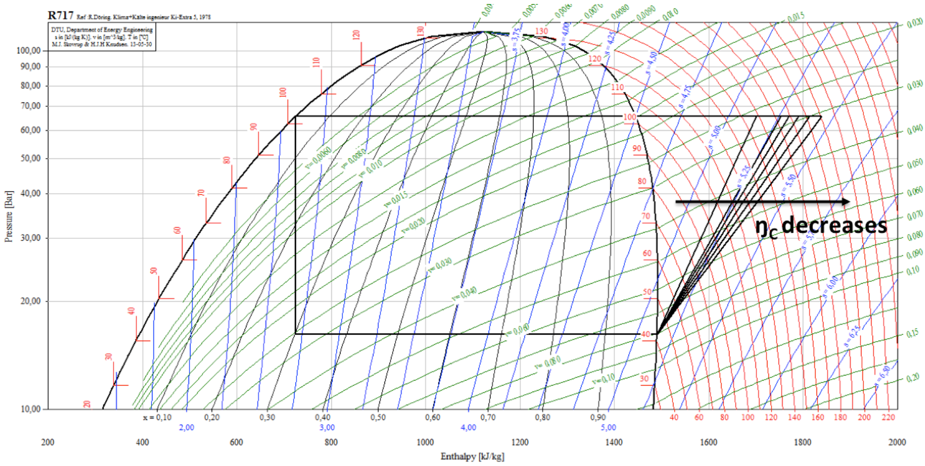


Figure 24: Ammonia's thermodynamic cycle(s) in pressure-enthalpy diagram

throughout the high temperature heat exchangers.

Two: Mass flow rates are steadily decreased, to retain the released heat quantities. It is attributed to the increased maximum temperature, and the elevated temperature driving forces, which requires a smaller mass flow to operate with equal effects in heat transfer. (The effect is aptly illustrated by figure 24, and the elongated enthalpy change at high pressure.)

An increased work input is required to balance the system's decreased heat transfer efficiency, and the percentage change increases steadily with compressor inefficiencies. The net work input when $\eta_C = 0.6$ is 120.4 kW, up 46% compared to idealized conditions. Losses increase the compressor outlet temperature as well, which has a significant influence on component applicability. Estimated effects on work requirements and other important thermophysical properties are summarized in table 8.

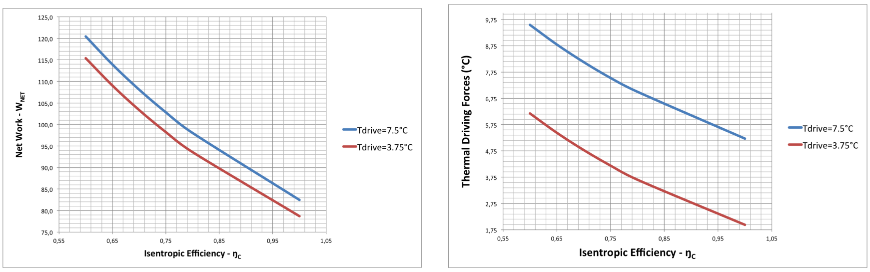
Some beneficial effects arise. Increased temperature driving forces has a beneficial effect on component costs, as heat transfer capacity depends on a directly proportional relation between the heat exchanger's driving forces and the designed heat transfer area. Increasing driving forces decrease the required component size, and yield cheaper components. A compressor's swept volume (and with it overall size and cost) decreases as the mass flow rates are reduced. If the mass flow rate is held constant at the isentropic level, it increases the sensible heat surplus further, accompanied by even larger thermal driving forces.

Identifying a process pinch point in operation with lower isentropic efficiencies is possible, and merit a lower condenser temperature and pressure. However, as identifying these requires extensive simulations, and the results have a significant margin of error (due to system inefficiencies less prominent than the isentropic effect and thermal driving forces, but nevertheless noticeable in real applications), it will have little effect on the thesis' concluding remarks and is omitted.

Both effects are important design parameters in heat pump applications, and must be accounted for when system configurations are considered for implementation. Balancing them against the cost of energy, a heat pump's largest operational expenditure across its lifespan, is beneficial when integrating heat pumps in industrial processes. The scope of this thesis focus on heat pump energy efficiency, and increased work requirements consequently outweigh the cost reduction potential held by large thermal forces in heat exchange in system evaluation.

The extensive driving forces identified in high pressure heat exchange suggest an examination of system performance as the pinch point temperature decrease. Halving the process pinch point to 3.75°C decreases the maximum system pressure to 62.5 bar and the condenser temperature to 100.0°C. Moist air now exits the condenser at 98.5°C after an idealized compression, as ammonia's latent heat load increases with decreasing pressures (illustrated by the widening two-phase area as pressure decreases in figure 24). This exceeds process pinch requirements, but is feasible as the moist air's condenser outlet temperature decreases quickly as compressor losses are introduced.

Estimations of the moist air stream's exit temperature from the condenser are performed to assess the fluctuations in thermal driving forces with the compressor's isentropic efficiency. Rough assumptions let the moist air stream's temperature increase linearly with the supplied heat load, by implying that its specific heat capacity is constant. This is not true (it actually fluctuates around 1.009 kJ/(kgK) ± 0.002 with the given relative humidity and temperature range [57]), but the calculations do not exceed the margins of error of other estimations performed in this thesis. Figure 25b sketches the development, and illustrates clearly how the driving forces escalate quickly from the isentropic values (set by $\eta_C = 1$) at the evaporator exit/gas cooler inlet, regardless of the appointed temperature driving force.



(a) Required net work against isentropic efficiency

(b) Thermal driving forces against isentropic efficiency

Figure 25: The isentropic efficiencies effect on system operation

The notable feature of figure 25 is the cost of decreasing net work by improving heat exchanger efficiency. Halving the thermal driving forces only led to small energetic performance improvements (ranging from 4.8% to 4.4% as the compressor's efficiency decrease from 1 to 0.6). The effect is likely to be more prominent if the working fluid's available thermal glides are extended, and provide a better match to the process stream's temperature development. Exploring this is important when designing to implement a heat pump, but falls outside the scope of this thesis.

The alterations do not present a significant decrease of the maximum system temperature nor other prominent improvements, required to merit selection based on a favorable influence on the other heat pump components. Examinations of the larger temperature driving forces are accordingly pursued in the discussions

Table 8 summarizes the most important thermophysical properties obtained in the cycle analyses. Mass flow rates ($\dot{m}$), maximum system temperature (T_{MAX}) and pressure (P_{MAX}), as well as the compressor's capacity (measured by its swept volume, V_s) are all pressing parameters in heat pump design, and included in the table. Released heat quantities (Q_C) are calculated with equation (13) and summarized alongside the heat pump's sensible heat release (Q_{SH}). Sensible heat loads are specified to illustrate the portion of heat exchange available with a thermal glide. Available latent heat quantities are readily obtained, as the remaining heat release. Compressor net work requirements are included, estimated by equation (14).

Single-Stage Compression of Ammonia EWOS							
T_P(NH_3/Air) 102.5°C/95°C	$\dot{m}$ [kg/s]	T_{MAX} [°C]	P_{MAX} [bar]	V_s [m^3/hr]	Q_C [kW]	Q_{SH} [kW]	W_{NET} [kW]
Ideal: $\eta_C = 1$	0.3935	158.4	65.6	76.3	383.8	100.4	82.5
$\eta_C = 0.80$	0.3740	173.7	65.6	70.1	383.8	115.1	98.0
$\eta_C = 0.75$	0.3671	178.9	65.6	68.0	383.8	119.4	102.8
$\eta_C = 0.70$	0.3602	185.1	65.6	65.9	383.8	124.3	108.1
$\eta_C = 0.65$	0.3526	192.3	65.6	63.5	383.8	129.8	113.9
$\eta_C = 0.60$	0.3441	200.9	65.6	60.8	383.8	136.0	120.4
T_P(NH_3/Air) 100°C/96.25°C	$\dot{m}$ [kg/s]	T_{MAX} [°C]	P_{MAX} [bar]	V_s [m^3/hr]	Q_C [kW]	Q_S [kW]	W_{NET} [kW]
Ideal $\eta_C = 1$	0.3902	153.7	62.5	74.1	383.8	94.0	78.7
$\eta_C = 0.80$	0.3712	168.4	62.5	68.1	383.8	108.1	93.6
$\eta_C = 0.75$	0.3652	173.5	62.5	66.2	383.8	112.5	98.2
$\eta_C = 0.70$	0.3587	179.4	62.5	64.2	383.8	117.4	103.3
$\eta_C = 0.65$	0.3514	186.3	62.5	61.9	383.8	122.8	109.0
$\eta_C = 0.60$	0.3433	194.6	62.5	59.3	383.8	128.8	115.4

Table 8: Single-stage mechanical compression of ammonia: Important thermophysical properties and performance parameters

5.1.3 Performance Analysis Tine

Figure 22b skecth the temperature and heat loads of the available waste heat utility, as well as of the desired heat sink. The heat requirement of 516.3 K elevates an air stream (mass flow: 5.2 kg/s, atmospheric pressure) from 87.0°C to 185.0°C, and is key to successful heat pump design. Low-grade heat is available in by heat exchange with a cooling process, which releases 408 kW with an assumed linear temperature glide from 60.0°C to 35.0°C.

Heat transfer driving forces are set relatively high to balance the poor thermal transfer effects of the air stream. The air stream is assumed to by dry, implying that there is no water entrained in the air stream. Water is an efficient heat transfer medium with regards to heat capacity as well as heat exchange (hence its attraction as a working fluid), and decreased water content lessens a process stream's heat transfer efficiency. Thus, a pinch temperature of 10.0°C is assumed in heat transfer between the heat pump and the air stream. This equals $\sim 10\%$ of the working fluids temperature elevation, feasible compared to the statements in section 3.3 as well as to the performance evaluations of EWOS' production process.

Heat is recovered from cooling of a thermal vapor recompression system, currently performed by cold water. Flash cooling thermal vapor recompression systems condenses the working fluid (steam) quickly, which attains higher thermal efficiency than ammonia/air heat transfer. The heat pump's cold side is consequently assumed to have a minimum temperature driving force of 7.0°C (the average is inevitably larger, as ammonia is vaporized isothermally).The evaporator design is similar to that utilized for heat recovery in EWOS' processes, with ammonia tube-side and the thermal reservoir mediums shell-side. This is the common configuration in hot water heat production system as well.[51]

Ideal Operation
The high initial air temperature (87.0°C) complicates utilization of ammonia as working fluid at the heat sink. A critical temperature of 132.4°C requires a large amount of sensible heat release to elevate the air stream to its desired outlet temperature (185.0°C). High pressures and temperatures exits the compressor, which strain heat pump application. Condensing ammonia at 120°C introduces a process pinch point between the condenser and the gas cooler, where the air holds a temperature of 110.0°C.

This temperature is chosen, as it balances the systems latent heat release (elevating the air temperature from 87.0°C to 110.0.°C) to the maximum pressure (91.1 bar) to a certain extent. Lower maximum temperatures are cumbersome to explore, as the condenser temperature quickly approaches the air's inlet temperature. This decreases the potential for latent heat release, with a direct repercussion for heat recovery (decrease) and net work input (increase). Increased condenser temperatures are is strenuous as well, as it approaches ammonia's critical temperature quickly (and the maximum system temperature exceeds 200°C almost immediately).

Initial examinations assume an evaporator temperature of 39.0°C, as it imposes a maximum temperature of 195.0°C. This 10.0°C difference to the air stream's peak temperature allows the heat pump to complete the air stream's required heat treatment.

The relatively cold process pinch temperature (compared to the air's outlet temperature) complicates design of the heat pump's thermal cycle. A large sensible heat load, taking the air stream to its peak temperature of 185.0°C, decreases the latent heat load released by condensation. The system fails to condense ammonia to a saturated liquid, and a gas-liquid mixture exits the condenser for expansion to a lower pressure. The decreased amount of liquid ammonia entering the evaporator limits the potential heat recovery from the cold thermal reservoir, which increases the heat pump's required net work input. Mass flow rates are also increased to avoid pinch point problematics in the gas cooler, and the compressor's swept volume follow suit.

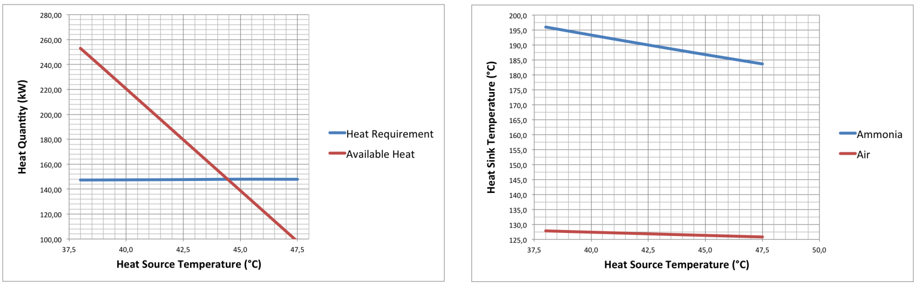
A simplified energy analysis estimates a sensible heat load requirement of 395.7 kW to heat the air from 110.0° to 185.0°C. 120.6 kW must be supplied to the air through the condenser, to heat the air from 87.0° to 110.0° before it enters the gas cooler. Super-cooling gaseous ammonia from an initial temperature of 195.0° to 120.0° with a mass flow rate of 1.003 kg/s releases of 369.5 kW, and elevate the air stream to its peak temperature.

A net work input of 281.8 kW produce the required pressure elevation from the heat pump's cold side, operating at 15.1 bar and 39.0°C. The remaining heat 234.5 kW is recovered from the waste stream, at an evaporator temperature of falling from 60.0°C to 46.0°C to abide the pinch temperature requirements.

Heat loads are extracted at maximized evaporator temperatures, supported by two main arguments. Firstly, it minimizes the net work input, due to a decreased difference between the heat pump's high and low pressure side. Secondly, it minimizes the compressor's output temperature. A large temperature lift is required to heat the air stream to 185.0°C, and decreased compressor efficiencies elevate the output temperature further. This can, and often do, strain system development as compressors are unable to deliver the necessary heat loads at sufficiently high temperatures. However, the examined system operates with a high output pressure and temperature.

These factors limit the number of market available, applicable system components, and significantly increase overall heat pump cost. An alternative system should therefore be assessed for application. The proposed solution utilizes a utility heat source to heat the air from the gas cooler outlet to the desired peak temperature, to limit the superheated temperature elevation and decrease the mechanical work input.A system design heating the air stream from 87.0°C to 110.0°C by condensing from a saturated vapor to a saturated liquid, and provide the remaining temperature elevation to 185°C by a high temperature utility, is consequently explored.

120.6 kW elevates the air stream temperature from 87.0°C to 110.0°C. A mass flow rate of 0.236 kg/s produces this heat load when gaseous ammonia release all its latent heat at a temperature of 120.0°C. As the aim of this exercise is to limit the compressor's outlet temperature, a high evaporator temperature is beneficial. Both the decreased mass flow rate and lowered maximum system temperature limits the sensible heat quantity, and an external heat utility is employed to produce the air stream's desired peak temperature. Ammonia is



(a) The heat source's effect on heat quantities

(b) The heat source's effect on heat sink temperature

Figure 26: The effects of varying heat source temperature

chosen to evaporate at 44.0°C, and impose a isentropic compression work of 60.2 kW on the thermodynamic cycle. This cools the waste heat source to 51.0°C, in accordance with the process pinch point. To arguments support the chosen evaporator temperature, supported by figure 26.

Figure 26 clearly illustrates the effects of varying the heat source temperature. (The available heat load in 26a has been shifted to the pinch point's cold side temperature, and 7.0.°C driving forces are employed throughout the analysis.) It indicates that the heat source must be cooled to 510°C to release enough heat to drive the heat pumps. A heat deficit is present if the ammonia evaporates at higher temperature than 44.0°C, and a surplus arise if it is kept colder. 147.7 kW vaporize liquid ammonia to a saturated vapor state at a temperature of 44.0°C.

Figure 26b sketches the maximum system temperatures against the cold side heat source temperature. A trade off is readily identified, where a decreased system temperature comes at the cost of a lower peak air temperature. As the heat sink temperature has a stronger influence on the internal heat pump temperature compared to the air stream, a maximized heat sink temperature is desirable to limit the thermal strains on compressor operation. This comes at the cost of a slight increase in utility heat requirements. (Real application requires a cost-benefit analysis, but this falls outside the scope of this thesis.)

The featured system requires a net work input of 60.2 kW, and compress saturated ammonia vapor from 17.3 bar to 91.1 bar and a maximum temperature of 188.1°C. The superheated vapor releases a heat load of 87.3 kW as it is cooled to the condenser temperature of 120°C, and is able to heat the air flow to a final temperature of 126.5°C. Pinch point problematics are avoided, as expected by a low mass flow rate system (with low heat transfer effects compared to the much larger air stream). A heat utility must perform the remaining temperature lift, and amounts to 308.4 kW under idealized operating conditions. Lowered strains on operation accordingly come at the cost of increased total external energy requirements, which sum to 368.6 kW.

Approximation to Realistic Conditions
Both the examined heat pumps show beneficial characteristics. Utilizing ammonia's sensible heat load to supply the entire heat load require a low work input, but present a large mass flow and a high vapor temperature after compression. There is, accordingly, a trade off between heat recovery and the maximum system temperature.

As the thesis focus on applicability as well as heat pump performance, the effects of compressor inefficiencies is examined for both systems to avoid exclusion of a potentially beneficial heat pump. Table 9 summarizes the most important thermophysical properties obtained in the cycle analysis, and quantifies the systems' differentiating performance parameters. Q_B represents the extra hot utility required to reach the air stream's peak temperature (typically produced by a boiler).

Single-Stage Compression of Ammonia TINE								
Focus: Superheating (I)	$\dot{m}$ [kg/s]	T_{MAX} [°C]	P_{MAX} [bar]	V_s [m^3/hr]	Q_C	Q_{SH} [kW]	Q_B	W_{NET} [kW]
Ideal: $\eta_C = 1$	1.003	194.7	91.1	236.1	516.3	395.7	0	281.8
$\eta_C = 0.80$	0.851	214.9	91.1	195.2	516.3	396.7	0	298.9
$\eta_C = 0.75$	0.810	221.9	91.1	184.2	516.3	369.5	0	303.5
$\eta_C = 0.70$	0.768	230.1	91.1	172.8	516.3	369.5	0	308.3
$\eta_C = 0.65$	0.725	239.6	91.1	161.2	516.3	369.5	0	313.4
$\eta_C = 0.60$	0.680	250.9	91.1	149.1	516.3	369.5	0	318.5
Focus: Condensation (II)	$\dot{m}$ [kg/s]	T_{MAX} [°C]	P_{MAX} [bar]	V_s [m^3/hr]	Q_C	Q_{SH} [kW]	Q_B	W_{NET} [kW]
Ideal $\eta_C = 1$	0.236	188.1	91.1	46.7	207.9	87.3	308.4	60.2
$\eta_C = 0.80$	0.236	206	91.1	45.3	222.9	102.3	293.4	75.2
$\eta_C = 0.75$	0.236	212.2	91.1	44.9	228.0	107.3	288.3	80.2
$\eta_C = 0.70$	0.236	219.4	91.1	44.4	233.7	113.1	282.6	85.9
$\eta_C = 0.65$	0.236	227.9	91.1	43.8	240.3	119.7	276.0	92.5
$\eta_C = 0.60$	0.236	238.0	91.1	43.2	248.0	127.4	268.3	100.3

Table 9: Single-stage mechanical compression of ammonia: Important thermophysical properties and performance parameters

5.2 A Multistage Cascade System

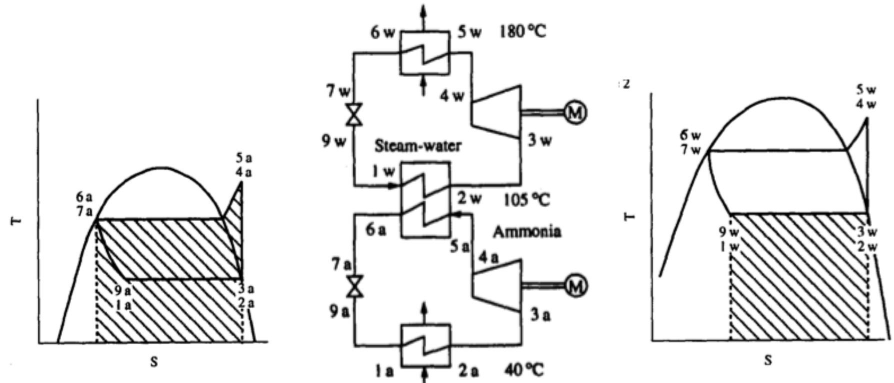
Multistage heat pumps operate as compound or cascade systems. A cascade comprises a series of single-stage mechanical compression systems, interconnected by internal heat exchangers. This facilitates use of several pure working fluids, to accommodate the thermal requirements of each compression stage. Compound systems are only operable with a single working fluid, and the thermophysical properties of most identified components commonly restrict high temperature application. To distinguish the complexities of multistage heat pump application from the limitations of singlestage mechanical compression heat pumps, cascade systems are chosen for further consideration.

Working Fluids
Steam-water and ammonia have been introduced as a binary thermal fluid in multistage cascade systems, to operate as a high temperature lift heat pump in industrial processes.[48] The original analysis features two systems: one with a power recovery expander, to take advantage of the working fluids' energy release between condenser and evaporator pressures by (isentropic) expansion of the working fluid; one using throttling valves and (isenthalpic) expansion to relieve system pressure. The latter is considered in this thesis' analysis, as the increased energetic efficiency provided by power expansion systems (reducing the heat pumps operational costs) rarely outweigh their capital cost within a reasonable payback period. Thus, the investigated cyclic arrangement comprises two standard singlestage mechanical compression systems, connected by an internal heat exchanger (ref. figure 27b).

