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High temperature heat pumps in dairy industry utilizing turbo compressor

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MASTER THESIS

for

Student Petter Olsen Rossvoll

Spring 2016

High temperature heat pumps in dairy industry utilizing turbo compressor

Høytemperatur varmepumper i meierinæringen basert på utnyttelse av turbo kompressor

Background and objective

The heat pump market has so far mainly focused on residential heat pumps for space heating and domestic hot water production. Less focus has been on heat pumps for higher temperature applications and industrial use due to high initial investment costs, competition with alternative investments, and non-mature or non-existing technologies for the applications to be served. New developments in turbo-machinery and compact high pressure components, e.g. compressors, ejectors and heat exchangers are important drivers for change of this situation.

The project will evaluate the application of turbo-machineries in industrial processes, which use steam as heat carrier. Low temperature excess heat will be used as heat source and the efficiency of the processes will be compared. Additionally the technological readiness level of the turbo-compressors for direct steam compression is evaluated.

The following tasks are to be considered:

1. Literature review of the main topic of the scope of work
2. Identification of relevant industrial process from the dairy industry
3. Determination and comparison of energy efficiency when turbo-machinery is applied
4. Evaluation of the technological possibilities for turbo compressors in high temperature applications for the Norwegian market
5. Make draft scientific paper of the main results from the work
6. Make suggestion for the further work

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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.

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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, scientific paper and all other material and documents should be given to the academic supervisor in digital format on a DVD/CD-rom or a memory stick at the time of delivery of the Master Thesis.

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

Department of Energy and Process Engineering, January 26th 2016

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CORRECTION OF THE TASKS TO BE CONSIDERED

From the original tasks to be considered, one task was added as a new task to be considered. The added task is formed as following:

3. Evaluation of correct heat pump placement in a relevant industrial process from the dairy industry using process integration as a tool.

Therefore the new tasks to be considered are:

1. Literature review of the main topic of the scope of work
2. Identification of relevant industrial process from the dairy industry
3. Evaluation of correct heat pump placement in a relevant industrial process from the dairy industry using process integration as a tool
4. Determination and comparison of energy efficiency when turbo-machinery is applied
5. Evaluation of the technological possibilities for turbo compressors in high temperature applications for the Norwegian market
6. Make draft scientific paper of the main results from the work
7. Make suggestion for the further work

These tasks are specified in agreement with the project supervisor.

Department of Energy and Process Engineering, June 20th 2016

Prof. Trygve M. Eikevik
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Preface

This master thesis is written to conclude the five year master degree program at The Department of Energy and Process Engineering at NTNU. The thesis investigates the energy efficiency of an existing UHT plant and the possibilities for integrating a water based heat pump with turbo compressors in the plant.

I would like to thank my supervisor Trygve M. Eikevik for his guidance during the writing of this thesis. I would also like to thank my co-supervisor Michael Bantle for his assistance on the topic. Lastly I would like to thank the staff at Tine Meieriet Aalesund for their hospitality during my stay there.

Petter Olsen Rossvoll

Abstract

There exist several high temperature processes in UHT plants where a great amount of heat is wasted. To avoid this can energy efficient equipment be installed, such as high temperature heat pumps, which can recycles this heat. Water is the high temperature working fluid that gives the best performance and is ideal in UHT plants because it can be used directly in heating treatment processes. However, it has its disadvantages in extreme low density and high compressor discharge temperatures. New development in turbo compressor technology is making this technology interesting in water based heat pumps, because it can to an extent cope with these flaws. Well functioning water based heat pump systems show good performance in this specific application, but the question is if its performance is enough to make up for the reliability of unproven technology. Heat recycling through optimal heat exchanger networks is commonly used in UHT plants, and the advantages of such implementations make it difficult for new equipment to penetrate the marked.

The existing UHT system, in addition to four new systems with several different modifications, were included in a simulation model to compare the performance of the systems for constant operating conditions. Two of these new systems included an identical heat pump with turbo compressors. Optimal heat exchanger networks were made for all systems, and the main goals were to minimize the combined heat and power consumption, maximize the usable system surplus heat and thereby minimize the net heat and power deficit in the systems.

Minimum net heat and power deficit of 24.04 kW were obtained by a modified system utilizing an optimal heat exchanger network without a heat pump. The heat pump implementation managed a COP of 3.826, and energy savings of 419.04 kW compared to existing steam boilers for the same heating purpose. The energy savings from heat pumps were however less than the potential usable surplus heat from optimal heat exchanger networks without heat pumps. Whether or not this surplus heat can be utilized is not known, whereas the savings associated with the heat pumps are guaranteed. Additionally are some systems gaining performance by diminishing flexibility, and are more vulnerable to change in operating conditions than the existing plant.