The heat pump is characterized by its two thermal cycles. Ammonia operates the cold cycle, and receive the heat supply from the cold thermal reservoir. Steam-water is employed to operate the hot cycle, and deliver high temperature useful heat to the designated heat sink. The configuration is chosen to operate both cycles at subcritical temperatures, to reduce the maximum system pressure.

Figure 27 sketches the ideal thermal cycles experienced by the working fluids, ammonia in 27a and steam-water in 27c. The hatched areas represent equal heat quantities, and illustrate how the energy output from the ammonia sub-system serve as a heat source in the steam-water sub-system.

A brief thermodynamic description characterizes the original heat pump cycles. Ammonia is vaporized through the system's evaporator (1a−2a in figure 27a and 27b), at a temperature of 40°C. A screw compressor pressurizes the saturated gas (3a−4a), to release useful heat to the hot side steam-water working fluid at the intermediate system temperature (5a−6a). Ammonia exits the intermediate



(a) Thermal cycle ammonia

(b) Cyclic arrangement

(c) Thermal cycle steam-water

Figure 27: Cyclic arrangement and the system's temperature-entropy diagrams

heat exchanger as a saturated liquid, and expand to the evaporator's pressure (7a−9a).

The internal heat exchanger evaporates the steam-water mixture on the system's hot side to a saturated vapor at 105°C (1w−2w in figure 27b and 27c). A compression stage increases the vapor's temperature to 180°C (3w−4w), the maximal system temperature. The vapor release useful heat to a designated thermal reservoir, and exits the condenser as a saturated liquid. The high temperature throttling valve depressurizes the water to the intermediate heat exchanger temperature (7w−9w), to be reheated by the ammonia. This completes the system's full thermodynamic cycle.

The appointed temperatures mismatch the process temperature requirements (at least in EWOS' case), which complicates direct application of the proposed heat pump configuration. Certain design parameters are, however, utilized in retrofit of the heat pump's operational ranges. A thermophysical evaluation is conducted to identify suitable operating ranges, which utilize the beneficial features of both working fluids, and distribute thermal loads evenly to increase the system's overall thermal efficiency.

Heat Exchange

The low temperature cycle (ammonia) accepts heat isothermally, by vaporizing a liquid-gas mixture to saturated vapor. Heat is released in the internal heat exchanger to a steam-water mixture, initially with a temperature glide (in the superheated region), then isothermally (as it condenses to a saturated liquid). The energy transfer evaporates entrained liquid water in the steam-water mixture, which exits the intermediate heat exchanger as saturated water vapor. This increase of latent heat is an isothermal process, and the cooling of superheated ammonia should accordingly be limited to minimize the temperature driving forces (and accordingly the thermal inefficiencies) in the internal heat exchanger.

The water vapor is compressed to the maximum system temperature (not necessarily its maximum pressure), to release its sensible (in the superheated region) and latent heat (by condensing through the wet, two-phase region) to a form a saturated liquid state.

Details of the heat exchange systems are not elaborated on in the research, but assuming that shell-and-tube heat exchangers are employed is not unreasonable. Gas coolers should be used to cool the superheated vapor. Air is assumed to flow shell-side in the heat exchangers, the working fluid tube-side (as in the system presented in section 5.1). The intermediate heat exchanger probably operates with water flowing shell-side and ammonia tube-side, as this is the common configuration in heat pump hot water production when ammonia or carbon dioxide are employed as pure working fluids.[51] Separate gas coolers are applicable to increase heat transfer efficiency in this heat exchanger, but a techno-economical consideration should be performed to validate/discard it in system design.

Compression
The original research appointed screw compressors to provide the system's work input in both subsystems, based on certain characteristics beneficial in this system configuration. Firstly, screw compressors feature equal performance over a wide range of heat outputs and handle partial loads effectively. Screw compressors are also operable with the steam-water as internal lubrication, whereas oil-lubricants are required by other compressor systems. Lastly, they are capable of dry and wet compression, beneficial if superheated liquid contaminates the saturated vapor exiting the evaporators. The researchers choice is accordingly validated by three solid arguments, and screw compressors are kept in both systems.

Open systems are preferred, to simplify maintenance work. It should also be noted that the cold compression stage, increasing the pressure and temperature of NH_3, is unable to self-lubricate the systems during compression, and require oil-lubrication to cool, seal and lubricate the compressor internals.

5.2.1 Performance Analysis EWOS

The heat requirement of 383.8 kW elevates the moist air stream to its peak temperature of 110.0°C. However, as water vapor rather than ammonia is used to release useful heat to this process heat sink, different heat transfer characteristics are prominent and demand careful consideration to match the thermophysical states of the working fluid and process stream.

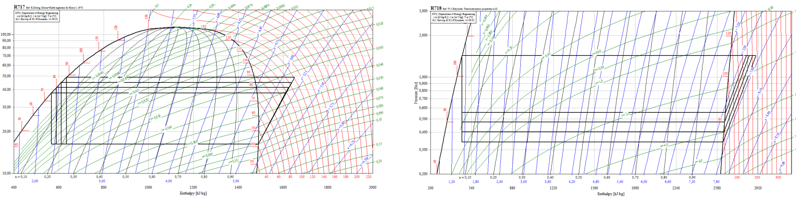
Low-grade heat is available in excess quantities, and supplied to ammonia cycling with at a thermodynamic state similar to that of the single-stage heat pump, and demand little change in system design.

Operational design with temperature driving forces of 7.5°C to the external thermal reservoirs predicts the working fluids' thermal properties at the different cyclic stages. The internal heat exchanger, where (mostly) latent heat is released from high pressure ammonia to a lower pressure gas-liquid water mixture, is commonly employed in high-pressure hot water production, and smaller temperature driving forces are feasible. The high specific heat capacity of liquid water (approximately four times that of air at the investigated temperatures and pressures) and water vapor (approximately twice that of air in the same range) support this statement as well. It is, accordingly, assumed to operate with a minimum thermal driving force of 4.0°C.

Ideal Operation

Compression of water vapor to a superheated temperature capable of elevating a moist air stream to 110.0°C, is the limiting design factor of cascade application in the given temperature range. This is due to water's high critical temperature ($T_C = 374.1$°C), which yield an energy intensive two-phase state in the scenario's operational temperature range. Cooling water vapor exiting the compressor to a saturated liquid releases a minimal amount of sensible heat, and demand a condenser temperature close to the air stream's peak temperature. Figure 28b illustrates the skewed heat release aptly.

A condenser temperature of 110.0°C is accordingly examined, where the air stream approaches a pinch temperature of 110.0°C/102.5°C as it exits the condenser. The gas cooler must release 61.5 kW to deliver the desired moist air temperature, estimated by equation (13). Performance at four intermediate heat exchanger temperature levels is evaluated, to disclose its effect on overall system performance (76°C/72°C, 80°C/76°C, 84°C/80°C, 88°C/84°C as the vaporized ammonia/water mixture temperature, respectively). The cycles are matched across figure 28a and 28b. Temperatures are chosen at an intermediate level, to distribute loads as evenly as possible between the two subsystems.



(a) Evaluated operational ranges ammonia

(b) Evaluated operational ranges water

Figure 28: Pressure-enthalpy diagrams

The heat pumps show certain beneficial features. System pressures peak as ammonia exits its compression stage, and range from 37.9 bar to 49.1 bar as the intermediate heat exchanger temperature increases from its low to high values. Net work inputs are approximately similar at ∼75 kW, and cannot merit a system selection.

Non of the identified systems are capable of producing the required sensible heat load by cooling water vapor, if the condenser cools the steam to a saturated liquid. The system operating with an intermediate temperature of 76.0°C/72.0°C produce the highest air stream output temperature (of approximately 106.5°C), as it provides the most energy intensive compression stage of saturated water vapor. If the intermediate temperature is maximized (at 88.0°C/84.0°C), the air stream's peak temperature reduces to 104.8°C. An increased average temperature driving force is also imposed on the intermediate heat exchanger, as the fraction of sensible heat increases as with ammonia's temperature lift.

Other prominent drawbacks with these operational modes arise as well. Superheated water vapor temperature rises rapidly with increased compressor work input, and a maximum system temperatures range from 175.3°C to 214.7°C as the intermediate heat source temperature decreases. A lowered system pressure and negligible increase of the outlet air stream temperature is accordingly made with an extensive trade off against the maximum system temperature, as illustrated by figure 29. A high intermediate temperature level is accordingly chosen, to limit the maximum operational temperature as compressor inefficiencies are introduced.

Examinating the performance of a heat pump designed to operate at an intermediate temperature of 88.0°C/84.0°C and a condenser temperature at 110.0°C, shifted to release a sensible heat load elevating the air stream to its desired peak

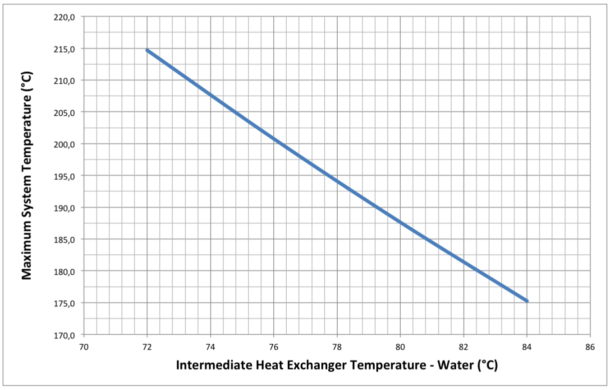


Figure 29: Maximum system temperature at increased intermediate temperatures

temperature, reveal an unfavorable solution. It requires a significantly increased mass flow rate, more than thrice the size of the originally proposed solution, and the net work input is increased proportionally (to 80.8 kW). Component sizes, and particularly compressor capacities, require an significant upscale and added component cost.

A high intermediate temperature cascade, taking advantage of its latent heat release, is accordingly beneficial from an energetic point of view, which is the scope of this thesis. Additionally, as a boiler system is used to elevate the air stream from 110.0°C to 130.0°C regardless of the applied heat pump performance, an additional 42.5 kW supplied of its bottom line is a valid settlement to the potential increase of compressor costs. These conclusions are drawn on ideal compression of the working fluid, and the required additional heat supply is expected to decrease as compressor inefficiencies arise.

Approximation to Realistic Conditions
The initially investigated, low flow rate system is examined further with the applied compressor inefficiencies. Thermodynamic properties and performance estimations are summarized in table 10. A multitude of system configurations exits (as the subsystems are operable with differentiated compressor efficiencies, a probable case as different working fluids are employed with vastly different swept volumes), but employed compressors are assumed to have equal efficiencies to limit the number of sampled systems. The decision is supported by application of equal compressor types (screw) in the subsystems.

Cascade Compression of Ammonia and Water EWOS								
Ammonia: 41.5°C | 88°C	$\dot{m}$ [kg/s]	T_{MAX} [°C]	P_{MAX} [bar]	V_s [m^3/hr]	Q_C	Q_{SH} [kW]	Q_B	W_{NET} [kW]
Ideal: $\eta_C = 1$	0.311	131.0	49.1	52.6	316.2	N/A	N/A	50.2
$\eta_C = 0.80$	0.299	142.9	49.1	48.6	316.2	N/A	N/A	60.3
$\eta_C = 0.75$	0.295	147.0	49.1	47.3	316.2	N/A	N/A	63.5
$\eta_C = 0.70$	0.291	151.7	49.1	45.9	316.2	N/A	N/A	67.1
$\eta_C = 0.65$	0.286	157.3	49.1	44.4	316.2	N/A	N/A	71.1
$\eta_C = 0.60$	0.281	164.0	49.1	42.6	316.2	N/A	N/A	75.6
Water: 84°C | 110°C	$\dot{m}$ [kg/s]	T_{MAX} [°C]	P_{MAX} [bar]	V_s [m^3/hr]	Q_C	Q_{SH} [kW]	Q_B	W_{NET} [kW]
Ideal $\eta_C = 1$	0.145	175.3	1.4	781.8	341.3	19.1	42.5	25.1
$\eta_C = 0.80$	0.145	197.0	1.4	744.3	347.6	25.4	36.2	31.4
$\eta_C = 0.75$	0.145	204.2	1.4	731.9	349.7	27.5	34.1	33.4
$\eta_C = 0.70$	0.145	212.5	1.4	717.7	352.1	29.9	31.7	35.8
$\eta_C = 0.65$	0.145	222.1	1.4	701.3	354.8	32.6	29.0	38.6
$\eta_C = 0.60$	0.145	233.3	1.4	682.3	358.0	35.8	25.8	41.8

Table 10: Multistage mechanical compression of ammonia and water: Important thermophysical properties and performance parameters

5.2.2 Performance Analysis TINE

Process integration with the dry air flow's steep temperature elevation from 87.0°C to 185.0°C at the hand of a 516.3 kW heat supply is key to create a advantageous heat recovery system. Low temperature heat extraction (below 60°C) from the waste heat source is performed by ammonia, whereas the water vapor release heat to the TINE's hot process stream. Both are operable with full or partial loads, depending on the chosen system characteristics. Heat transfer driving forces are set relatively high (10.0°C) to balance the poor thermal transfer effects of the air stream. A 7.5°C driving force is employed in the cold sub-system (due to the previously discussed beneficial effects, ref. section 5.1). The intermediate heat exchanger is still assumed to operate with a 4.0°C driving force.

Water is an efficient heat transfer medium, know to release large quantities of latent heat in the investigated temperature range. The superheated temperature rises quickly as water enters a gaseous state, and an examination is conducted to identify a hot sub-system operational range to favor maximum system temperature (expected to rise in the hot cycle) and pressure (expected to form in the cold sub-system). This avoids superfluous driving forces, which has the potential to skew the results compared to pure ammonia systems (as they are capable of longer temperature glides during heat transfer).

TINE's process is quite similar to that presented by the paper initially investigating the performance of this cascade system, and their suggested water vaporization temperature of 105.0°C and 74.3 bar is accordingly chosen as a vantage point in the ensuing performance analyses.

The hot side water vapor is expected to most strenuous in application, due to its extensive latent heat potential. Allowing the water to condense at a midrange temperature (140.0°C) provide a suitable vantage point, to determine its thermal elevation capacity when coupled with the process heat sink.

Ideal Operation
Compressing saturated water vapor from a temperature of 105.0°C reveal a rapid increase of the system's maximum temperature. A peaking at 349.0°C achieved when the condenser temperature is held at 180.0°C. Figure 30 is an apt illustration of this effect, which encourages system designs with a relatively low condenser temperature.

Lowering the water's vaporization temperature has a similar effect. A maximum temperature of 244°C is required to condense at 140.0°C, if water vapor is compressed from a saturated state exiting the intermediate heat exchanger at 96.0°C. The temperature increase is a trade off against maximum system pressure (from the initially assumed 74.3 bar to 62.5 bar), at which ammonia condenses to the recovered heat at intermediate temperature. Overall work requirements decreases as well, but by a negligible amount compared to the initially assumed system design.

Water vapor is accordingly deemed to operate a thermodynamic cycle with isothermal heat exchangers kept at 105.0°C and 140.0°C, to relieve the maximum system temperature. System pressure ranges up to 3.6 bar, and requires a swept volume of 491.9 kg/hr. The mass flow rate is held at 0.105 kg/s, to condense the vapor to a saturated liquid through the condenser, and yield a net work requirement of 22.6 kW.

The maximum system pressure arises in the cold sub-system, where ammonia is compressed to a 74.3 bar from a saturated gas state at 41.5°C (to abide the pinch point requirements). An extended compression ratio and increased flow rate elevates the net work input to more than twice that applied the high temperature sub-system, at 56.5 kW. The cold sub-system temperature peaks at 170.5°C.

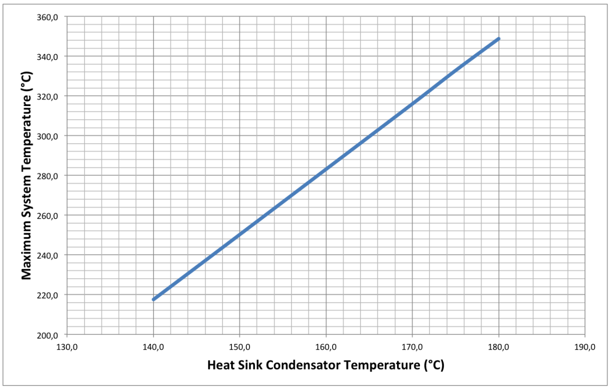


Figure 30: Maximum system temperature at increased condenser temperatures

A prominent performance parameter is the limited superheat energy transfer of 17.3 kW when the heat pump operates isentropically. This design feature minimizes temperature driving forces during heat transfer, and limits is available temperature lift as well. A peak air temperature of 132.6°C is delivered by the proposed heat pump design, which leaves an external heat utility to provide the remaining temperature elevation (an additional 273.3 kW). This yields a net energy input requirement of 349.6 kW in the hot air production.

A complete temperature elevation is possible with a condenser temperature of 140.0°C, as the maximum system temperature far exceed the air stream's final temperature of 185.0°C (at 217.4°C with an idealized compression stage). This is achieved by increasing the mass flow rate, which has a direct effect on the system's sensible heat release. It does, however, demand an additional work input which is scaled proportionally (refer to equations (13) and (14) and their dependence on mass flow rate). This shifts the thermodynamic cycles to carry larger sensible heat loads, to the right of the bell-curve sketched in figure 27. As the scaling is readily introduced after accounting for compressor inefficiencies as well, and these effects on the thermodynamic cycles are accordingly investigated first to simplify system design. An increased flow rate's influence on compressor swept volumes cannot be forgotten either, and must be accounted in such examinations.

Approximation to Realistic Conditions

The approximations to realistic circumstances are examined under the restraints of the initial system configuration, operating with a cold ammonia sub-system recovering heat at 41.5°C and evaporating a gas-liquid water mixture at 109.0°C at a maximum system pressure of 74.3 bar. Water vapor exits this heat exchanger as a saturated gas at 105°C, and condenses at 140°C.

Performance parameters and prominent thermodynamic parameters are summarized in table 11, and modifications are presented as discussion goes, if deemed necessary.

Cascade Compression of Ammonia and Water TINE								
Ammonia: 41.5°C | 109°C	$\dot{m}$ [kg/s]	T_{MAX} [°C]	P_{MAX} [bar]	V_s [m^3/hr]	Q_C	Q_{SH} [kW]	Q_B	W_{NET} [kW]
Ideal: $\eta_C = 1$	0.232	170.6	74.3	47.1	220.4	N/A	N/A	53.7
$\eta_C = 0.80$	0.218	187.3	74.3	43.0	220.4	N/A	N/A	63.2
$\eta_C = 0.75$	0.214	193.1	74.3	41.8	220.4	N/A	N/A	66.2
$\eta_C = 0.70$	0.210	199.9	74.3	40.4	220.4	N/A	N/A	69.4
$\eta_C = 0.65$	0.205	207.8	74.3	38.9	220.4	N/A	N/A	73.0
$\eta_C = 0.60$	0.199	217.2	74.3	37.2	220.4	N/A	N/A	77.0
Water: 105°C | 140°C	$\dot{m}$ [kg/s]	T_{MAX} [°C]	P_{MAX} [bar]	V_s [m^3/hr]	Q_C	Q_{SH} [kW]	Q_B	W_{NET} [kW]
Ideal $\eta_C = 1$	0.105	217.4	3.6	491.9	243.0	17.3	273.3	22.6
$\eta_C = 0.80$	0.105	243.6	3.6	478.7	248.6	23.0	267.7	28.3
$\eta_C = 0.75$	0.105	252.3	3.6	474.3	250.5	24.9	265.8	30.1
$\eta_C = 0.70$	0.105	262.3	3.6	469.3	252.7	27.0	263.7	32.3
$\eta_C = 0.65$	0.105	273.9	3.6	463.5	255.2	29.5	261.2	34.8
$\eta_C = 0.60$	0.105	287.3	3.6	456.8	258.0	32.4	258.3	37.7

Table 11: Multistage mechanical compression of ammonia and water: Important thermophysical properties and performance parameters

5.3 A Hybrid System

Hybrid systems are developed to take advantage of the beneficial heat transfer effects of sorption processes between a volatile fluid and a heavier solution. As most mechanical compression systems, hybrid heat pumps operate with a high and low pressure side, separated by the compressor and solution pump (elevating pressures of the working fluid and the absorbent, respectively) and the expansion valve (lowering the rich solution's pressure). Figure 31 illustrates the typical hybrid system, in its simplest state. A separator (not included in the figure) is normally employed after the desorber, to improve separation of the vapor and lean solution exiting it and ensure sound operational conditions for the compressor (dry compression is preferred to reduce wear and tear).

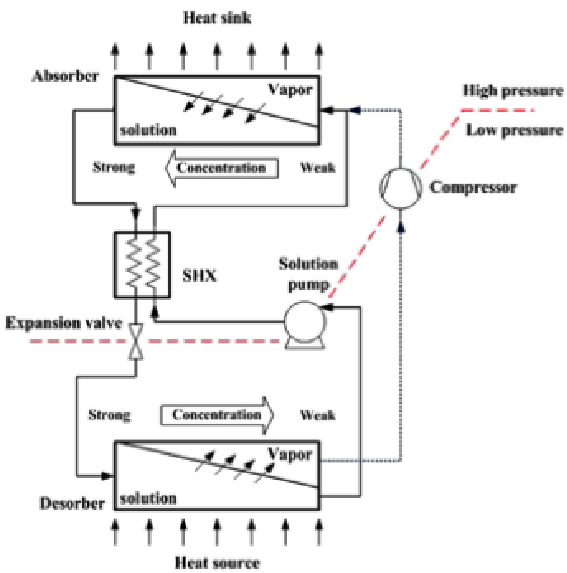


Figure 31: Hybrid cycle arrangement

Most research evaluates ammonia/water mixtures, and systems are already commercialized for hot water production.[59] The hot water output temperature ranges from 75-110°C, produced with a temperature glide of up to 35°C. Waste heat is extracted by the heat pump, and cooled from 15-65°C to 2-60°C. Comparable output temperatures are obtained with transcritical CO_2 heat pumps, but system pressures are vastly different. The hybrid heat pump in question is operable at pressures below 25 bar, whereas CO_2 systems quickly exceeds 100 bar as temperatures increase. Decreased system pressures lower the threshold for application of standard industrial components, a beneficial feature in commercialization of new developments.

Available temperature glides depend on the gas/liquid ratios across the heat exchangers (on both the high and low pressure side), and increase with heat exchanger lengths. A notable system requirement is absorption/desorption of equal mass fractions, to sustain a regenerative thermodynamic cycle. Different thermodynamic characteristics in high and low temperature sorption processes do, however, support unbalanced thermal glides if necessary.

A current research project evaluates the reliability of an ammonia/water mixture in hybrid heat pumps releasing heat at 180-250°C.[60] It aims to construct a heat pump based on newly developed standard components, capable of high operating pressures. The research comprises both theoretical and experimental investigations, and a demonstration of the developed system with a potential industrial end user. A project budget of DKK 12.14 million has been appointed to the project, funded by governmental grants and industrial participants, indicative of the potential hybrid heat pumps are believed to hold in industrial application.

Working Fluids

Ammonia/water mixtures are the prominent working fluid in hybrid heat pumps, used in commercialized systems and researched for future application, and consequently chosen for evaluation in this thesis.

Ammonia is the more volatile component, vaporized in low-pressure desorption and resorbed at high pressures. Water remains an aqueous solution, with a designed amount of entrained ammonia. The favorable features of both fluids (ref. section 2.4.3) combine, and lessen the restraints of individual application of the two components. Ammonia is known to operate with high saturation pressures as the condenser temperature increases, but the effects of high temperature absorption decrease the pressure requirements. And as water is pumped to higher pressures rather than compression as a dry, superheated vapor, hybrid systems evade the challenges of large volumetric ratios and swept volumes in its compression stages.

Figure 32 [16] illustrates the pressure/temperature relation of an aqueous solution with variable amount of entrained ammonia. It indicates the benefits of introducing a secondary component to hybrid heat transfer processes, and the effects of varying component concentrations. Ammonia is absorbed with a heat release at P_a (illustrated by the red left-pointing arrow), and decreases the mixture's temperature. Temperature glides are accordingly manipulated by controlling the absorption rates through the heat exchanger. P_c indicates the

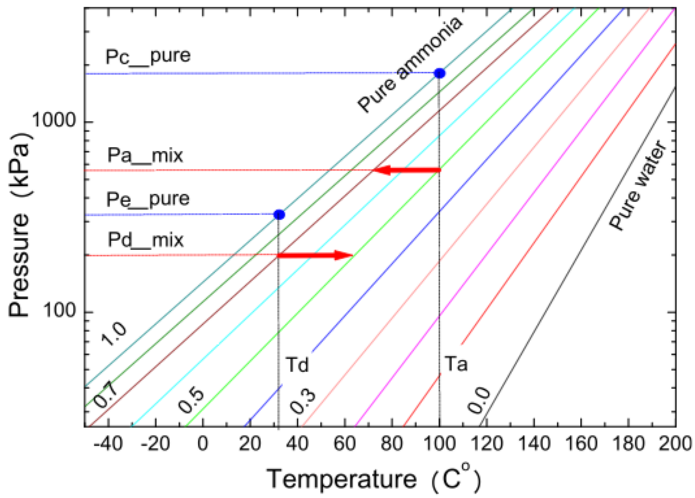


Figure 32: Pressure levels of a hybrid cycle compared to a pure vapor cycle

required pressure levels of a pure ammonia heat release at similar temperatures, an the significant maximum system pressure is apparent. The system's pressure difference to the desorber (evaporator) level is also reduced, which affects the overall system efficiency when compressor and throttling losses are introduced (figure 32 has logarithmic pressure axis). Compressing/depressurizing the working fluid follow the indicated concentration lines, and the achieved heat transfer is accordingly a function of the isobaric heat temperature changes. Design for high temperature application is achieved by controlling ammonia/water concentrations during operation (horizontal shifts in figure 32), and capacity control is achieved by manipulating the saturation pressures (vertical shifts).

It is, however, also evident that mixtures must approach lean solutions (a lean water solution, low on ammonia) or higher pressure levels, for systems to approach the requirements of high temperature application.

Heat Exchange
The varying concentrations affect heat transfer as well as system pressures, and temperature glides are present in the two-phase heat exchangers. This is known as the lorentz effect, which was introduced in section 3.1.2. The evaporator is often assumed to fully desorb ammonia from the water, and the vapor enters the compressor in a saturated state.

Shell-and-tube heat exchangers are favored over compact heat exchangers, as experience with water/ammonia mixtures in compact heat exchangers is scarce.[18] The working fluid flows at the tube-side of the heat exchanger, and face the challenges introduced by pure working fluid heat pumps (liquid-vapor concentrations, etc) to a greater extent. Plate-exchangers are, however, also applicable.

Pressurized ammonia and water are typically mixed before they enter the heat exchanger counter-current to the secondary heat exchanger fluids (in our case the process air streams). Both falling film heat exchange and bubble type absorbers are applicable, and have been investigated experimentally.[16]. The falling film type lets gravity drive the working fluid from top to bottom during heat exchange, and is an efficienct means of controlling heat exchange.

Compression
As ammonia vapor exits the desorber via a separator, dry compression is a feasible assumption. Dynamical compressors are preferred, as in section 4.4.1, and both reciprocating and screw compressors are applicable. Reciprocating compressors are often chosen for laboratory scale testing (due to the volumetric compression rates).[18] Screw compressors are, however, equally applicable and can be favorable when system size increases. Compound compressor systems are also applicable, as ammonia is assumed to compress in a pure, gaseous state.

The lean solution (mostly water, but some ammonia vapor remain entrained after separation in the desorber/evaporator) is pumped to the high pressure required in the absorber. Cavitation is a potential problem, and the pump must cope with small amounts of entrained vapor.

5.3.1 Performance Analyses

Heat exchange with dual working fluid components are challenging to estimate, due to the complexities introduced by boiling point trajectories, bubble points and other fluid mixture effects. Estimating the mixture's thermophysical properties at different states in the regenerative heat pump cycle is accordingly difficult, and cannot be achieved within a reasonable margin of error (comparable to that of the pure working fluid calculations) unless extensive numerical simulations are employed.

A distinct performance analysis has not been conducted for hybrid heat pumps, as the lack of thermodynamic parameters render the calculation tool developed in section 3.2 inapplicable. However, an indication of its operable temperature and pressure levels is provided, based on the single-stage mechanical compression of ammonia (section 5.1) and rough estimations drawn from figure 32.

The obtained results are not indicative of overall energy consumption (and accordingly the energy efficiency compared to the current utilities), but provide an estimate of the ammonia's vapor pressures and temperatures at the in- and outlet of the compression stage. This sets bounds for its energetic performance, upwards by pure ammonia, single-stage compression, downwards by the idealized, thermally gliding COP net work estimations (ref. table 6).

5.4 Promising Research

The preliminary selection nominates mechanical compression heat pumps as most-likely-to-succeed systems in the identified industrial scenarios. Investigation of their performance, accounting for the appointed working fluid's influence of on compression and heat transfer, have been performed in sections 5.1-3.

Thermally driven heat pumps also show certain beneficial characteristics in the initial performance analyses, applicable in the investigated temperature ranges. Their basic working principles do, however, require an increased heat supply compared to the currently utilized heat loads (although at a lower temperature) and categorize them with manageable applicability. This omits thermal heat transformers from the main performance analysis, alongside the scarcity of applicable (and available) scientific research and necessity of complex simulation tools.

Their potential is, however, undisputable as they can produce high temperature heat with a negligible amount of external energy input (compared to mechanically driven systems) if suitable waste heat reservoirs are available. Two currently researched systems are therefore described, to illustrate ongoing heat pump developments.

5.4.1 An Absorption Heat Transformer

Working Fluids
Commonly investigated high temperature absorption heat transformers use aqueous lithium-bromide mixtures as their working fluid. Pure water act as the working fluid, absorbed and desorbed by an aqueous solution of the lithium-bromide (LiBr) salt. A numerical investigation of system operation with heat release to industrial hot water production at 130°C, utilizing waste heat at 90±2°C, is applicable to the scenarios at hand.[40]

The heat pump's working principle is sketched in figure 33 and bear a striking

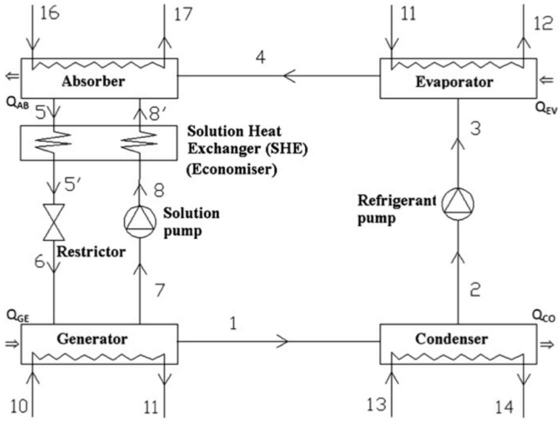


Figure 33: Schematic of the absorption heat transformer cycle

resemblance to the hybrid heat pump cycle. Aqueous LiBr-solution is circulated between the generator and absorber. It enters the generator in at low pressure and in a lean state (low LiBr-to-water content, state 6 in figure 33), where medium grade waste heat vaporize water from the solution. A stronger solution (with an higher LiBr-to-water ratio), is pumped back $(7 - 8)$ to the high absorber pressure. Desorbed water vapor, state 1, flows through the condenser, driven by pressure differentials, where it condenses to release low grade heat to its environment. Liquid water is pumped to the absorber pressure $(2 - 3)$, and evaporated $(3 - 4)$ by the medium grade heat supply in the evaporator. As water vapor enters the absorber, it is met by the strong aqueous LiBr-solution. High temperature heat release originates from the absorption of high-pressure water vapor in the strong LiBr-solution. A weak, aqueous LiBr-solution exits the absorber, the fifth thermodynamic state in figure 33. An economizer is employed to extract the weak solution's remaining energy $(5 - 5')$, and preheat the high-pressure strong solution entering the absorber $(8 - 8')$. It reduces the solution pump's energy requirements, and lowers the weak solution's temperature to generator levels. A restrictor $(5' - 6)$ depressurizes the weak solution, and it enters the generator at low pressure.

Absorption heat transformers operate with a high-pressure side, held at to the evaporator's saturation pressure, and a low pressure side held at to the condenser's saturation pressure. The pressure differences are produced to operate a regenerative thermodynamic cycle at two temperature levels. Exothermic processes (absorption and condensation) are beneficial at high temperature, whereas the endothermic processes (desorption and evaporation) are desirable at low temperatures, to utilize available waste heat. However, only a fraction of the supplied heat quantities are available at high temperature if extensive, external energy requirements are to be avoided. The condenser is accordingly run at a minimum system temperature, to facilitate high temperature heat release in the absorber (which has superior heat transfer capabilities).

The liquid refrigerant differentiates the absorption heat transformer from hybrid heat pumps, and requires pressurization by a pump rather than a compressor stage. This reduces the required work input and certain operational constraints (such as high system pressures). Liquefying the working fluid requires two heat transfer stages, low grade energy is removed to condense the fluid, and higher grade energy is used to vaporize it. This additional thermal input compensates the decreased mechanical work requirements.

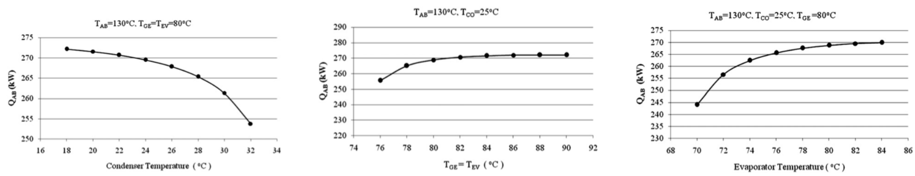
Heat Exchange
Heat exchange in absorption heat pumps is similar to previously described systems. Pure water is condensed and vaporized at constant temperatures, and exits both heat exchangers in a saturated state (liquid and gaseous, respectively). Absorption/desorption processes operate with thermal glides, created by the mixture's continuously changing component ratio. However, the study assumed these to operate at constant temperatures. Heat exchangers are not specified, but standard shell-and-tube systems are applicable and often utilized in absorption heat pumps.

The research's numerically obtained results showed that the condenser temperature relates to the minimum system pressure, and that increased condenser temperatures increase the minimum pressure levels (and affects the generator, which is kept at the evaporator pressure). Elevating generator pressures decrease water desorption, and a higher flow rates are required to sustain the transferred heat quantities. This decreases the absorber's heat capacity, and lower's system COP. The condenser should therefore be kept as cool as possible (it is still restrained by the heat pump's environment and the thermophysical properties of water).

It was also showed that system performance benefit from higher evaporator temperatures, as this increases the maximum system pressure. More water is absorbed by the strong solution at higher pressures, and allow reduced system flow rates. This increases system COP.

Another beneficial influence on system performance are the temperatures held in the evaporator and generator. If the evaporator temperature exceeds the generator temperature, the system's pressure difference increases and higher maximum pressures are achieved. This increased pressure difference reduces the flow rate requirements, and enhance performance. The investigated system is therefore designed to supply waste heat in the evaporator first, and then release its remaining energy potential in the generator. A similar design should be pursued if implemented in either of the identified scenarios, as the proposed intermediate temperature stream will operate with a thermal downgrading in heat transfer.

These three effects are represented graphically in figure 34a, 34b and 34c, respectively. Heat quantities in figure 34 are not directly transferable to the scenarios at hand in this paper, as the analysis evaluates hot water production rather than heating of air. The trends are, however, applicable to the heat transformer performance, and accordingly valuable to the ensuing analyses.



(a) Variable condenser temperature (b) Variable evaporator and generator temperature (c) Variable evaporator temperature

Figure 34: Absorber heat capacity against heat exchanger temperatures

These results coincide with the preliminary performance analysis in chapter 4, which evaluated heat transformer performance based on the thermal reservoirs' temperatures alone. The research uses an extended numerical model to obtain component heat capacities and evaluate system performance, which increases the accuracy significantly. This validates the calculation tool supporting the results of the preliminary selection, and confirms the concluding remarks as an applicable foundation for further investigations.

Compression

Solution pumps produce the system's pressure differential, as heat exchange is assumed isobaric for all systems evaluations. Liquid water is pumped to higher pressures without problems. Strong aqueous LiBr-solutions contain entrained salt particles, but the LiBr-salt's extreme hygroscopic character renders use of standard solution pumps applicable. The work performed by these pumps is minute compared to the heat transfer rates, and accordingly excluded from the energetic performance analyses conducted in the papers numerical investigation.

Crystallization is a potential challenge, if the aqueous solution is too strong. This is, however, not a common encountered, as strong solutions decrease system performance as well, and accounted for in heat pump design.

Table 7 summarizes the most important thermodynamic quantities of the investigated absorption heat transformer, operating with thermal reservoirs most similar to the identified industrial scenarios of this thesis.

Thermodynamic Quantities		
	Unit	Quantity
T_{AB} (Absorber Temperature)	°C	130
T_{CO} (Condenser Temperature)	°C	25
T_{GE} (Generator Temperature)	°C	73
T_{EV} (Evaporator Temperature)	°C	80
T_{WGE} (Waste Water Outlet Temperature, Generator)	°C	75
T_{WEV} (Waste Water Outlet Temperature, Evaporator)	°C	82
Flow Ratio		18.63
Water Vapor Flow Rate (STEP 1 – 4)	kg/hr	807
Q_{AB} (Absorber Heat Transfer)	kW	486.91
Q_{CO} (Condenser Heat Transfer)	kW	556.8
Q_{GE} (Generator Heat Transfer)	kW	495.58
Q_{EV} (Evaporator Heat Transfer)	kW	558.13
Q_{SOL} (Solution Heat Exchange)	kW	337.11

Table 12: Performance parameters of an absorption heat transformer

The flow ratio refers to the ratio of strong solution flow rate to the desorbed water vapor mass flow (ref. figure 34a). It is an important design and optimization parameter, which greatly affects system heat transfer quantities. And as heat exchanger pressures and temperatures are kept constant, the working fluid's mass flow dictates heat transfer rates. The estimated heat transfer quantities are accordingly manipulated by altering the system's mass flow rates in accordance with equation (13). Heat transfer rates obtained through the research's simulations are accordingly applicable as highly simplified results in the investigated scenarios (as deviations from the available thermal reservoir temperatures are unaccounted for), but give an indication of the intermediate temperature, external heat requirements.

The absorption heat pump elevates EWOS' moist air stream to its peak temperature of 110°, with a minimum thermal driving force of 20°. This requires a thermal input of approximately 830.5 kW and a lean solution mass flow rate of 12 500 kg/hr entering the generator. Pump work has been neglected. An improved thermal efficiency should be pursued to make this heat pump an applicable alternative, as the current heat utility is less than halve at idealized boiler efficiency (Q_B =383.8 kW), and 318.8 kW less with the imposed boiler efficiency of 0.75 (Q_B =511.7 kW).

The tendency is seen in application with TINE's hot air production as well, where the absorption heat pump produces less than half of the required temperature lift with significantly increased energy consumption.

Both processes would accordingly experience significantly increased consumption rates by installing this heat pump. And although it comes at a significantly lower temperature, the cost reduction is unlikely to balance the quantitative cost elevation. It is, accordingly, neglected from the ensuing discussions of energy efficiency in section 5.5.

5.4.2 Chemical Reaction Systems

Mechanical heat pumps are limited in high temperature industrial application by the extensive pressure ratios across their compression stages. Elevated pressure ratios lower the system's energetic performance, and increase both capital and operational costs of implemented components. Thermally driven sorption heat transformers also depend on large pressure differences between their components, which affects operational costs and maintenance costs negatively. These problems can be evaded if heat pumps are based on reversible reactions, and several chemical reactions are suitable for heat pump application. However, studies of such heat pump systems are limited, and most are based on theoretical and experimental investigations. A system is nevertheless introduced, to highlight the potential these systems may hold in the future.[63]

Isopropanol-Acetone-Hydrogen as Working Fluid
The most prominent chemical reaction is the based on an isopropanol-acetone-hydrogen system, due to a high upgrading temperature, energy storage potential and predictable application. Most of these systems are evaluated as heat amplifiers, releasing useful heat at an intermediate temperature [62, 63], but it is also capable of operation as a heat transformer [32]. The reversible reaction introduced below. ΔH is the reaction enthalpy, positive in endothermic reactions, negative in exothermic reactions.

Dehydrogenation of isopropanol:
A) $(CH_3)_2CHOH_{(g)} \longrightarrow (CH_3)_2CO_{(g)} + H_{2(g)} \qquad \Delta H_{deh} = 54.4 kJ/mol$

Hydrogenation of acetone:

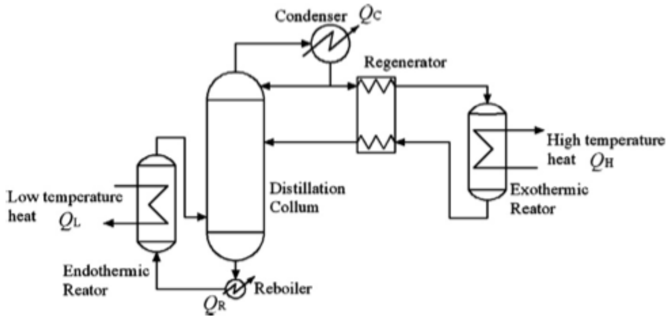


Figure 35: An isopropanol-acetone-hydrogen heat pump

B) $(CH_3)_2CHOH_{(g)} \longleftarrow (CH_3)_2CO_{(g)} + H_{2(g)} \qquad \Delta H_{hyd} = -54.4 kJ/mol$

Reactions A and B are really a chemical equilibrium, pushed towards dehydrogenation of isopropanol by a heat supply from an source at intermediate temperature (Q_L). A partial heat load (Q_H) is released at higher temperature by the exothermic reaction regenerating isopropanol. The remaining part is rejected at low temperature (T_C). Figure 35 [63] is a schematic of the heat transformer system.

Isopropanol is dehydrogenated in an endothermic reactor, driven by the heat pumps main heat input. The reaction is in gas or liquid phase, depending on the reboiler temperature (gas phase if $T_L > 82.4°C$, liquid otherwise). Gas phase is most commonly studied.[63] Acetone, hydrogen and unreacted isopropanol is fed back to the distillation column as an equilibrium mixture at T_L. A acetone/hydrogen gas mixture is fed to the exothermic reactor (with some residual isopropanol still entrained from the distillation column), and reacts to equilibrium with isopropanol at T_H. Heat transfer is assumed to be isothermal in both reactors, and molar quantities reacting must balance to facilitate system regeneration. Ruthenium-platinum (Ru-Pt) usually catalyze the endothermic reaction, whereas nickel (Ni) is used to catalyze the hydrogenation in R2.[4]

The distillation column is driven as a heat engine by a different intermediate temperature heat supply, Q_R, slightly colder than Q_L. Low temperature heat is released by a partial condensation of the acetone/hydrogen leaving the distillation column, to minimize the residual isopropanol sent to the exothermic reactor. This is necessary to avoid high temperature dehydrogenation of isopropanol, which takes place if the equilibrium balance is shifted. Pure isopropanol separated in the column is returned to the endothermic reactor, alongside the regenerated isopropanol fed to the separator from the hydrogenation chamber.

A proposed system [32] operates with a reboiler temperature of ∼83°C (the boiling point of pure isopropanol) and a condenser temperature of ∼56°C (the boiling point of pure acetone). Performance evaluations are conducted with temperature differences of 10, 20 and 30°C separating the endothermic and exothermic reactor. Variable amounts of residual isopropanol in the exothermic reactor are also examined. The useful heat release was investigated at temperatures ranging from 120°C to 250°C.

The achieved thermodynamic efficiency is greatly affected by internal entropy production in both chemical reactors, and performs at less than half of the predicted carnot efficiency (∼0.55-0.825). COPs range between ∼0.22 and ∼0.325, depending on the entrained amount of isopropanol entering exothermic reactor, and on internal temperature differences.

Increasing the thermal output temperature and/or the entrained amount of isopropanol fed to the hydrogenation stage affected system performance positively. The former is explained by the fixed temperature lift between the chemical reactors, and the increased efficiency of operating with higher temperature heat sources. The latter is explained by a decreased conversion rate of acetone and hydrogen, as equilibrium is achieved faster when the entrained amount of isopropanol increases (which decreases entropy production). Decreasing internal temperature differences also had a positive effect on system performance, illustrated by the simplified equations developed in chapter 3, as the driving heat source approaches output temperatures.

Other Working Fluids
Studies of other working fluids applicable in heat transformer systems also show promising analytical results. Solid sorption ammonia systems are particularly attractive [64], and utilize a combination of thermal heat generation and mechanical compression to release high temperature heat with an intermediate temperature heat supply. Figure 36 represents a typical system, where the investigated system situate its mechanical compressors between the high and low temperature salts, to increase the vapor pressure of ammonia during loading processes. The heat pumps typically use magnesium-chloride ($MgCl_2$) as the high temperature salts and calsium-chloride ($CaCl_2$) as the low temperature salts.

The analytical investigations, based on available adsorption/desorption data and system pressures, estimates that a system supplied with waste heat ranging from

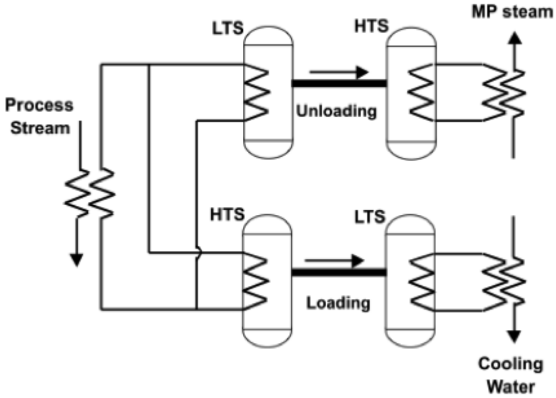


Figure 36: An generic solid-vapor sorption heat pump

60.0-100.0°C can release a partial heat quantity to a designated heat sink at 130.0-200.0°C. The remaining heat is discharged to the heat pump's environment at 30.0-50.0°C. The compressors increase the system's potential temperature lift from its intermediate heat source temperature to its high heat sink temperature, by increasing the internal pressure differences.

Unfortunately, the study fails to address some critical aspects. Heat transfer is assumed to be isothermal in the analysis, whereas real systems are prone to experience temperature glides as sorption processes progress. Heat pump systems based on salt pairs require regeneration, as absorption/desorption are batch processes. The challenges of batch production are evaded by employing multiple reactor systems, but application and combination with continuously operating compressor systems require further investigations. Problems related to salt reaction irreversibilities, and thus a system degeneration, must also be addressed.

Current research investigate these topics, both for the solid sorption system[64] and the isopropanol-acetone-hydrogen heat pump [32]. Focus is, however, currently on analytical development, and experimental evaluations of heat pump performance are scarce. Analyzing either system further will accordingly fall outside the scope of the thesis, is valuable performance parameters are to be obtained. Both are consequently excluded from the ensuing discussions. A presentation is nevertheless considered valuable, as applicability seem to be restricted by system development rather than by singular components, and developments are considered in attractive operational temperature ranges.

5.5 Discussion

Section 2.2 introduces a wide range of heat pump working principles with potential applicability in high temperature industrial heat recovery. A two-step appoints their applicability within the identified industrial scenarios, initiated by a preliminary investigation based on process requirements, and finalized by a thermodynamic evaluation of specific performance parameters. The examinations treat a specter of thermodynamic cycles, ranging from ideal operation to fairly large operational losses (both thermal and mechanical). Table 6, 8, 9, 10 and 11 provides the key inputs.

Mechanical compression heat pumps proved most applicable, and two systems were introduced. First, a single-stage mechanical compression of ammonia, operating with a subcritical thermodynamic cycle. Secondly, a mechanically driven cascade heat pump, with ammonia and water as a binary working fluid. Water operates the hot sub-cycle in a single-stage compression cycle, ammonia the cold with the same design. A display of the heat pumps' applicability as waste heat recovery systems in EWOS' and TINE's production processes ensue, including a presentation of market available components with favorable design features.

Several other systems were also described, but not evaluated analytically due to complex working principles and extensive simulation requirements to approach feasible margins of error in the performance analysis. A hybrid heat pump, using sorption processes between its two-component fluid (water and ammonia) during heat transfer, and a single pressurizing stage, is most prominent. (Its complexity is aptly illustrated by an aforementioned research project currently undertaken in Denmark, ref. section 5.3) Favorable features of the hybrid heat pump are nevertheless emphasized as market available components are considered for application.

5.5.1 EWOS

Heating EWOS' moist air flow is key to produce the warm, dry air used in pellet production. A heat recovery system is already applied at low temperatures, and takes advantage of heat release from dehumidification of the circulated process stream. However, a peak outlet temperature of 63.3°C requires an external boiler system to elevate it to the drier inlet temperature at 130.0°C. Physical restrictions prevent a potential heat recovery system to supply heat above 110.0°C, which the air moist stream requires an ideal heat input of 383.8 kW to achieve this temperature elevation. A waste stream was identified, able to supply low grade heat in excess quantities to a potential heat recovery system.

Performance coefficients developed in section 3.1 facilitate a preliminary examination of the external energy input. Equation (2) estimates the idealized mechanical heat pump, transferring heat isothermally without thermal driving forces, to recover heat at 48.0°C and produce an useful heat release at 110.0°C) with a net work input of 62.1 kW. A significant reduction of the mechanical work requirements is, however, achievable if the heat transfer processes designed to match the process streams' thermal development. Equation (3) identifies the minimized work input at 39.5 kW.

A range of system inefficiencies (quantified by thermal driving forces in heat transfer and isentropic compressor efficiencies) is examined, to address the energy requirements with a decreased margin of error. Minimized driving forces at 9.3°C and a compressor efficiency at 0.6 has a profound effect on system performance, and increases the net work inputs to 131.5 kW (with isothermal heat transfer) and 96.5 kW (with thermal glides). The mechanical driving forces more than double in both designs, but remain vastly superior to the current heat utility. Introducing an expected boiler efficiency of 0.75 (a reasonable assumption, ref. TINE's specified boiler consumption) increases the current utility to 511.7 kW, and limits the energy requirements to less than a third of current utility demands, to emphasize a heat pump's potential energy efficiency. Table 13 includes the performance of a thermally gliding heat pump, as it provides the ultimate energy efficiency.

The extensive energy gains in suggest that system inefficiencies are a prominent influence in the analyses, and that system descriptions require a significant level of detail to obtain numerical results validating (or discarding) heat pump application. Nevertheless, the preliminary performance analysis has set the bounds for heat pump performance, limited downwards at 39.5 kW, upwards at 511.7 kW (realistic boiler performance).

While researching analytical and experimental heat pump developments, as well as currently available heat pump systems, ammonia and steam-water protrude as the most applicable working fluids in mechanical compression systems operating in the desired temperature range. A set of challenges quickly follow suit, imposed by the fluids' thermodynamic properties in the investigated temperature range.

Both ammonia and water are far superior to air as heat transfer mediums, and consequently require process design to avoid pinch point problematics as well as excessive driving forces during heat transfer. It is deemed desirable to release a certain heat quantity by condensation (as is common in subcritical thermodynamic cycles), to limit the heat pumps' internal mass flow requirements.

A heat transfer pinch point arise between the gas cooler and condenser (the most strenuous point in heat release), and initiates examinations of the working fluids' thermodynamic properties at various cycle states. Equations (13) and (14) are employed to estimate respective thermal loads, once the necessary data is obtained.

A pinch point of 102.5°C/95.0°C (and thus thermal driving forces $\geq$ 7.5°C) is established for the single-stage heat pump, and requires an isentropic compression stage with peaking thermal properties in a superheated state at 65.6 bar and 158.4°C. A mechanical work load of 82.5 kW drives the process, and heat the air stream to a pinch point temperature of 97.8°. Introducing compressor inefficiencies increases sensible heat release, and the heat pump approaches and surpass the set driving forces.

The net work input increases rapidly as the compressor's performance reduces. A work load of 120.4 kW is required if it operates at 60% of isentropic capacity, and attributes a sensible heat release elevating the air stream to a pinch point temperature of 93.5°C, well outside the confines assumed for manageable heat transfer. A compressor efficiency of 0.71 yield a 7.5°C pinch point, exactly, and implies a reasonable process design as system inefficiencies are expected to rise significantly in real applications.

Operational bounds are accordingly set for single-stage mechanical compression of ammonia. External energy requirement amounts to a minimum of 82.5 kW, and peaks at 120.4 kW when the most strenuous system inefficiencies are introduced. Midrange values are presented in table 8. Table 13 summarizes the heat pump's expected energetic performance in EWOS' dehumidification process, evaluated at its extremities (with an idealized compression stage and at a 60% efficiency).

An examination of the same system with decreased thermal driving forces (ref. section 5.1) is omitted from the discussion. It fails to improve overall performance significantly, and introduces extensive uncertainties the obtained results' validity (as there is a lack of knowledge on the performance of heat transfer with air at these temperatures).

Thermal effects on the cascade cycle's high temperature side dominate heat pump design, attributed to the thermodynamic properties of steam-water working fluid application. A condenser pinch point at 110°C/102.5°C limits the hot cycle's mass flow rate, and simultaneously the sensible heat release. Assuming both sub-cycles to operate with ideal compressor performance yields a process stream peak temperature of 104.8°C, produced by a total mechanical work input of 77.3

kW and 264 kW of recovered waste heat. The system abides the pinch temperature requirements, as the water vapor mass flow rate is carefully controlled to release 322.2 kW as latent heat, the exact amount required to elevate the air stream temperature to 102.5°C. A prominent draw back of this system is the process stream's decreased outlet temperature, which requires the boiler system to supply an additional 42.5 kW to elavate the moist air to 130°C.

The influence of inefficient compression stages is readily identified in table 10. Compression at 60% of isentropic efficiency (in both subsystems) increases the net work input to 117.4 kW, and represents the largest external energy supply to the heat pump. There is, however, a benefit attached to the increased workload. As the hot sub-cycle's mass flow rate is kept constant (to release identical latent heat loads regardless of system performance), any additional work input used to compress the water vapor is consumed by a sensible heat transfer. This increases the process stream's outlet temperature to 106.9°C, and reduces the required boiler consumption by 16.7 kW. The price of decreasing compressor efficiency is accordingly paid in the cold sub-cycle only. A set of bounds are accordingly identified and presented in table 13, where the mechanical work input increases with compressor efficiencies, whereas the additional boiler efficiency steadily decreases. Table 10 provides the intermediate quantities.

Repercussions of the proposed heat pump's inability to deliver the desired air stream output temperature are diminished, by the boiler nevertheless employed to heat the air to the drier's inlet temperature. Boiler efficiencies do, however, influence overall system performance (and should be compared to the heat pump's overall efficiency). The boiler is assumed to operate with an efficiency of 0.75 in examinations of realistic operating conditions.

System Performance [kW]				
	Current	Estimated Energy Input		**Net**
Ideally	Utility	Compressor	Boiler	**Reduction**
Lorentz Heat Pump	383.8	39.5	0	**344.3**
Single-stage (NH_3)	383.8	82.5	0	**301.3**
Cascade (H_2O | NH_3)	383.8	50.2 | 25.1	42.5	**266.0**
Realistic	$\eta_B = 0.75$	$\eta_C = 0.6$	$\eta_B = 0.75$	
Lorentz Heat Pump	511.7	96.5	0	**415.2**
Single-stage (NH_3)	511.7	120.4	0	**391.3**
Cascade (NH_3 | H_2O)	511.7	75.6 | 41.8	34.4	**359.9**

Table 13: Expected external energy requirements: EWOS

The cascade cycle requires a net work input approximately equal to that of the single-stage system, and an additional external heat input, to produce the same temperature elevation in the moist air stream. This is attributed to the heat pump's increased thermal losses, arising in the intermediate heat exchanger and with shorter thermal glides through the heat exchangers. It is, however, a trade off against a decreased system pressure, which limits the operational strain applied the heat pump components (particularly the compressors) and accordingly their capital cost.

The power of thermal glides in heat transfer is aptly illustrated by the superior performance of a lorentz heat pump. A hybrid heat pump, designed with a working fluid composition to match the moist air stream's thermal development in its sorption processes (at least to a larger extent than the introduced pure fluid systems), is accordingly expected to operate with a net reduction of the energy input ranging 391.3 kW − 415.2 kW. Proposing a suitable component ratio does, however, require extensive research and development, whereas the pure fluids systems are more readily implemented.

Applying heat pumps in EWOS's dehumidification process introduces a significant net reduction of the current energy input. It is beneficial, from an energetic point of view, to implement heat pumps with significant thermal glides. Both the examined heat pumps decreased the external work input to less than 30% of the current utility, close to the performance of a heat pump designed with heat exchange to match the process stream's thermal development (operates with a net reduction of 81.1%). Two concluding remarks are drawn from this. *One*: There is strong reason to believe that the imposed inefficiencies are too optimistic, and that real application has lower energy efficiency. *Two*: The heat pumps cannot be differentiated as more or less applicable than the other, as uncertainties in the analysis are likely to exceed their difference in performance. Further examinations are accordingly required to quantify the heat pump characteristics with a greater accuracy.

5.5.2 TINE

TINE uses an ambient air stream in their whey production, required to enter the spray drier at 185°C. A heat recovery system takes advantage of the effluent air exiting the drier, and preheats the ambient air from its inlet temperature (~ 6.0°C) to 42.0°C. It recovers waste heat from the current high temperature heating utility as well, and the net reduction of heat recovery must be accounted for if the boiler is replaced by a heat pump. A purchased hot water heat load elevates the air temperature to 87°C, and is expected to balance the low temperature heat deficit arising with a heat pump application. The heat pump is designed to operate at the hot water heat utility exit, and ideally supply the 516.3 kW elevating the air stream to its drier inlet temperature. An applicable waste stream is available between 60.0°C and 35.0°C, and carries a waste heat load of 408.3 kW.

A preliminary examination of heat pump performance, based on the coefficients of performance developed in section 3.1, predicts the idealized work input of basic heat pumps. The carnot efficiency of an isothermal heat pump (ref. equation (2)) estimates a net work requirement of 161.9 kW, when useful heat, recovered at 35.0°C, is released to a heat sink holding 185.0°C. Extended thermal glides in both reservoirs merit a closer examination of other operating principles, and a lorentz cycle is expected to increase overall efficiency significantly. Equation (3) predicts an idealized net work requirement of 110.0 kW, when the heat pump is assumed to operate with thermal glides to match both thermal reservoirs' individual temperature developments. A maximized heat pump performance is attained, which includes a heat recovery of 406.3 kW, and effectively drains the available heat source.

Reduced energy recovery cannot be omitted from the analysis. A crude estimate, based on the applicability of equation (13) in heat transfer analyses, predicts the heat recovery to decrease by 116.0 kW, which must be balanced by the hot water heat utility. However, as the air stream is split between these two heat exchanger, the additional heat load does not match the decrease in heat recovery and amount to 99.0 kW. A ratio between the air stream's energy requirement and the specified hot water utility consumption (ref. appendix 2) estimates the utilities heat transfer effect to 0.8.

As the heat pump is subject to a range of system inefficiencies (quantified by thermal driving forces and a specified compressor efficiency in this study), further examinations are necessary to estimate system performance with reasonable certainty. Introducing a pinch requirement of 10% of the heat pump's internal temperature elevation (15.0°C), and a isentropic compressor efficiency of 0.6, has a profound effect on system performance. A mechanical work input of 327.5 kW drive the isothermal heat pumps, whereas thermally gliding solutions manage operation with a 238.0 kW input. Thermal inefficiencies are accordingly prominent parameters in system design and integration, mostly due to the steep temperature elevations accompanying relatively small heat inputs.

Air enters the heat pump's operational range at high temperature, which complicates use of ammonia as a working fluid in a single-stage compression heat pump. Two system configurations are accordingly examined, in an attempt to point out a favorable system design. Both are deemed applicable with a condenser temperature of 120.0°C, which introduces a pinch point between the gas cooler and the condenser with an air temperature of 11.0°C.

An assessment of a heat pump extending the air temperature to its drier inlet requirements, by carefully controlling ammonia's mass flow rate, is performed. An isentopic work load of 281.8 kW compress ammonia to a superheated temperature of 194.7°C from an evaporator temperature of 39.0°C. This abides the pinch requirements and produces a peak air temperature of 185°C. A mass flow of 1.003 kg/s is necessary to sustain the thermodynamic cycle's required energy transfer rates.

Introducing compressor inefficiencies are known to increase the sensible heat load, which merits a decreased mass flow to avoid superfluous driving forces in heat transfer. Compressors operating at 60% of the isentropic efficiency produce a maximum internal temperature of 250.9°C with a 318.5 kW input. Cooling to 120°C with a mass flow rate of 0.68 kg/s release the heat load required to elevate the air stream from 110.0°C to 185.0°C, and represents the least effective heat pump considered in the analysis. Required net work inputs at increased efficiencies are found in table 9. This renders the boiler system useless, and demand a 99 kW heat load from the medium temperature heat utility (ideally).

Strenuous effects of high temperatures at the compressor outlet merits an energetic examination of a heat pump which partially fulfill the air stream's required temperature elevation. Ammonia's mass flow is reduced to 0.236 kg/s, which matches ammonia's latent heat release (from a saturated gaseous state to a saturated liquid at 120.0°C) to the heat supply required to heat the process stream from 87.0°C to 110.0°C. This decreases the available sensible heat release considerably, as well as the energy recovery. 60.2 kW is applied in isentropic compression of a saturated gas (exiting the evaporator at 44.0°C), to yield an outlet temperature of 188.1°C. A total heat release of 207.9 kW elevates the air stream to 126.5°C. A boiler system is employed to produce the required spray drier inlet temperature, and supplies 308.4 kW to the air stream.

The available sensible heat load increases rapidly with increased compressor efficiencies. At 60% of the isentropic value, a workload of 100.3 kW is required. Additional workloads are consumed as a sensible heat release, and decrease the system's boiler requirements accordingly. The air stream only need a heat supply of 268.3 kW when the net work applied to the compressor is 100.3 kW. Intermediate workloads are readily estimated, by the performance parameters provided in table 9.

The influence increased compressor inefficiencies have on the heat recovery system cannot be neglected. As it decreases from isentropic levels to the aforementioned, "worst case" scenario at 60%, the process stream's hot water heating utility requirement is expected to increase from 39.8 kW to 47.5 kW under idealized circumstances.

Steam-water's rapidly increasing maximum system temperatures restricts condenser temperatures in the cascade cycle to 140°C, elevating the process stream to 130.0°C between the condenser and gas cooler. A latent heat release of 225.7 kW is produced by condensing water vapor from a saturated gas to a saturated liquid at 140.0°C, if the mass flow rate is controlled at 0.105 kg/s. Isentropic compression from a intermediate heat exchanger temperature of 105.0°C requires 22.6 kW, and produce a sensible heat load of 17.3 kW. This elevated the air stream to a peak temperature of 133.1°C, and leaves a high temperature utility requirement of 273.3 kW.

Ammonia supplies heat to balance the hot sub-system's energy, attained from the available heat source 41.5°C. A isentropic compressor work load of 53.7 kW is required to deliver the necessary heat load at a temperature of 109.°C. The net work input accordingly amounts to 76.3 kW.

Accounting for inefficiencies in both sub-systems increases system temperatures and work load requirements. Increased compression work is a direct loss in the cold sub-system, consumed by increased thermal driving forces at the intermediate heat exchanger. At 60% of isentropic efficiency, compression of ammonia amounts to 77.0 kW. The hot sub-system releases increased work input as a sensible heat load, which elevates the air stream to a higher temperature and decreases the high temperature utility requirement accordingly. A work input of 37.7 kW compresses water vapor to release 258.0 kW as it cools to a saturated liquid state at 140.0°C, if an efficiency of 0.6 is assumed. This reduces the remaining energy demand to 258.3 kW, which elevates the process flow temperature to 185.0°C.

Workloads at intermediate efficiencies are readily identified in table 9, and estimations of the increased hot water heating utility easily obtained. It amounts to an idealized process stream input of 46.5 kW and 49.4 kW, at the extremities of ideal compression and 60% of the isentropic efficiency, respectively.

It is evident that the extensive thermal glides decrease the heat pumps approach to idealized heat pump performance (represented by a lorentz system, with thermal glides designed to match the thermal process stream developments). Single-stage ammonia is proposed to elevate the air stream to drier inlet temperatures, which

System Performance [kW]					
Ideally	Current Utility	Estimated Energy Input: Compressor	Boiler	Utility	**Net Reduction**
Lorentz Heat Pump	516.3	110.0	0	99.0	**307.3**
Single-stage (NH_3) I	516.3	281.8	0	99.0	**135.5**
Single-stage (NH_3) II	516.3	60.2	308.4	39.8	**107.9**
Cascade (NH_3 | H_2O)	516.3	53.7 | 22.6	273.3	46.9	**119.6**
Realistic	$\eta_B = 0.75$	$\eta_C = 0.6$	$\eta_B = 0.75$	$\eta_U = 0.8$	
Lorentz Heat Pump	688.4	238.0	0	123.2	**327.2**
Single-stage (NH_3) I	688.4	318.5	0	123.2	**246.7**
Single-stage (NH_3) II	688.4	100.3	357.7	59.1	**171.3**
Cascade (NH_3 | H_2O)	688.4	77.0 | 37.7	344.3	61.5	**167.9**

Table 14: Expected external energy requirements: TINE

requires an impressive thermal glide. It is, however, achieved as a trade off against a significantly increased mass flow and a high maximum temperature. A hybrid system would again be expected to show a preferable performance with its potential for thermal glides in heat transfer. However, a set of rough conclusions regarding a hybrid heat pump's performance is readily drawn from figure 32. System pressures are expected to reach ∼100 bar, and the aqueous solution entering the absorber should be kept as lean as possible. An extensive quantity of ammonia must be absorbed, if the thermal glide is to match the process stream (and drop from 185.0°C to below 100.0°C, represented by a horizontal shift in the figure). Both process requirements severely strain component application, and extensive research is required to develop a system with these characteristics. Less intensive glides are also applicable, to obtain a preferable heat pump system compared to those presented in this thesis.

However, all evaluated heat pump solutions prove beneficial in application, and decrease the external energy input by 25%−35%. Differentiating factors are created mostly by the (already implemented) heat recovery and boiler system inefficiencies. Thus, the external energy requirements decrease rapidly as the current high temperature heating utility is replaced. It does, however, come at the cost of increased compressor capacities, temperatures and net work requirement. Determining the optimal relation between these systems is possible, but requires extended numerical work and fall outside the scope of this thesis (which aims to identify applicable heat pump solutions for waste heat recovery and overall energetic improvement). The examinations have also shown that application of these systems are differentiable, despite their inherent similarities when the evaluations focus on energetic performance. These differences appear as the heat pumps' internal thermodynamic cycles are assessed.

5.5.3 Applicable Components

The heat pump's thermodynamic cycle and arising thermodynamic properties are of significant importance to component application. The mechanical compression cycles operate with three prominent features, all with a potential to restrict component application. These three parameters are quantified in the performed thermodynamic analyses, as maximum system temperature (T_{MAX}), the maximum system pressure (P_{MAX}) and the compressor's swept volume (V_s). Tabulated values express the parameter's range as system inefficiencies increase.

Waste heat recovery is a low-pressure process, and the working fluid is subsequently pressurized to release high temperature, useful heat. Heat transfer is assumed to be an isobaric (constant pressure) process, which implies two main system pressures. Components must be designed accordingly, and present the first limiting feature of component selection.

Useful heat release requires a high working fluid temperature. Thermodynamic properties of applicable working fluids often demand an internal heat pump temperature significantly higher than the peak process stream requirements. The peak temperature is found at the compressor outlet/gas cooler inlet, where a superheated working fluid is yet to release its high grade heat. The compressor outlet temperature accordingly introduces the second restrictive feature.

The working fluid cycles the heat pump with a given mass flow rate, which has a significant effect on overall system performance. Equations (13) and (14)'s dependence on mass flow rate is an readily applicable example, which illustrate the importance of capacity control in heat pump application. Compressors are often particularly prone to this parameter, and proper design is key to applicability as well as performance.

The examinations reveal differing thermodynamic parameters despite the heat pump's inherently similar energetic performance. The basic heat pump components are introduced in section 2.5, and reveal certain characteristics to simplify this analysis.

Both heat pumps use a subcritical thermodynamic cycle in operation, and are assumed to pressurize from a saturated gaseous state. Heat recovery is managed by a single heat exchanger, used to vaporize the working fluid. Heat release is optimized by a two-stage design, where superheated gas is cooled to a saturated state and subsequently condensed to a gas-liquid mixture. As heat exchange is assumed to be isobaric processes, pressure and temperature are expected to strain gas cooler application first.

Shell-and-tube heat exchangers are typically employed in latent heat transfer. The working fluid flows tube-side, to maximize heat transfer effects by elongating the air's flow path through the exchangers. Gas coolers have a similar design, but fins and tube serpentines are generally included to increase the overall heat transfer area.

Applicable, market available components are readily identified. A maximum temperature of 287.3°C arises at the compressor outlet of the cascade system's high temperature compression stage, as it elevates TINE's air stream to 136.0°C. A peak pressure of 91.1 bar is experienced in ammonia's high temperature heat release to the same process stream. Alfa Laval produces gas coolers and condensers designed specifically for heat transfer with an air process stream.[65] Their product portfolio fits a wide range of operating conditions, and the industrial range designs handle temperatures of 400.0°C, pressures exceeding 130.0 bar and heat transfer loads down to 15.0 kW. Air-cooled condensers features solutions designed specifically to accommodate condensation of ammonia (typically considered an aggressive medium, with strong corrosive effects on several metals, including copper), with nominal capacities ranging from 20.0 kW to 1200.0 kW. Water vapor systems are not restrictive to heat exchanger design, exemplified by its common application in air heating systems at lower temperatures.

Although their original purpose is cooling the working fluid, the presented features indicate that hot air production is feasible at the examined pressures and temperatures. Further research is, however, necessary to optimize process flow heating rather than working fluid cooling.

Strenuous Thermodynamic Parameters			
EWOS	T_{MAX} [°C]	P_{MAX} [bar]	V_s [m^3/hr]
Single-stage (NH_3)	158.4-200.9	65.6	76.3-60.8
Cascade (NH_3 | H_2O)	131.0-164.0 | 175.3-233.3	49.1 | 1.4	52.6-42.6 | 781.8-682.3
TINE			
Single-stage (NH_3) I	194.7-250.9	91.1	236.1-149.1
Single-stage (NH_3) II	188.1-238.0	91.1	46.7-43.2
Cascade (NH_3 | H_2O)	170.6-217.2 | 217.4-287.3	74.3 | 3.6	47.1-37.2 | 491.9-456.8

Table 15: Relevant thermodynamic parameters.

Applied compressors produce the maximum system pressure and temperature, and their operational capacity is likely to restrict the overall heat pump operation. numerous compressor designs are applicable in industrial heat pumps. Section 2.5.2 argue positive displacement pumps as beneficial, due to favorable operational characteristics and decreased component cost at the examined net work inputs (ranging from 40.0 kW to 320.0 kW with at 60% of isentropic efficiency). Screw compressors are proposed for further examinations, due to several preferable features in application.

Current technologies restrain the outlet temperature of pressurized gas to approximately 250.0°C. The required oil-lubrication found in most systems is likely to restrict application further, as the pressurized oil-mixtures are prone to deteriorate as temperatures increase. This deterioration eliminates the lubricating effects, and increase system wear and tear as well as working fluid leakage. Screw compressors are, however, applicable without internal lubrication, if steam-water is employed as the working fluid. This increases system costs and decreases the mechanically loaded pressure differences (from 20 bar to 12 bar), but eliminates the need for oil-recovery systems and other system impositions by the lubricant.

The examinations indicate that high vapor pressures are likely to restrict compressor application in ammonia systems. A condenser pressure of 65.6 bar is expected in the single-stage heat pump used in EWOS' dehumidification process. It increases to 91.1 bar if the heat pump is utilized in TINE's hot air production. Pressure levels are expected to remain unaffected by the compressor's efficiency, but the swept volumes are not. The reduced mass flow rate (due to an increased sensible heat load) decreases the compressor capacity requirements accordingly. Excessive temperatures at the compressor outlet, ranging from 158.4°C to 250.9°C, are also a restrictive factor, where high compressor efficiencies are favorable to limit the temperature elevation.

The cascade system reclaims the increased thermal inefficiencies (identified in the preceding sections) with a decreased maximum system pressure. It peaks in the cold sub-system, where ammonia condenses at a significantly reduced temperature compared to the single-stage system, at 49.1 bar and 74.3 bar when implemented in EWOS' and TINE's production. Water vapor pressures are significantly lowered, and easily accommodated by most compressor systems. However, capacity requirements skyrocket compared to compression of ammonia with similar work inputs, estimated at approximately ten times that of an ammonia system in the same operating range. Compressor outlet temperatures are a restrictive factor of the cascade system as well, and ranging from 175.3°C to 287.3°C in the examinations.

GEA Grasso manufactures screw compressors well suited for industrial application, and have four series to meet the demands of various operational ranges.[2] The AC series are specifically designed for applications with the natural refrigerants CO_2 and NH_3, and is currently used in hot water production at an industrial scale. The compressor is operable with a system pressure ranging up to 130.0 bar, and a capacity between 60.0 m^3/hr and 155.0 m^3/hr. It is accordingly capable of producing the required maximum system pressures (although the number of compression stages requires a closer investigation of both process requirements and specific compressor's capabilities, ref. the aforementioned mechanically loaded pressure differences). The estimated swept volumes fall just outside its operational range, but not enough to discard system application (the estimations bear a significant uncertainty, due to the crude estimations of component efficiencies).

Their MC series is designed with swept volumes ranging from 471.0 m^3/hr to 860.0 m^3/hr, to readily accommodate the pressurization of water vapor. It is, alongside the two remaining series, designed to work with a design pressure of 28.0 bar. All the compressors are also available in high-pressure models handling outlet pressures up to 52.0 bar (certain models can be upgraded to a maximum design pressure of 63.0 bar).

Market available components are, accordingly, applicable with the identified working fluids as well as their imposed maximum system pressures and system capacities. Maximum system temperatures are, however, more cumbersome to overcome with current compressor designs (and the applied oil-lubricants). And as the temperatures range in an area close to and above the maximum temperature of screw compressor systems ($\sim$250.0°C), extensive examinations are necessary to deem compressors applicable. Designing to optimize compressor performance will, however, reduce the system's maximum internal temperature, which decreases the overall compressor strain and increase applicability of the available heat pump components.

A commonly applied solution to diminish the thermal restrictions of compressor application is interstage cooling of the working fluid. This implies that the working fluid is compressed in several stages. As pressurized gas exit a compression stage, it undergoes an isobaric cooling process to approach a saturated gas state. Further pressurization accordingly yields a lower outlet temperature than a single compression stage would have. The benefits of such processes should be carefully considered, and further examined in application of high temperature, high-pressure heat pumps.

Application of hybrid heat pumps also face the restraint of high compressor outlet temperatures. This is aptly illustrated by figure 32, where the beneficial features first and foremost are presented as a pressure reduction. An increased understanding of the complex sorption processes may, however, allow heat pump designs which achieve the similar performance at a lower maximum system temperature. Matching the heat pump's useful heat release to the process stream's thermal development facilitates this. The poor match between the heat pump and air stream temperature provide a considerable contribution to the compressors outlet temperature (the examined systems have a thermal driving force generally exceeding 50.0°C as air exits the gas cooler at its peak temperature).

Solution pumps and expansion valves are not expected to be cumbersome in application. Valves are simple mechanical designs, and readily upgraded to withstand the imposed pressure differences. Solution pumps are common industrial components, and expected to handle the operational pressure and temperature ranges with significantly less strain than the compressors applied alongside them. However, the limited knowledge of the liquid pressurization stage complicates evaluation of the thermodynamic requirements and performance of applicable components. Industrial manufacturers of these systems are readily identified, and include Alfa Laval [65] (produces both valves and pumps) and Grundfos Industry [31] (which mainly produces solution pumps).

5.6 Further Work

Application of industrial heat pumps for high temperature heat recovery is a highly complex task, which requires extensive analytical work, experimental work and market research to be completed successfully. This thesis aims at identifying the most prominent restrictions, and appoints available heat pumps which challenge these restrictions. Accordingly, the further work springing from this thesis takes two forms:

- A reexamination of the proposed scenarios and heat pump technologies, with increased analytical accuracy, is necessary to obtain performance estimations with a real applicability.
- An examination of different industrial scenarios is necessary, in order to determine whether the approach used to identify heat pump technologies in this thesis, may be applied under other circumstances.

An important part of this thesis is the technical review, where the industrial scenarios, available heat pump technologies, promising working fluids and market available components are identified. The performance review is used to establish the characteristic features of different solutions, and provides a crude estimate of various performance parameters and the overall energy efficiency. However, it fails to quantify the performance of more complex heat pump technologies (ref. hybrid heat pumps, absorption heat pumps, where sorption processes are key to understanding and estimating performance). Extensive numerical work is necessary to obtain a manageable accuracy in the examinations. Thus, development of such a tool should be pursued, as the hybrid heat pump is assumed to hold a vast potential in high temperature industrial applications. (This is supported by the ongoing studies at a Danish research institute, in collaboration with industrial participants, expected to finish in February, 2015.)

Overall system efficiency is largely determined by the applied compression stages and the effectiveness of heat transfer. Both must therefore be examined further, analytically and experimentally. Process integration should also be considered more carefully. Heat pump integration is not optimized in the performed analysis, despite the estimated advantages of such integration. The required optimization should be performed at the next step of the analysis. Repercussions on other process streams should be evaluated, to maximize the benefits of heat pump application.

A series of effects has been omitted from the analyses (dry/wet compression, leakage effects, frictional effects, etc.). Oil-lubricants should be thoroughly investigated, as should oil-free operation, and a choice between such lubricants should be made to complement the evaluated system pressures and temperature ranges. This analysis assumes a single compression stage to be feasible, and further analysis should accordingly examine the applicability of multistage compression systems (ref. the compound systems introduced in section 2.3).

Alternatives to the proposed GEA Grasso compressors should be identified and evaluated for application. Specialized designs are highly desirable (ref. GEA Grasso's AC series developed specifically for use with CO_2 and NH_3). Rotrex is a manufacturer with a component range used for steam-water compression [29], and deserves a closer look.

Heat transfer depends on heat exchanger design, the selected working fluid(s) and the process stream medium. Shell-and-tube heat exchangers are typically employed, but other designs are available and should be investigated. Plate-and-fin exchangers are among the most prominent alternatives, and Alfa Laval's plate-and-shell design[65] should also be examined more closely.

The working fluids heat transfer effects must be assessed more closely, as the formation of liquid during heat exchange complicates heat transfer predictions. Pressure losses through the heat exchangers have repercussions for the overall system design, which has to be accounted for when detailed system descriptions are given. Application of fluid mixtures should also be evaluated, as is necessary to add hybrid heat pumps to the performance analysis.

Research of effects on heat transfer to air is scarce, and both experimental and numerical examinations are necessary to determine a best-practice solution. This is particularly important, as the evaluated drier processes commonly employ air as their service medium. A thorough understanding of the heat transfer processes must therefore be attained, if heat pumps are to provide a lasting solution to increase these processes.

Thermally driven heat transformers may require minor process modifications in the industrial production where it is introduced, in order to successfully integrate as a heat recovery system. A techno-economical analysis is accordingly necessary to validate/discard these systems in further analyses.

It would exceed the scope of this thesis to endeavor to provide answers to the different questions outlined above.

6 Conclusion

This thesis has shown that high temperature application of industrial heat pumps depends on numerous factors. Industrial heat pumps effectively recover excess process energy, and an abundance of technologies are readily applied at low temperatures. Available heat at higher temperatures is, however, more challenging to recover effectively, due to the limited applicability of different heat pumps, or their components. Thus, a study of industrial heat pumps which may handle these operational restrictions must be carefully conducted, and is preferably performed in several steps with increasing accuracy.

Contact with industry is key to combine actual performance evaluations with relevant theoretical aspects. Two industrial scenarios were chosen. Both consider drier processes, notorious for high temperature requirements, release of waste heat and widespread industrial application. Distinct process characteristics justify and necessitate performance of two case studies, which reflects the various challenges of high temperature heat recovery.

A simplified calculation tool is developed to examine the energy efficiency of various heat pump technologies, and illustrate the restraining factors of current heat pump development. The calculation tool is two-fold. A preliminary analysis nominates select heat pump designs for further analysis. The evaluations are based on the generic working principle's applicability within the identified process streams, and provides a crude estimate of their potential energetic benefit compared to the heating utility currently applied by the respective manufacturers. The following analysis, which forms the main part of this thesis, examines performance when market available systems are introduced, and includes an identification of strenuous thermodynamic parameters.

The technical review contains three sections necessary to evaluate high temperature heat pump application. It is initiated by an identification of industrial heat pump designs. Two main solutions are available, mechanical heat pumps and thermally driven heat pumps, both with sub-systems with various designs and operational characteristics. Applicable working fluids are briefly introduced, as knowledge of their individual thermodynamic characteristics is key to successful heat pump operation. A review of the most prominent heat pump components finalizes the section, and identifies manufacturers of components with feasible designs for high temperature application. Mechanical compression of a pure working fluid is commonly applied in industrial heat recovery, as it is extensively researched and readily implemented with the available thermal reservoirs. However, as the understanding of and experience with fluid mixtures is increasing, use of more complex heat pump designs are prone to spur high temperature application in the years to come.

The preliminary selection analysis nominates closed-cycle mechanical heat pumps as most-likely-to-succeed within the identified scenarios, due to two beneficial features: They have a significantly higher energetic performance than the current heating utilities and they are readily applied between the available thermal reservoirs. Thermally driven heat transformers are expected to increase the external energy requirements significantly, and are therefore omitted from further examinations within the scope of this thesis. Other heat pumps are discarded from the following studies, due to a prominent mismatch between their operating principles and the available thermal reservoirs.

Suitable working fluids must be appointed to pursue further examination. Single-stage mechanical compression of ammonia, and a cascade heat pump with steam-water/ammonia as a binary fluid are deemed most applicable. Prominent research and current industrial applications support the selections. Analytical studies of the systems' thermodynamic cycles predict the net reduction of external energy requirements, and reveal a significant benefit of heat pump application (the net reduction exceeds 70% in EWOS' production, and 25% in TINE's). The benefits of a hybrid heat pump could not be determined accurately by the calculation tool (due to the complexities of a two fluid component's sorption processes and its influence on heat transfer and pressurization).

The compressor restricts high temperature application, and internal maximum pressure/temperature/capacity are the strenuous parameters. However, as GEA Grasso produces compressors which cope with pressures exceeding those experienced in the identified processes, it should be possible to adapt operation to the utilized working fluids. Maximum temperatures cannot exceed 250°C, but interstage cooling can solve the problem.

The results cannot be used to predict the applicability of heat pumps in high temperature, industrial processes. The undeniable amount of uncertainty related to these analyses, are attributed to two main factors. One factor is related to the validity of the calculation tool, and its significant margins of error. Simulations should be performed to support the decision-making. The tool's structural integrity is, however, satisfactory for application. It generates results to indicate important characteristics of heat pump design, necessary for further examinations.

The other factor is related to a limited knowledge of the identified heat pump components. Although restrictive operational parameters are known and applicable, a sound coupling of the vaious solutions has not been determined. Extended evaluations and market research is therefore required, to establish a real heat pump system based on the conducted examinations.

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A Appendix 1

Processes at Tine Meieriet Verdal

The tables include input and output temperatures of all processes at Tine Meierier Verdal, as well as the energy requirements of each process. Working fluids used for heating and cooling purposes, and their respective mass flows, are also disclosed. The most relevant processes for investigation in the thesis, mainly due to their temperature levels, are highlighted in purple.

Heating Processes:

Avdeling	Prosess	Servicemedie	Energibruk		Prod. Temp. inn	Prod. Temp. ut	m	m	ΔH	Start	Slutt	Drifttid	
			kWh/dag	MWh/år	°C	°C	kg/d	kg/h	kW	kl	kl	h/d	d/år
Ysteri	Ystemelkpasteur - råmelk	sparevann	8 364	2 389	4	27	327 273	23 477	600	1.5	17.9	13.9	286
Ysteri	Temp.justering sparevann	sparevann	1 545	441	30	35	264 860	19 000	111	1.5	17.9	13.9	286
Kjerneri	Prosesstank 1B	sparevann	341	104	5	20	24 800	8 267	114	5	8	3.0	306
Kjerneri	Prosesstank 2B	sparevann	341	104	5	20	24 800	8 267	114	7	10	3.0	306
Kjerneri	Prosesstank 3B	sparevann	341	104	5	20	24 800	8 267	114	9	12	3.0	306
Kjerneri	Forvarmer	sparevann	637	195	4	14	69 440	4 960	45	7	21	14.0	306
Ysteri	Mysefløte tank + pasteur P5	sparevann/hetvann	292	83	30	95	4 900	12 000	715	0	24	0.4	286
Mysefabrikk	Permeattørke-fv	sparevann/hetvann	24 631	7 537	30	92	1 416 000	59 000	1 026	6	6	24.0	306
Mysefabrikk	WPC-tørke fv	sparevann/hetvann	6 666	2 040	42	87	528 000	22 000	278	6	6	24.0	306
Ysteri	Ny grensemelkløsning varme	sparevann/hetvann	495	151	4	85	5 500		720	0	2	2.0	306
Mysefabrikk	CIP mysepasteurer	sparevann/hetvann	120	37	6	52		559	30	19	23	4.0	306
Mysefabrikk	UF CIP	sparevann/hetvann	800	245	6	50		3 896	200	21	1	4.0	306
Mysefabrikk	RO CIP	sparevann/hetvann	210	64	6	50		1 023	53	21	1	4.0	306
Mysefabrikk	WPC80 pre-treatment	sparevann/hetvann	2 078	636	4	66	34 272	1 428	87	6	6	24.0	306
Mysefabrikk	MVR hetvann	sparevann/hetvann	10 556	3 052	4	72	288 997	16 609	1 161			17.4	289
Generelt	Varmt tappevann	sparevann/hetvann	8 180	2 503	6	65	145 950	6 081	419	0	24	24.0	306
Generelt	CIP kjerneri	sparevann/hetvann	1 960	600	6	92	19 535	814	82	0	24	24.0	306
Generelt	CIP fløtemottak	sparevann/hetvann	1 593	487	6	92	15 872	661	66	0	24	24.0	306
Generelt	Romoppvarming	sparevann/hetvann	7 000	2 100	6	70	93 750	3 906	292	0	24	24.0	306
Ysteri	Brukssyrepasteur	sparevann/hetvann	149	43	30	62.5	5 000	5 000	149	4	5	1.0	286
Ysteri	CIP Ysteri, past. side	sparevann/hetvann	3 749	1 170	30	92	65 968	2 749	156	0	24	24.0	286
Ysteri	CIP Ysteri, upast. side	sparevann/hetvann	2 076	648	30	92	36 521	1 522	86	0	24	24.0	286
Ysteri	Ystemelkpasteur - ystemelk	hetvann	1 781	509	68	73	320 650	23 002	128	1.5	17.9	13.9	286
Ysteri	Fløtepasteur	hetvann	46	13	70.1	75	10 240	1 200	5	7	19	8.5	286
Ysteri	Mysepasteur 1. avtapp varme	hetvann	644	184	67.9	72	171 300	11 233	42	4	21	15.3	286
Ysteri	Mysepasteur 2. avtapp varme	hetvann	1 100	314	66	72	200 000	12 579	69	5	23	15.9	286
Ysteri	Ystevannspasteur	hetvann	1 874	535	56	85	70 000	7 500	199	2	16	9.4	286
Kjerneri	Fløtepasteur varming	Hetvann	449	137	68	75	69 960	12 000	77	1	6.83	5.8	306
Mysefabrikk	Mysepasteur varming	Hetvann	1 455	445	68	72	352 000	22 000	91	1	19	16.0	306
Mysefabrikk	TVR hetvann	Hetvann	959	277	62	70	138 269	7 474	52			18.5	289
Mysefabrikk	MVR CIP	Hetvann	4 083	1 181	60	100	12 500		583			7.0	289
Mysefabrikk	TVR CIP hetvann	Hetvann	4 083	1 181	60	100	12 500		583			7.0	289
Ysteri	HTT varming	damp/hetvann	2 501	714	55	122	39 527	2 836	179	1.5	17.9	13.9	286
Mysefabrikk	MVR oppstart damp	Damp	414	120				600	333			1.2	289
Mysefabrikk	TVR damp	Damp	5 910	1 709				575	319			18.5	289
Mysefabrikk	TVR CIP damp	damp	2 181	631					312			7.0	289
Mysefabrikk	MVR oppstart hetvann/damp	Damp	870	252				15 000	700			1.2	289
Mysefabrikk	Permeattørke-gass	Gass	30 017	9 185	92	160	1 416 000	59 000	1 126	6	6	24.0	306
Mysefabrikk	WPC-tørke gass	Gass	18 168	5 559	87	185	528 000	22 000	605	6	6	24.0	306
	Totalt		158 655	47 679									

Cooling Processes:

Avdeling	Prosess	Servicemedie	Energibruk		Prod. Temp. inn	Prod. Temp. ut	m	m	ΔH	Start	Slutt	Drifttid	
			kWh/dag	MWh/år	°C	°C	kg/d	kg/h	kW	kl	kl	h/d	d/år
Ysteri	HTT kjøling	kjølevann	1 946	556	56	50	291 900	21 000	140	1	17	13.9	286
Ysteri	Fløtepasteur II	kjølevann	479	137	63	15.5	11 000	1 200	52	7	19	9.2	286
Ysteri	Brukssyrepasteur P8A	kjølevann	414	118	92	21	5 000	2 500	207	5	7	2.0	286
Ysteri	Mysepasteur 1. avtapp II	kjølevann	3 712	1 060	34	14.5	171 300	11 233	243	4	21	15.3	286
Mysefabrikk	TVR flash cooler	kjølevann	9 800	2 834	60	35		7 030	98			24.0	289
Mysefabrikk	Krystalliseringstanker	kjølevann	1 297	375	40	18	83 916	4 536	70			18.5	289
Generelt	Kuldeanlegg - oljekjøler	kjølevann	7 200	2 203	42	35			300			24.0	306
Generelt	Overhetingsvarme fra kuldeanlegg	kjølevann	7 800	2 387	30	46	432 000	18 000	325			24.0	306
Ysteri	Ystevann	kjølevann	40	12	65	50	2 310	11 000	193	7.00	7.21	0.2	286
Ysteri	Ystevann	kjølevann	74	21	65	50	4 200	20 000	350	2.00	2.21	0.2	286
Ysteri	Mysepasteur 2. avtapp II	kjølevann/isvann	6 000	1 714	41	14	200 000	12 579	377	5	23	15.9	286
Mysefabrikk	MVR cooling water cleaning	kjølevann/isvann	2 115	612	30	10		12 950	302			7.0	289
Ysteri	Grensemelkkjøler	kjølevann/isvann	187	53	30	4	6 480	270	8	0	24	24.0	286
Ysteri	Fløtepasteur I	kjølevann/isvann	116	33	15.5	4	11 000	1 200	13	7	19	9.2	286
Ysteri	Ny grensemelkløsning kjøling	kjølevann/isvann	495	151	85	4	5 500		720	2	3	1.0	306
Ysteri	Brukssyrepasteur P8B	kjølevann/isvann	99	28	21	4	5 000	5 000	99	1	2	1.0	286
Ysteri	Lakekjøler	kjølevann/isvann	495	141	38	12.5	33 420	1 393	21	0	24	24.0	286
Kjerneri	Prosesstank 1C	kjølevann/Isvann	136	42	20	14	24 800	12 400	68	10	12	2.0	306
Kjerneri	Prosesstank 2C	kjølevann/Isvann	136	42	20	14	24 800	12 400	68	12	14	2.0	306
Kjerneri	Prosesstank 3C	kjølevann/Isvann	136	42	20	14	24 800	12 400	68	14	16	2.0	306
Kjerneri	Oljekjøler	kjølevann/Isvann	211	65	30	2	13 580	970	15	7	21	14.0	306
Ysteri	Råmelkkjøler	isvann	1 078	308	7	3	250 000	19 231	83	7	20	13.0	286
Ysteri	Fløtetank T21/T22	isvann	35	10	8	4	9 600	400	1	0	24	24.0	286
Ysteri	Mysepasteur 1. avtapp I	isvann	1 237	353	14.5	8	171 300	11 233	81	4	21	15.3	286
Ysteri	Mysepasteur 2. avtapp I	isvann	1 333	381	14	8	200 000	12 579	84	5	23	15.9	286
Ysteri	Myse kjøling før RO	isvann	956	273	8	6	430 298	30 464	68			14.1	286
Ysteri	Myse kjøling etter RO	isvann	879	251	10	3	113 000	8 000	62			14.1	286
Kjerneri	Fløtebiler	Isvann	327	100	7	4	119 040	39 680	109	6.25	16	3.0	306
Kjerneri	Fløtepasteur kjøling	Isvann	455	139	12.1	5	69 960	12 000	78	1	6.83	5.8	306
Kjerneri	Prosesstank 1A	Isvann	45	14	7	5	24 800	12 400	23	3	5	2.0	306
Kjerneri	Prosesstank 2A	Isvann	45	14	7	5	24 800	12 400	23	5	7	2.0	306
Kjerneri	Prosesstank 3A	Isvann	45	14	7	5	24 800	12 400	23	7	9	2.0	306
Kjerneri	Kjøler fløte	Isvann	665	203	14	4	72 492	11 923	109	14	20.08	6.1	306
Kjerneri	Modningstank	Isvann	64	19	5	4	69 440	5 748	5	14	2.08	12.1	306
Kjerneri	Kjernemelkkjøler	Isvann	646	198	15	3	48 454	3 461	46	7	21	14.0	306
Kjerneri	Elting	Isvann	114	35	15	10	34 090	2 435	8	7	21	14.0	306
Kjerneri	Kjøler vann BDS	Isvann	4	1	8	2	500	36	0	7	21	14.0	306
Kjerneri	Kjøler add on system	Isvann	65	20	12	10	48 692	3 478	5	7	21	14.0	306
Mysefabrikk	Biler med mysekonsentrat	Isvann	530	162	6	4	255 600	31 950	66	8	19	8.0	306
Mysefabrikk	Mysepasteur kjøling	Isvann	1 803	552	11	6	352 000	22 000	113	1	19	16.0	306
Mysefabrikk	UF kjøling	Isvann	2 800	857	10	4		25 120	147	1.5	20.5	19.0	306
Mysefabrikk	RO loop kjøling	Isvann	3 200	979	10	6		40 356	168	1.5	20.5	19.0	306
Mysefabrikk	WPC tanker, WPC80	Isvann	24	7	4	3	24 130	1 005	1	0	24	24.0	306
Mysefabrikk	MVR kjøling av vakuumpumper	Isvann	203	59	1	6		2 000	12			17.4	289
Mysefabrikk	TVR kjøling av vakuumpumpe	Isvann	162	47	1	6		1 500	9			18.5	289
Mysefabrikk	Permeattørke kjøling	isvann	308	94	1	6		22 000	128	6	6	2.4	306
Mysefabrikk	WPC-tørke kjøling	isvann	31	9	1	6		2 200	13	6	6	2.4	306
	Totalt		59 943	17 725									

Both tables are retrived from IFE's Case Study of Tine Meieriet Verdal [39].

B Appendix 2

The calculations have been performed using equation (13), and the thermodynamic tables for water and air available in the appendices of *Fundamentals of Engineering Thermodynamics*. All heat transfer effects are idealized, and the ideal effect represent the heat quantities required to cool or heat the moist air. The indicated real effects where provided by external sources, and are incorrect.

The heat exchangers, numbered 1 through 6, refer to the heat exchangers in figure 4. Abreviations *HR*, *HP* and *B* represent the *H*eat *R*ecovery system, the *H*eat *P*ump and the *B*oiler, respectively.

	EWOS Base Case Scenario													
		HX1 (HR)		HX2 (HP1)		HX3		HX4 (HR)		HX5 (HP1)		HX6 (B)		
Given Effect (kW)		-195		-221		-738		195		244				
Air Mass Flow Rate (kg/s)	8,33		8,30		8,25		8,07		8,07		8,07		8,07
	in		out|in		out|in		out|in		out|in		out|in		out
Base Case Scenario														
Air Flow Temperature (°C)	52,0		40,2		35,5		20,0		42,4		63,3		110,0
Air Flow Humidity (%)	50,0		88,3		100,0		100,0		27,7		10,0		1,6
Air Flow Enthalpy (kJ/kg)														
Ideal Effect (kW)		-184,0		-148,3		-580,8		184,0		171,3		383,8	
Real Effect (kW)		-195		-221		-738							
Energy Consumption (kW)	**407**													

C Appendix 3

The calculations have been performed using equation (13), and the thermodynamic tables for air available in the appendices of *Fundamentals of Engineering Thermodynamics*. All heat transfer effects are idealized, and the ideal effect represent the heat quantities required to heat the dry air. The indicated real effects where provided by external sources, and have been used to obtain a value for the boiler efficiency (η_B) used in the developed calculation tool.

The heat exchangers, numbered 1 through 6, refer to the heat exchangers in figure 5. Abreviations *HR*, *HW*, *HP* and *B* represent the *H*eat *R*ecovery system, the district *H*ot *W*ater utility, the *H*eat *P*ump and the *B*oiler, respectively. HX3 is the imagined heat pump implemented after the hot water utility, to reduce the boiler's energy consumption.

Tine Base Case Scenario

		HX1 (HR)		HX2 (HW)		HX3 (HP)		HX4 (B)		SPRAY	DRIER		HX5 (HR)		HEAT	RECOVERY	HX6 (HR)
Effect (kW)				293				700									
Air Mass Flow Rate (kg/hr)		22 000		18 700		18 700		18 700					18 700				
	in		out|in		out|in		out|in		out			in		out			
Base Case Scenario																	
Air Flow Temperature (°C)	6		42		87		87		185			70		50			
Air Flow Enthalpy (kJ/kg)	279		315		361		361		460			343		323			
Ideal Effect (kW)		221		235		0		516					-105				-116
Real Effect (kW)		221		293		0		700					-105				-116
Energy Consumption	**993**																

D Appendix 4

COPs are calculated using equation (2), (3) and (9), found in table 3. Temperature glides in the heat transformers were estimated with the mathematical relation for average temperatures ($\overline{T}$) from section 3.1.2.

ΔT_{drive} is the difference between the varmest heat sink temperature and the coldest heat source temperature. It was only developed for the original heat source temperature, as the alternative heat sources are considered only briefly.

Heat extraction has been increased for thermal heat production in the scenario based on EWOS' production process, to balance the increased heat requirements accompanying these operating principles. The temperature glide was extended merely 3°C, from 48°C 45°C, due to the stream's entrained energy. TINE's waste stream cannot be extended to extract more energy in real applications, as its energetic potential is fully utilized at ∼408.3 kW.

Idealized COPs and significant operational condtions are emphasized by a bold typeset, to highlight the values discussed in section 4.2.2.

	Heat Exchanger Temperature Driving Forces		
	1 (EWOS)	2 (Tine Meieriet)	
ΔTdrive (°C)	62,00	150,00	
5% of ΔTlift (°C)	3,1	7,5	
10% of ΔTlift (°C)	6,2	15	
15% of ΔTlift (°C)	9,3	22,5	
	Alternative Heat Sources		
	1 (EWOS)	2 (Tine Meieriet)	
20% decrease (°C)	-9,6	-7	
10% decrease (°C)	-4,8	-3,5	
10% increase (°C)	4,8	3,5	
20% increase (°C)	9,6	7	
	High Pressure Hot Water		Enthalpy (kJ/kg)
Mass Flow - Isothermal (kg/s)	12,03	N/A	
Mass Flow - Glide (kg/s)	9,23	N/A	
Thigh (K)	363,00	N/A	376,92
Tavg (K)	357,98	N/A	355,90
Tlow (K)	353,00	N/A	334,91
	District Heat Source		Enthalpy (kJ/kg)
Mass Flow - Isothermal (kg/s)	N/A	10,75	
Mass Flow - Glide (kg/s)	N/A	7,65	
Thigh (K)	N/A	375,00	427,49
Tavg (K)	N/A	363,89	402,05
Tlow (K)	N/A	353,00	334,91
	Current Heat Utilities		
	1 (EWOS)	2 (Tine Meieriet)	
	kW	**kW**	
Boiler Efficiency Ideal	383,8	516,3	
Boiler Efficiency 0.75	511,7	688,4	

	Scenario					
	1 (EWOS)			2 (Tine Meieriet)		
Mechanical Compression	Source	Sink	Extra	Source	Sink	Extra
Thigh (°C)	50,00	110,00	11,00	60,00	185,00	11,00
Tavg (°C)	48,99	84,33	10,49	46,38	129,90	10,49
Tlow (°C)	48,00	63,00	10,00	35,00	87,00	10,00
Thermal Heat Production						
Thigh (°C)	50,00	110,00	11,00	60,00	185,00	11,00
Tavg (°C)	47,46	84,33	10,49	46,38	129,90	10,49
Tlow (°C)	45,00	63,00	10,00	35,00	87,00	10,00
Mechanical Compression	Source	Sink	Extra	Source	Sink	Extra
Thigh (K)	323,00	383,00	284,00	333,00	458,00	284,00
Tavg (K)	322,00	358,99	283,50	320,34	407,04	283,50
Tlow (K)	321,00	336,00	283,00	308,00	360,00	283,00
Thermal Heat Production	Source	Sink	Extra	Source	Sink	Extra
Thigh (K)	323,00	383,00	284,00	333,00	458,00	284,00
Tavg (K)	320,49	358,99	283,50	320,34	407,04	283,50
Tlow (K)	318,00	336,00	283,00	308,00	360,00	283,00
NB! For Graphical Representation						
Qavailable - Mechanical (kW)	616,37	232,57		108,00	0,00	
Qdelivered - Mechanical (kW)	0,00	616,37		516,33	516,33	
Qavailable - Thermal (kW)	1 619,69	1 235,89				
Qdelivered - Thermal (kW)	0,00	1 619,69				

	Coefficient of Performance (COP)					
	COP	Wnet (η=1)	Wnet (η=.6)	COP	Wnet (η=1)	Wnet (η=.6)
Carnot		**kW**	**kW**		**kW**	**kW**
Ideal	**6,18**	**62,1**	103,5	**3,05**	**169,1**	281,8
Tdrive = 5%	5,66	67,8	113,0	2,82	183,0	305,0
Tdrive = 10%	5,23	73,4	122,3	2,63	196,5	327,5
Tdrive = 15%	**4,87**	78,9	**131,4**	**2,46**	209,5	**349,2**
20% decrease	6,02	63,7	106,2	3,01	171,7	286,2
10% decrease	6,10	62,9	104,9	3,03	170,4	284,0
10% increase	6,25	61,4	102,3	3,08	167,8	279,7
20% increase	6,33	60,6	101,0	3,10	166,6	277,6
Lorentz		**kW**	**kW**		**kW**	**kW**
Ideal	**9,71**	**39,5**	65,9	**4,69**	**110,0**	183,3
Tdrive = 5%	8,38	45,8	76,3	4,08	126,7	211,1
Tdrive = 10%	7,39	51,9	86,5	3,62	142,8	238,0
Tdrive = 15%	**6,63**	57,9	**96,5**	**3,26**	158,3	**263,8**
20% decrease	7,71	49,8	83,0	4,34	118,9	198,1
10% decrease	8,59	44,7	74,5	4,51	114,4	190,7
10% increase	11,15	34,4	57,4	4,89	105,5	175,9
20% increase	13,11	29,3	48,8	5,11	101,1	168,5
Heat Amplifier		**kW**	**kW**		**kW**	**kW**
Isothermal	N/A	N/A	N/A	N/A	N/A	N/A
Glide	N/A	N/A	N/A	N/A	N/A	N/A
		Qnet			Qnet	
Heat Transformer (Isothermal)		**kW**			**kW**	
Ideal	**0,42**	**910,5**		0,21	2 430,6	
Tdrive = 5%	0,40	949,8		0,20	2 533,2	
Tdrive = 10%	0,39	987,4		0,20	2 622,7	
Tdrive = 15%	0,38	1 023,4		0,19	2 699,7	
20% decrease	**0,32**	**1 216,7**		0,16	3 299,1	
20% increase	0,52	736,1		0,27	1 942,0	
High Pressure Hot Water	0,76	505,3		N/A	N/A	
District Heat	N/A	N/A		0,52	994,9	
Tdrive = 5% (Alt. Heat Source)	0,73	527,7		0,50	1 040,2	
Tdrive = 10% (Alt. Heat Source)	0,70	549,1		**0,48**	**1 080,5**	
Heat Transformer (Glide)		**kW**			**kW**	
Ideal	**0,55**	**694,4**		0,38	1 349,9	
Tdrive = 5%	0,52	737,5		0,36	1 451,0	
Tdrive = 10%	0,49	778,7		0,34	1 541,1	
Tdrive = 15%	0,47	818,1		0,32	1 620,8	
20% decrease (heat source)	**0,42**	**905,5**		0,32	1 625,1	
20% increase (heat source)	0,67	569,4		0,44	1 161,6	
20% decrease (cold heat sink)	**0,62**	**622,7**		0,43	1 200,9	
20% increase (cold heat sink)	0,47	815,5		0,33	1 567,6	
High Pressure Hot Water	**0,99**	**387,9**		N/A	N/A	
District Heat	N/A	N/A		**0,73**	**707,8**	
Tdrive = 5% (Alt. Heat Source)	0,93	412,3		0,68	763,0	
Tdrive = 10% (Alt. Heat Source)	**0,88**	**435,8**		**0,64**	**812,8**	

High Temperature Heat Pumps for Industrial Applications: A Review

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Abstract

This review aims at identifying technological advances which support the development of high temperature, industrial scale heat pumps. An assessment of their applicability within the confines of two chosen industrial scenarios is performed. Industry contact is key to combine the heat pump examinations with relevant theoretical aspects, and perform a meaningful performance analysis. Drier processes are considered, and reveal a peak temperature requirement ranging from 110°C to 185°C. A calculation tool examining the energy efficiency of various heat pumps is developed, to nominate the most suitable alternatives among the identified technologies. A preliminary selection considers each heat pump's match to available thermal reservoirs, and estimates their expected energy efficiency. The most prominent solutions are further investigated, after developing system descriptions to mimic real life applicability. A reexamination of their energetic performance (with increased accuracy in the calculation tool), emphasizes the limitations inflicted by select working fluids. Strenuous operational parameters are identified at various thermodynamic stages in the heat pump cycle, to examine the applicability of market available components.

I. Introduction

Long term projections estimate the energy consumption in the industrial sector, a sector already consuming about half the world's delivered energy, to increase by an average of 1.5% per year until 2035.[1] Increasing costs of feedstock energy, and tightening regulations on process emissions, have led the industrial sector to focus on reducing its process energy consumption for years. Numerous systems have been employed to increase energy efficiency, among them heat pump systems recovering useful heat from excess energy available in process waste streams. Primary energy sources used in the industrial processes can be fully or partially replaced by this produced heat, to increase the system's energy efficiency and decrease it's emissions. However, current publications estimates that manufacturing industries based in OECD countries can improve their energy efficiency by 18-26% compared to 2004 levels, and that CO_2-emissions can be reduced by 19-32%, in terms of primary energy sources alone.[2]

Successful early edition heat pump applications were typically limited to an upper operating temperature of 75-80°C, and they are rarely employed in processes requiring heat above ∼100°C today. As illustrated in figure 1, significant amounts of waste heat are available around and above these temperature levels. Over the years, technological advances have found their way to market, which allow heat pumps to operate at these temperatures. Introducing these heat pumps to the industrial sector can have a significant effect on the energetic performance of their processes. However, the arising of new heat pump solutions make selection of a suitable heat pump to a specific industrial process a complex task.

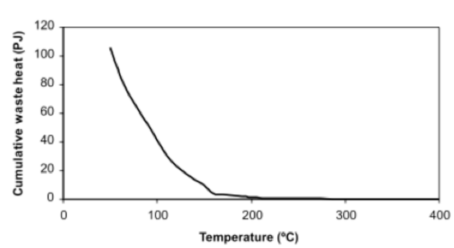


Figure 1: Cumulative waste heat in the Dutch refining and chemical industry, 1999.

II. Focus

The study aims to identify industrial heat pumps challenging these operational restrictions, and both theoretical and experimental heat pump developments are examined. The initial research focused at process heat sink temperatures ranging from 180-250°C, and an evaluation of heat pump performance against the current process heat utilities. A generic calculation tool is developed to support the performance analyses.

A set of real industrial scenarios is proposed, with explicit temperature levels and heat requirements, to examine the identified heat pumps' energetic efficiency. Drier processes are chosen for consideration, as they are notorious for high inlet temperature requirements, release of waste heat and widespread application, a bullseye for heat pump application.[3] Contact with industrial participants and scenario analyses reveal lower process temperature requirements than initially expected, limiting heat sink temperatures to range between 110-185°C. Nevertheless, as heat pump application at reservoir temperatures exceeding 100°C is scarce, the temperature operating levels are shifted to obtain industry relevant results, without compromising usefulness of the performed analyses at this temperature level.

Based on the results of the performance evaluations, heat pump systems are proposed, with their potential energy recovery. Market available components capable of operating under the applied thermodynamic constraints are also identified, to indicate the current applicability of the proposed solutions. Emphasizing these aspects should be beneficial to spur the implementation of industrial scale heat pumps in high temperature operations, and potentially introduce heat pump systems to new markets.

III. Scenarios

The investigated scenarios are developed in collaboration with two industrial producers, to ensure that the obtained results coincide with the challenges of real world application of high temperature heat pumps. The first scenario is based on EWOS' production of fish feed. The second scenario is based on the spray driers processes at Tine Meieriet Verdal, utilized in whey production. Table 1 and figure 2, respectively, present the scenarios numerically and graphically.

EWOS: Production of pellet fish feed is the center of EWOS' operations.[4,5] Production processes require both heat supply and extraction to obtain the desired consistency, and is carefully controlled to customize the pellets to consumer specifications.

A high temperature heat supply (at 130°C) is needed to dry the pellets. EWOS uses air as their dryer medium. To avoid cumbersome filtration processes and emission control, EWOS circulates this air and reapplies it in the drying process. This requires a dehumidification process, to extract entrained water from the air exiting the pellet dryer (at 52°C). The moist air is cooled and most of the entrained water condenses. A heat recovery system reheats the air stream, but it can only achieve a temperature elevation to 63°C. The remaining temperature lift (67°C to the dryer inlet temperature of 130°C) is produced by a steam boiler. This is the heat sink evaluated for heat pump application in EWOS' production line. It should be noted that heat addition is bounded upwards at 110°C, as higher temperature levels require extensive retrofit of the process' air flow canals. Idealized operation requires a heat supply of 383.8 kW to elevate the moist air temperature from 63°C to 110°C.

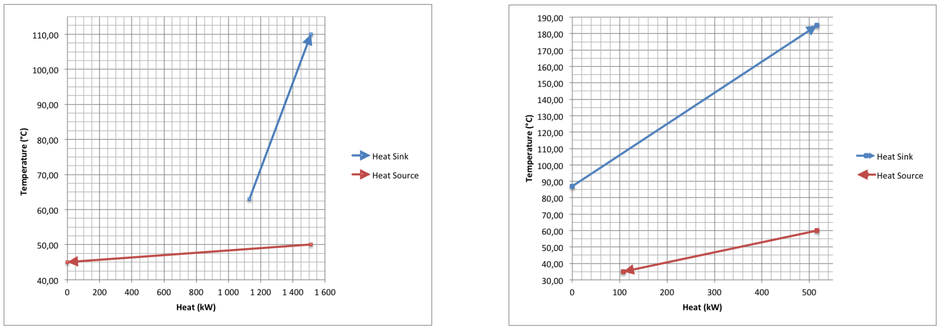
A saturated moist air stream, initially at 50°C, is identified as the prominent heat source for heat pump application. It holds a large amount of energy, and covers the heat sink requirements with ease after an elevation to useful temperature levels (refer to the graphical process descriptions in figure 2). It is a waste stream currently vented to the surroundings.

TINE: TINE produces two whey products, Whey Protein Concentrate (WPC) and Whey Permeate Powder. Spray driers are crucial process components in whey production, and require a high temperature heat input.[6,7] Whey is a powder product, and both TINE's spray driers use a hot, dry air stream to achieve correct powder dryness for storage and distribution.

WPC production requires a higher process peak temperature than permeate drying (185°C versus 160°C). Drier inlet air stream temperatures are produced by boilers (fired by primary energy sources), elevated from 87°C and 92°C,

High Temperature Heat Pump Application				
	Heat Source		Heat Sink	
Scenario	Temperature	Heat Release	Temperature	Heat Requirement
EWOS	50°C →	Not restrictive	63°C → 110°C	≥ 383.8 kW
TINE	60°C → 35°C	≤ 408.33 kW	87°C → 185°C	≥ 516.33 kW

Table 1: Scenarios for high temperature heat pump application.



(a) EWOS. *(b) TINE.*

Figure 2: Heat quantities and temperature glides of the identified scenarios.

respectively. Analyzing the operational heat pump performance in both heating processes is unnecessary, as the same working medium (dry air) is heated over similar temperature ranges. Hot air production used in the WPC drier is chosen for further evaluations, as it operates with the longest temperature span. The process air stream is heated from ambient temperature to 87°C by a heat recovery system (using direct heat exchange, passive heat recovery) and purchased district heat (active heat recovery). Heat pumps are introduced to heat the air exiting these systems, and ideally elevate the temperature to its spray drier inlet temperature of 185°C. The 98°C temperature lift requires a 516.33 kW heat load (with optimized performance).

Applied heat pumps can utilize a 408.33 kW heat source, cooled from 60°C to 35°C (flash cooling of a thermal vapor recompression system). The spray drier's outlet air (initially at 70°C, also assumed to be dry air) was considered as a potential heat source. However, the heat recovery system already utilizes this air stream, and lowers its temperature to 50°C before heat pump systems are applicable. (Passive recovery systems should not be removed due to their inherent energetic efficiency, with no need for external energy input.) This one significantly lowers the outlet air stream's recoverable energy, which renders it an inferior heat source compared to the flash cooler.

IV. Heat Pump Technology

Excess energy (heat) is recovered passively or actively. Passive heat recovery implies direct energy transfer between two temperature levels within a working system. Active heat recovery requires energy transfer from an external source to elevate low-grade energy to a higher level, and facilitates energy savings where conventional passive recovery is unattainable.

Heat pumps are active heat recovery systems, capable of reversing the spontaneous flow of energy from hot to cold energy reservoirs.[8] An external energy source, equal to the difference in energy between the reservoirs, is applied in the process (under idealized operating conditions). The applied energy is used to elevate the system temperature from cold reservoir levels to hot reservoir levels.

Heat pumps are broadly classified by their work input, driven by mechanical or thermal compression. Systems are further categorized as open or closed, indicating whether its working fluid is circulated or not.

The heat pump working fluid absorbs the energy input provided by the applied thermal/mechanical work, with a pressure and temperature increase. As it circulates the heat pump, useful heat is released to a designated heat sink and waste heat is recovered from a chosen heat source (both by conventional heat exchange systems). A working fluid's thermophysical properties have profound effects on the heat transfer dynamics and temperature elevations in a heat pump, and it must be chosen to complement a heat pumps working principle and the available thermal reservoirs. A brief investigation of prominent working fluids accordingly ensues the introduction of the basic operating principles of commonly applied industrial heat pumps. Extended system descriptions are provided in the Master's Thesis reviewed by this paper.[21]

Working Principles:
Open-Cycle Mechanical Compression Heat Pumps compress excess steam (the only operable working fluid) in a single-stage mechanical operation, and release useful heat by condensating against a high temperature thermal reservoir. Neither condensate nor cold vapor is circulated for recompression. [10]

Closed-Cycle Mechanical Compression Heat Pumps circulate a single working fluid, which extracts excess heat from a cold thermal reservoir (normally isothermally, by vaporizing). A mechanical compressor increases system pressure and temperature, to release high temperature, useful heat to a designated heat sink.[8]

Multistage Heap Pumps expand single-stage systems with two or more compression stages, as compound or cascade systems.[11] Compound systems compress the working fluid with a series of compressors, and operate with a single working fluid. Cascade systems are series of mechanical compression systems, where intermediate heat exchangers transfer heat between separate thermal cycles. This facilitates use of several working fluids with different thermal characteristics.

Expanding systems has two main advantages: increased energy efficiency over each compression stage, and a potential for higher temperature lifts.

Hybrid Heat Pumps are based on the principles of mechanical compression, but use a two component (with differentiated volatilities) working fluid mixture.[12] Low temperature waste heat desorbs the volatile component, and a compression stage increases the vapor pressure. A solution pump elevates the remaining liquid's pressure. Reintroducing the components at high temperature triggers an absorbtion process, which releases high temperature, useful heat.

Open-Cycle Thermocompression Heat Pumps mixes industrial excess vapor with a high-pressure motive stream, and use a steam jet compressor to increase the mixture's energy thermally.[10] Pressure elevations are produced by a conversion of the stream's kinetic energy. The high temperature stream condenses against a heat sink in an isothermal process.

Absorption Heat Pumps exist is two basic forms (type I and II), distinguished by their operable temperature levels.[8,13] Both systems produce temperature elevations thermally, assisted by a negligible mechanical work input. Two component working fluids are required (a volatile and a heavy component). Heat release/supply is attained through absorption/desorption processes, and by condensation/revaporization of the volatile component in gaseous form. A negligible mechanical work input is also required.

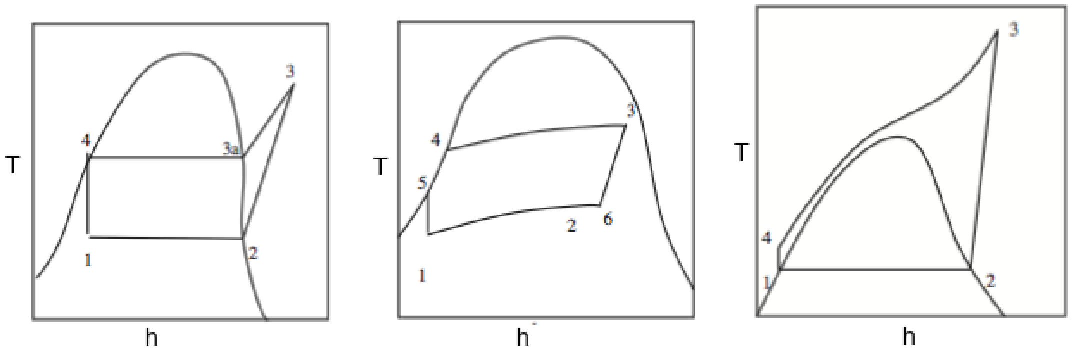
Type I systems (heat amplifiers) use a high temperature heat source to the produce a temperature lift of heat quantity supplied by a low temperature thermal reservoir. The supplied heat is released in its entirety at an intermediate temperature level.

Type II systems (heat transformers) use waste heat at an intermediate to increase system temperatures. A fraction of the heat supply is released at a high temperature, the remaining portion is disposed of at low temperature.

Operable temperature levels are summarized to differentiate the systems (subscripts S/R refer to heat supply/release, respectively): *Type I*: $T_S>T_R, T_R>T_S$; *Type II*: $T_R>T_S, T_S>T_R$.

Adsorption Heat Pumps are solid-gas sorption systems, used to release useful heat at intermediate temperature levels.[14] Desorption processes require a high temperature heat source. This implies operation as heat amplifying heat pumps, with thermal energy supplied by a high and a low temperature source, and delivered to the process heat sink at intermediate temperatures.

Chemical Heat Pumps use reversible chemi-



(a) Subcritical vapor compression. *(b) Compression-resorption cycles.* *(c) Transcritical vapor compression.*

Figure 3: Temperature-enthalpy diagrams of typical heat pump systems.

cal reactions to generate and/or store heat, by transforming thermal energy.[15] Systems are broadly classified as organic and inorganic heat pumps. Both are operable as heat amplifiers and heat transformers.

Inorganic systems typically involve solid-gas or solid-liquid reactions, most commonly applied as absorption or adsorption systems. Although other inorganic systems exist, no further introduction is provided, as such is not necessary for the purpose of this study.

Organic systems include hydrocarbons or hydrocarbon derivative working fluids, and take form as liquid-liquid, liquid-gas or gas-gas reactions. At least one endothermic and one exothermic reactor are required to qualify as heat pump systems.[16]

Working Fluids:
A multitude of different chemical compositions have been developed and applied with various degrees of success. Pure fluids, fluid mixtures and solid components are all applicable, and chosen to maximize heat transfer efficiency with the heat pump's thermal reservoirs. Heat transfer utilizes a change in the working fluid's latent heat (isothermally, by a phase change), sensible heat (thermally gliding, in a single phase) or both. Figure 3 [9] sketch three typical thermodynamic cycles, and the nature of their heat exchange.

A pure working fluid has a critical temperature and pressure [22] (ref. fig. 3 and the bell curves' peak temperature), above which it exists solely in gaseous form. Heat release at lower pressures has an initial temperature glide, required to form a saturated vapor (step 3-3a in fig.3a). This is, in turn, condensed to a saturated liquid by latent heat release (step 3a-4). This is known as subcritical heat release. Compression past the critical pressure implies that heat is released with a temperature glide, illustrated by fig. 3c and step 3-4, and referred to as transcritical heat transfer. These heat transfer effects must be accounted for in heat pump design.

Working fluid mixtures utilize sorption-based effects in heat transfer. Absorption/adsorption processes are exothermic, desorption is endothermic. Systems are designed to exchange heat isothermally (with azeotropic fluid mixtures) or with thermal glides (zeotropic mixtures).[23] Manipulating liquid/vapor concentration ratios across the heat exchangers creates boiling point trajectories, and is used to control the thermal glides (ref. fig. 3b). Temperature differences across the heat exchange are, however, not comparable to those of transcritical heat transfer.

Natural refrigirants are favored, due to beneficial environmental effects in operation.[23] CO_2 is already commonly applied in industrial applications, but presents unfavorable thermal characteristics in high temperature processes (extreme pressure requirements due to a low critical temperature). Ammonia and steam-water systems are accordingly prominent in current research. Short hydrocarbons show beneficial thermodynamic capacities, but are typically omitted in application, as they are difficult to handle safely. A lithium-bromide salt is commonly chosen when heavy aqueous solutions are employed in heat pump design, and in solid salt sorption system when they are deemed applicable.

V. Performance Parameters

Whereas all the identified heat pumps are applicable as high temperature industrial systems, section III and IV indicate a preferance for certain solutions, in terms of available sink/source temperatures and temperature glides.

The calculation tool accounts for two immediately identifiable sources of energy loss associated with heat pump systems; heat transfer effects between the heat pump and its surroundings, and production of the system's required temperature lifts.

The Coefficient of Performance (COP) quantifies the performance of a heat pump operating between two or more heat reservoirs, based solely on the energy input and output from a process. COP is defined as the ratio of produced heating effect (Q_H) to the net work input (W_{NET}) running the heat pump, and yield a theoretical performance through equation (1).

$$COP_{Real} \leq COP_{Heat} = \frac{Q_H}{W_{net}} \tag{1}$$

Equation (1) is simplified once the thermal reservoirs of a heat recovery system are identified, and maximized COPs are estimated by their respective temperature levels alone.[17] The simplifications are based on the power of reversible cycles and manipulations of temperature-entropy relations, enabled by the second law of thermodynamics. The reduced equations illustrate a heat pump design's dependence on available reservoir temperatures, and how manipulation of these affect their overall performance and applicability. Real system performance is always inferior to these estimations, but the obtained results are nevertheless valuable indicators of a heat pump's energy efficiency.

Isothermal Heat Transfer utilizes a working fluid's latent heat in heat transfer, and the system COP is estimated by equation (2). There are two available heat reservoirs, T_H represent the warm reservoir (heat sink) temperature, T_C the cold (heat source) temperature. A derivation of equation (2) is found in [17a].

$$COP_{Isothermal} = \frac{T_H}{T_H - T_C} \tag{2}$$

Heat Transfer with Thermal Glides accounts for changing temperatures across the system heat exchangers. A mathematical model is employed to estimate the average temperatures during heat transfer.[18] Inserting this model into equation (2) yields the COP of a Lorentz cycle (equation (3)), which estimates the performance of a heat pump with one or more thermal glides in its thermodynamical cycle.

$$COP_{Glide} = \frac{\overline{T}_H}{\overline{T}_H - \overline{T}_C} \tag{3}$$

$$= \frac{\Delta T_{H,g} \Big/ ln(\frac{T_H}{T_H - \Delta T_{H,g}})}{[\Delta T_{H,g} \Big/ ln(\frac{T_H}{T_H - \Delta T_{H,g}})] - [\Delta T_{C,g} \Big/ ln(\frac{T_C + \Delta T_{C,g}}{T_C})]}$$

ΔT_H and ΔT_C represent the temperature glide at the hot and cold heat exchanger, respectively, T_H and T_C the highest and lowest operational temperatures of the heat pumps. Equation (3) is also used when the system operates with temperature glides in only one heat exchanger, by setting ΔT=0 in the affected heat exchanger.

Thermal Heat Generation produces temperature elevations by chemical reactions, and takes several forms. Most systems operate between more than two heat reservoirs, which affect the COP calculations. Temperature relations are developed with type I and II absorption heat pumps as a vantage point. The systems represent, respectively, thermal heat generation by systems delivering heat at an intermediate level (referred to as heat amplifiers), driven by a high temperature energy source, and at high temperature (referred to as heat transformers), using an intermediate temperature energy source. Both assume the intermediate temperature heat exchangers to hold equal temperatures, and evaluates the performance of all sorption based heat pumps.

Equation (4) evaluates systems releasing useful heat at an intermediate temperature level, (5) systems releasing useful heat at high temperature.[9,19] T_I is the intermediate, T_H the hot and T_C the cold reservoir temperature.

$$COP_{Amplifier} = (\frac{T_{H:S} - T_{C:S}}{T_{H:S}})(\frac{T_{I:R}}{T_{I:R} - T_{C:S}}) \quad (4)$$

$$COP_{Transformer} = (\frac{T_{H:R}}{T_{H:R} - T_{C:S}})(\frac{T_{I:S} - T_{C:S}}{T_{I:S}}) \quad (5)$$

If thermally driven heat pumps operate with temperature glide(s), average heat transfer temperatures are estimated with the mathematical relation developed for equation (3). These equations are omitted, as their applicability in real systems is limited.

Energetic Efficiency: Closer investigations of each heat pump's efficiency is necessary to compare the heat pumps' performance to current heating utilities. A thermodynamic analysis based on mass and energy balances is performed for each component of a heat pump, once suitable working fluids are identified. The utilized mathematical model analyzing heat pump performance is simplified by the following assumptions[13]:

- Refrigerants and solutions are in steady state and thermodynamic equilibrium conditions at all states.
- Heat losses/gains and pressure losses are neglected unless directly related to the heat pump operations.
- Changes in kinetic and potential energy are neglected.
- Mechanical work consumed by pumps is negligible, as pressure lifts consume significantly less work when the working fluid's specific volume increases.
- Intermediate heat sources are considered to operate at identical temperature levels.

Equations (6) and (7) estimate heat capacities and work inputs in the main system components, based on changes in the working fluid's thermophysical state. Both are derived from the general energy equation, based on the above-mentioned assumptions.[17b] Enthalpies (h) are obtained from a computational program [24], once working fluids are selected. Subscripts i and e label the working fluid's inflow and outflow of each component.

Equation (6) assesses energy transfer in heat exchangers coupled with a heat pump system's thermal reservoirs. $Q < 0$ represents heat release to a designated heat sink, $Q > 0$ heat supply from a heat source. When the temperature lift is produced thermally, $Q > 0$ estimates the heat pump's thermal work input.

$$\dot{Q} = \sum_e \dot{m}_e(h_e) - \sum_i \dot{m}_i(h_i) \quad (6)$$

Equation (7) estimates the work input from mechanical compression of a working fluid. $W < 0$ as work is added to the systems. It indicates the energy requirement of any working fluid and mechanical operating principle, and can be employed whether the pressure increase is produced by compressors (when the working fluid is gaseous) or pumps (when the working fluid is in liquid phase).

$$\dot{W} = \sum_i \dot{m}_i(h_i) - \sum_e \dot{m}_e(h_e) \quad (7)$$

When the energy consumption of chemical heat pumps cannot be estimated by a circulated working fluid's enthalpy, isothermal enthalpies of transformation are used to characterize the heat release from endothermic (Q_{EN}) or exothermic (Q_{EX}) reactions.[20] Heat loads related to other sources, sinks or mechanical compressors in the system are estimated by the produced enthalpy changes, and calculated in accordance with equation (6) and (7).

Equations (1)-(7) estimate energy efficiencies under idealized conditions, chosen conciously due to the diversity of losses in heat pump components. The calculations are manipulated to mimic actual energy consumption, by the assumed component efficiencies.[9] (A boiler efficiency is actually derived in the case study of Tine Meieriet's production, and the specified boiler consumption compared to the energy requirements under idealized circumstances.) The obtained results are therefore easily comparable to the current energy consumption, and clearly illustrate the potential benefits of heat pump application. Implementation of the proposed heat pumps will, however, require further techno-economical investigations to determine its performance parameters accurately.

Large thermal driving forces and a relatively low compressor efficiency are chosen to strain the heat pumps' performance, a necessary precaution when merely rough discriptions of heat pump are considered. A suitable boiler efficiency is introduced from TINE's production process, where idealized boiler performance is identified as 75% of their specified energy consumption.

- Equal heat exchanger temperature driving forces: 15% of ΔT_{LIFT}
- Isentropic compressor efficiency (η_C): 0.6
- Boiler efficiency (η_B): 0.75

Air, the high temperature reservoir medium in both scenarios, is known to have poor heat transfer characteristics (a low specific heat capacity and low heat transfer coefficient), and research on heat pump application in hot air production is scarce. This warrants extended margins of error in numerical performance analyses.

VI. Preliminary Selections and Analyses

This study is limited by a preliminary selection from the wide assortment of heat pumps systems. Thus, the calculation tool can omit an uncertainty analysis, which otherwise would have been required to maintain a satisfactory level of precision in the results (if factors such as the economics of purchasing energy from multiple sources to run the heat pumps are accounted for).

The preliminary selection nominates heat pumps suitable as heat recovery systems in the identified industrial processes (see section III), using two key characteristics. Systems are categorized by their applicability with the available thermal reservoirs, and their working principle's expected energetic performance (estimated by equations (2)-(5)).

The characteristic feature of both industrial scenarios is the available heat source temperatures compared to the heat sink requirements. Current high temperature heat utilities consist solely of external boiler systems (gas fired or electrically powered). Working principles requiring a high temperature heat supply are accordingly discarded from further performance analyses, as purchasing a high temperature heat supply to run a heat pump system actually decrease the production processes' energy efficiency (due to system inefficiencies). The exclusion affects all heat amplifying systems.

Open-cycle heat pumps require steam (a scarce resource) to produce high-grade energy. Mechanical compression and thermally driven open-cycle heat pumps are therefore deemed difficult to apply without considerable process retrofit and design cost, yielding unreasonable payback periods.

Thermally driven heat transformers are applicable, and the required thermal reservoirs are readily identified. However, complexities arise as their COP is evaluated. Large temperature lifts separate the waste heat and the process requirements, and strain their applicability severely. The fractional heat release to the high temperature reservoir diminishes quickly, which increases component sizes and overall system cost in real application.

High temperature hot water (at $\sim$100°C) is available in both EWOS' and TINE's production processes, generated internally or purchased from district heating systems. Assuming this heat to be cheaper than the high temperature heat utilities is not unreasonable, and it is implemented as a potential heat source driving heat transformers. COPs of thermally driven heat pumps with hot water heat sources of 90°C (EWOS) and 102°C (TINE) are summarized in table 2, evaluated with isothermal and thermally gliding heat exchange.

Heat pumps based on closed-cycle mechanical compression are directly applicable in both scenarios, as their only operational requirement is two thermal reservoirs holding different temperatures. Singlestage and multistage compression, and hybrid heat pumps fall in this category. The COP analysis does not distinguish between these systems, as it ignores the effects of working fluid compositions. Table 2 summarizes their overall performance, with both isothermal heat transfer and temperature glides fitted to the key process requirements.

The net work requirements presented in table 2 are estimated by numerically obtained COPs and its relation to the desired heat outputs (ref. equation (1)).

VII. Performance Analyses

The results of the preliminary selection cannot be used to nominate a best-practice, energy efficient heat pump, but it does point to the limitations of each working principle. Thus, it is able to designate a set of most-likely-to-succeed concepts included in a closer performance examination.

Thermally driven heat amplifiers, alonside both the open-cycle systems, are deemed unapplicable by cumbersome process integration.

Preliminary Performance Evaluations [kW]				
	EWOS		**TINE**	
System	Idealized	Realistic	Idealized	Realistic
Boiler	383.8	511.7	516.3	688.4
Mech. Compression Heat Pumps	Heat Source: Original			
Isothermal	62.1	131.4	169.1	349.2
Thermal Glide	39.5	96.5	110.0	263.8
Therm. Driven Heat Transf.	Heat Source: Alternative			
Isothermal	505.3)	549.1	994.9	1080.5
Thermal Glide	387.9	435.8	707.8	812.8

Table 2: Preliminary performance analysis. Required heat and work inputs.

Table 2 indicates the extensive thermal energy requirements of heat transformers, where the heat input (although at a reduced temperature) significantly outweighs the current energy consumption. Attaining reasonable payback periods on system application is diffucult, and requires extensive techno-economical consideration to be determined accurately.

Mechanical compression heat pumps are accordingly identified as the preferable solution. Three basic system are identified to estimate the performance of heat pump application, and quantify the reduced energy consumption.

Single-stage mechanical compression of pure ammonia is least complex of the evaluated heat pump configurations, and designed to operate subcritically to avoid impossition of extreme system pressures. Ammonia is commonly investigated in the identified temperature range, due to preferable thermal properties compared to CO_2 and other natural refrigerants. Ammonia is vaporized to a saturated gas by the cold thermal reservoir. A single compression stage pressurizes it to a superheated pressure, at which it releases its heat to the desired heat sink. Heat release is initially sensible, then latent, and ammonia is finally depressurized to the evaporator's pressure level.

A two-stage cascade cycle investigates the influence of multiple compression stages and dual working fluids on overall performance, pressure and temperature levels. Research [25] suggests ammonia to operate the cold stage cycle, recovering heat from the available waste heat source. Steam-water cycle the hot side, and release the heat pump's useful heat. Both fluids operate standard, single-stage cycles, as that described for ammonia in the previous paragraph.

Water/ammonia is a prominent fluid mixture in development of high-temperature **hybrid heat pumps** as well. Ammonia is most volatile, and undergoes the sorption processes. High-pressure absorption release useful heat, after being pressurized from the low desorber pressure. The heat pump's thermodynamic processes are highly complex compared to the two pure fluid systems, but show promising characteristics and are currently investigated extensively to expose its potential in high temperature application.[26]

CoolPack identifies thermodynamic parameters relevant to the process cycles, used estimate the system's energy efficiency with equation (6) and (7). Sorption processes are, unfortunately, too complex to estimate without extensive simulations and omitted from the main performance analysis. However, comments on the expected performance parameters can be provided with relative certainty, and merit inclusion in this section.

System performance is evaluated and summarized with all the introduced thermodynamic inefficiencies accounted for. The applied thermal driving forces vary between system analyses, and are introduced accordingly. Compressor efficiency is evaluated as $\eta_C = 0.6$, and the boiler efficiency as $\eta_B = 0.75$. Ideal efficiencies are omitted, as they are unlikely to differ significantly from the results in table 2. A two-fold analysis ensues, separate for EWOS and TINE.

EWOS:
383.8 kW is required the air stream's temperature from inlet conditions (63.3°C) to its peak

at the outlet (110°C), and is key to successful heat pump implementation. A pinch point temperature is set to 7.5°C (assuming an ideal compressor efficiency), to ensure satisfactory driving forces throughout all employed heat exchangers.

A condenser temperature of 102.5°C was proposed for the single-stage heat pump, to maintain a manageable system pressure and limit the maximum system temperature (the compressor's outlet temperature). The high mass flow rate and low heat capacity of the treated air stream complicate the combination of sensible heat release and heat recovery (which requires vaporization of liquid ammonia), and support the chosen temperature levels.

Ammonia is pressurized from a saturated gas state at 41.5°C by a net work input of 120.4 kW. The superheated vapor exits the compressor at 65.6 bar and 200.9°C. A mass flow rate of 0.3441 kg/s provides a sensible heat release of 136.0 kW, and a latent heat release of 247.8 kW. This results in a slightly larger pinch point temperature (as heat transfer changes from sensible to latent) than 7.5°C, as the sensible heat load increases with accumulating compressor inefficiencies, but do not compromise the validity of the results.

Water vapor releases heat to the moist air stream in the cascade heat pump, and present a restrictive design factor. A high critical temperature implies that extensive latent heat quantities are available in the investigated temperature range. Several system designs are evaluated, and conclude that an elevated condenser temperature is beneficial to avoid excessive mass flow rates (due to the infinitesimal fraction of available sensible heat) which increase component size and system cost. The condenser temperature is accordingly set to 110°C.

Compression from a saturated gas state at 84°C elevates the water to a maximum temperature of 233.3°C, and consumes 44.8 kW in the process. Latent heat release of 322.2 kW brings the air to pinch temperatures (102.5°C), before a limited amount of sensible heat, 38.4 kW, elevates the air stream to an outlet temperature of approximately 107°C. The desired outlet temperature is not reached, but as a boiler system must be employed to elevate the air stream from 110°C to 130°C regardless of the applied heat pump performance, an additional 30.9 kW (accounting for the boiler efficiency) supplied of its bottom line is a valid settlement to the potential increase of compressor costs.

Ammonia recovers heat from the waste heat source, and elevates it from 41.5°C to 88°C, the intermediate heat exchanger temperature (which implies a 4°C driving force). This requires a net work of 82.0 kW, and produce the maximum system pressure (49.1 bar at the compressor outlet). This indicates a total external energy requirement of 126.8 kW to run the pump plus the additional 30.9 kW supplied by the boiler.

TINE:
516.3 kW is required to elevate the air stream from 87°C to 185°C. TINE uses a heat recovery system to preheat the air stream, which takes advantage of residual heat from the current boiler. Any losses in the recovered heat must accordingly be accounted for in the performance evaluations.

The single-stage ammonia heat pump operates with a condenser temperature is set at 120°C and 91.1 bar, elevated from an evaporator temperature of 44°C by a net work input of 100.3 kW. To avoid the strenuous effects of elevating the air stream to its peak temperature, mass flow (0.236 kg/s) is controlled to condense ammonia from saturated gas to saturated liquid, and heat the air flow to the chosen pinch temperature of 110°C with a 120.6 kW heat load. A limited amount of sensible heat is available (127.4 kW), and the boiler is required to complete the air's heat treatment (with a high temperature heat load of 268.3 kW). This produces an maximum internal temperature of 238°C in the heat pump.

The decreased heat recovery must not be forgotten, and amounts to 59.1 kW purchased as medium temperature heat. The total external heat requirement is, accordingly, 517.7 kW.

The cascade system's condenser is set at a midrange temperature of 140°C, and an intermediate heat exchanger temperature of 109°C/105°C, to avoid extreme water vapor temperatures exiting the compressor. An maximum vapor temperature of 287.3°C is nevertheless produced by a net compressor work of 37.7 kW. Combined with a pinch temperature of 10°C, this produce a peak air temperature of 136°C. The remaining temperature elevation is taken care of by a boiler utility, amounting

System Performance [kW]					
EWOS	Current Utility	Compressor Input	Boiler Consumption	Utility Requirement	**Net Reduction**
Single-stage (NH_3)	511.7	120.4	0	0	**391.3**
Cascade (NH_3 ∣ H_2O)	511.7	75.6 ∣ 41.8	34.4	0	**359.9**
TINE					
Single-stage (NH_3)	688.4	100.3	358.3	59.1	**170.7**
Cascade (NH_3 ∣ H_2O)	688.4	77.0 ∣ 37.7	344.3	61.5	**167.9**

Table 3: Expected consumption.

to 344.3 kW.

Ammonia cycle the cold subsystem, and elevates waste heat from 41.5°C to 109°C with a net work input of 77.0 kW. This produces the maximum system pressure of 74.3 bar at the compressor outlet.

The medium temperature hot water utility is required to compensate the decreased boiler utility with a 61.5 kW heat supply. A total external energy quantity of 520.5 kW is accordingly required to elevate the air stream temperature to drier inlet temperatures with the cascade heat pump system.

Table 3 summarizes the estimated energy efficiencies, and concludes that heat pumps are highly applicable in waste heat recovery. The single-stage compression system is superior to cascade application at in lower pressure applications (EWOS), as it is implemented with longer temperature glides (available in superheated cooling), and operable without thermal losses at an intermediate temperature (ref. the heat exchanger connecting the ammonia and steam-water subsystems). The former is imposed by the applied working fluids, and attributed to the significant difference in critical temperature and accordingly fluid states. The latter factor is accepted by cascade systems as a trade off against system pressure ratios, which affects the applicability and cost of currently available components. Energetic benefits of depressurized systems are accordingly visible when the internal pressure system increases (TINE), where the cascade system is superior to single-stage compression.

A hybrid heat pump is expected to present similar or improved energy efficiency in application, a statement supported by two beneficial system characteristics. *One*: the two-component mixture produces a boiling point trajectory as ammonia is absorbed/desorbed by water, attributed to their differing thermodynamic properties (ref. fig. 3b). Heat transfer efficiency is significantly improved by the achieved thermal glides. *Two*: the mixture releases heat at significantly lower pressure than pure ammonia, see fig. 4 [12], which decrease both compressor work and -strain. Net work requirements of pumping water to a higher pressure are small compared to the compressor's consumption, as pressure lifts require significantly less work when the working fluid's average specific volume increases.

The latter factor also ease application of market available components, as systems can be designed to operate within certain limiting temperature and pressure ranges by altering the component ratios in heat exchange. Figure 4 is easily maneuverable, and crude estimations of suitable working fluid mixtures are readily identified.

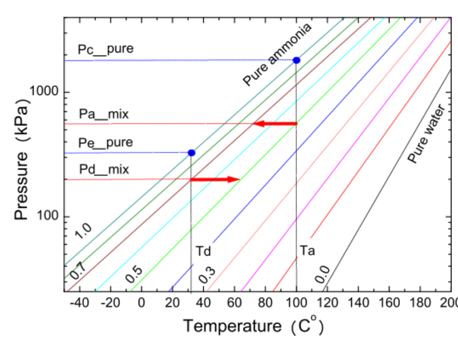


Figure 4: Effect of two component mixture in hybrid heat pumps.

VIII. Applicable System Components

The basic system components include heat exchangers, mechanical compressors, solution pumps and expansion valves. Neither heat exchangers, pumps nor expansion valves are assumed to be restricted in application by the extended pressure and temperature ranges, yet it is beneficial to reduce system pressure as it decreases the cost of component materials and construction.

Shell-and-tube heat exchangers are typically used in the condenser and evaporator, where latent heat is released (although plate-and-fin exchangers are applicable as well). Design and heat transfer bear the same characteristics in both. The working fluid flows tube-side, and counter-current to the treated process stream. As air is a poor heat transfer medium, baffles are used to increase its path through the heat exchanger. The hot side heat exchanger is, however, generally more expensive as it must be designed to withstand higher pressures during the (typically assumed) isobaric heat transfer processes.

Gas coolers, used to cool the superheated fluids to a saturated gas, are quite similar to shell-and-tube heat exchangers, but often implemented as a separate component to avoid pinch point problematics and improve heat transfer effects (reduce the negative effects of gas-liquid mixtures).

Due to their readily applicable nature, heat exchangers, as well as pumps and valves, are not treated further in this review. Available components with satisfactory performance are available from numerous manufacturers. Alfa Laval produces all the necessary components, and various designs are found in their online product catalogue.[29]

Compressors do, however, require a closer examination to determine the availability of applicable designs. Restraining factors are typically related to pressure, temperature, mass flow and the swept volume. (Both pressure and temperature reaches a maximum value at the compressor outlet.) Table 4 summarizes these parameters from the evaluated system designs.

Screw compressors are suitable for application, as they are significantly cheaper than the centrifugal systems at low shaft power outputs (less than 1 MW, based on a proposed power output/cost-relation with sampled compressors from four manufacturers - accurate enough for the purposes of this selection).[9]

Their operational temperature cannot exceed 250°C [27], but designs pressures exceeding 100 bar are available. GEA Grasso's Screw Compressor Series [28] are designed with a wide range of flow capacities and pressure ratios. Their AC Series is designed specifically for heat pump application with CO_2 and NH_3 as working fluids, and achieve maximum pressures of 130 bar. The AC screw compressor is operable with swept volumes ranging from 60 to 155 m^3/hr. The identified working fluid pressures (particularly prominent in the single-stage heat pumps, where it ranges up to 91.1 bar) are, accordingly, within the range of current components. Closer investigations are, however, necessary to confirm the applicability within the specified scenarios.

Oil-lubricants are often required in operation, to prevent leakage and lubricate moving parts to reduce wear and tear on the system. They are, however, often prone to deterioration

Strenuous Thermodynamic Parameters				
EWOS	$\dot{m}$ [kg/s]	T_{MAX} [°C]	P_{MAX} [bar]	V_s [m^3/hr]
Singlestage (NH_3)	0.344	200.9	65.6	60.8
Cascade (NH_3 | H_2O)	0.281 | 0.145	164.0 | 233.3	49.1 | 1.4	42.6 | 682.3
TINE				
Singlestage (NH_3)	0.236	238.0	91.1	43.2
Cascade (H_2O | NH_3)	0.199 | 0.105	217.2 | 287.3	74.3 | 3.6	37.2 | 456.8

Table 4: Relevant thermodynamic parameters, at the lowest examined efficency.

at higher temperatures and limit the potential output temperatures. Screw compressors are accordingly operable in steam-water compression without internal lubrication by pressurized oil. The working fluid act as a lubricant, and mechanically loaded pressure differences up to 12 bar are achievable. (It does come at a trade off with system cost.) The generic, oil-flooded compressors are operable with pressure differences up to 20 bar, and extended pressure ratios are achieved by multiple compression cycles.[27]

Water vapor compress at a low pressure, but requires extensive swept volumes exceeding 730 m^3/hr. Applicable systems are nontheless identifiable, and GEA Grasso's MC Series span both swept volumes presented in table 4.

The identified designs, with specified applicability in comparable thermodynamic ranges, support establishment of heat pumps as industrial waste energy recovery systems in the identified temperature ranges. Direct applicability cannot be promised, but systems should be within reach in the foreseeable future if analytical and experimental research identifies certain limiting characteristics and potential solutions.

IX. Further Work

Application of industrial heat pumps for high temperature heat recovery is a highly complex task, which requires extensive analytical work, experimental work and market research to be completed successfully. This thesis aims at identifying the most prominent restrictions, and appoints available heat pumps which challenge these restrictions. Accordingly, the further work springing from this thesis takes two forms:

- A reexamination of the proposed scenarios and heat pump technologies, with increased analytical accuracy, is necessary to obtain performance estimations with a real applicability.
- An examination of different industrial scenarios is necessary, in order to determine whether the approach used to identify heat pump technologies in this thesis, may be applied under other circumstances.

An important part of this thesis is the technical review, where the industrial scenarios, available heat pump technologies, promising working fluids and market available components are identified. The performance review is used to establish the characteristic features of different solutions, and provides a crude estimate of various performance parameters and the overall energy efficiency. However, it fails to quantify the performance of more complex heat pump technologies (ref. hybrid heat pumps, absorption heat pumps, where sorption processes are key to understanding and estimating performance). Extensive numerical work is necessary to obtain a manageable accuracy in the examinations. Thus, development of such a tool should be pursued, as the hybrid heat pump is assumed to hold a vast potential in high temperature industrial applications. (This is supported by the ongoing studies at a Danish research institute, in collaboration with industrial participants, expected to finish in February, 2015.)

X. Conclusion

Numerous technologies applicable as industrial heat pumps are identified, and examined within the confines of two selected scenarios, to combine actual performance evaluations with relevant theoretical aspects.

The results indicate that mechanical compression heat pumps are most suitable for heat recovery within the identified scenarios, and that both the examined heat pumps reveal a significant potential for increasing energy efficiency of the drier processes. The performance of a hybrid heat pump cannot be determined accurately by the calculation tool, but the reveiw indicates that it has beneficial characteristics in the examined operational range.

Compressors are expected to restrict high temperature application, and internal maximum pressure/temperature/capacity are the strenuous parameters. However, as GEA Grasso produces compressors which handle pressures exceeding those experienced in the identified processes, it may be possible to adapt operation to the utilized working fluids. Maximum temperatures cannot exceed 250°C, but interstage cooling can solve the problem. Further examinations are, however, imperative to validate application.

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