There is suggested to conduct further work with an energy analysis on this specific UHT plant with varying operating conditions. An investigation of the modifications made in this thesis is another relevant approach for further work. Additionally is development of cheap oil-free turbo compressor technology necessary before it can be implemented in this specific application.

Sammendrag

Det eksisterer en andel av prosesser på høye temperaturer i UHT anlegg, hvor store mengde varme blir forkastet. For å unngå dette kan energieffektivt utstyr, som høytemperatur varmepumper, bli innstallert for å gjenvinne denne varmen. Vann er det høytemperatur arbeidsmedie som gir best ytelse og er ideell for UHT anlegg fordi det kan bli brukt direkte i prosesser for varmebehandling. Likevel har det sine ulemper, som ekstremt lav tetthet og høy utløpstemperatur fra kompressor. Ny utvikling i turbokompressorer gjør denne teknologien interessant i vannbaserte varmepumper, fordi den kan til en viss grad håndtere disse ulempene. Velfunksjonerende vannbaserte varmepumper viser god ytelse i denne spesifikke applikasjonen, men spørsmålet er om dette er nok for å gjøre opp for påliteligheten til uprøvd teknologi. Varmegjenvinning gjennom optimale varmevekslernettverk er vanligvis brukt i UHT anlegg, og fordelene med slike implementeringer gjør det vanskelig for nytt utstyr å trenge seg inn på markedet.

Det eksisterende UHT systemet, i tillegg til fire nye systemer med forskjellige modifikasjoner, ble inkludert i en simulasjonsmodell for å sammenligne ytelsen til systemene for konstant driftsforhold. To av disse nye systemene inkluderte en identisk varmepumpe med turbokompressorer. Optimale varmevekslernettverk ble laget for alle systemene, og hovedmålet var å minimere det kombinerte varme- og kraftforbruket, maksimere den anvendelige overskuddsvarmen og dermed minimere netto varme- og kraftsunderksudd i systemene.

Minimum netto varme- og kraftunderskudd på 24.04 kw ble oppnådd av et modifisert system som utnyttet et optimal varmeverkslernettverk uten varmepumpe. Varmepumpene oppnådde en COP på 3.826, og energisparingene lød på 419.04 kW sammenlignet med eksisterende dampkjeler for den samme hensikten. Energisparingene fra varmepumpene var likevel mindre enn den potensielle brukbare overskuddsvarmen fra optimale varmevekslernettverk uten varmepumper. Hvorvidt denne overskuddsvarmen kan bli utrnyttet er uvisst, mens energisparingene fra varmepumpene er garantert. I tillegg oppnår noen systemer vinning i ytelse ved redusering av flexibilitet, og er mer sårbar for endring i driftsfohold enn det eksisterende systemet.

Det er foreslått å utføre videre arbeid med en energianalyse av det spesifikke UHT anlegget med varierende driftsfohold. En gransking av modifikasjonene som ble gjort i denne avhandlingen er en annen relevant tilnærming for videre arbeid. I tillegg er utvikling av billig oljefrie turbokompressorteknologi nødvendig før det kan bli implementert i denne spesifikke applikasjonen.

Contents

List of Figures

List of Tables

Nomenclature

Latin Letters

C	Heat capacity flow rate	kW/K
Cp	Specific heat capacity	$kJ/kg*K$
D	Steam	-
E	Collective term for heat duty and work	kW
h	Specific enthalpy	kJ/kg
H	Hot stream	-
K	Cold stream	-
$\dot{m}$	Mass flow rate	kg/s
M	Number of hot streams in one interval	-
N	Number of cold streams in one interval	-
P	Pressure/Product	bar/-
Q	Heat duty	kW
R	Amount of intervals	-
T	Temperature	$°C$
W	Power/Water	kW/-
x	Vapour quality	-
y	Mass fraction	-
Y	Annual energy use	kWh

Greek Letters

η	Efficiency	-
π	Pressure ratio	-
ρ	Density	kg/m^3

Subscripts

bp	Boiling point	-
$crit$	Critical	-
C	Cold	-
$carbs$	Carbohydrates	-
$compr$	Compressor	-
$cond$	Condenser	-
$corr$	Corrected	-

$evap$	Evaporator	-
g	Gas	-
H	Hot/Hot stream	-
HP	Heat pump	-
i	Index	-
I	Interval	-
inj	Injector	-
is	Isentropic	-
IW	Ice water	-
IWP	Ice water production	-
j	Index	-
K	Cold stream	-
l	Liquid	-
min	Minimum	-
P	Product	-
PH	Preheating	-
sat	Saturated	-
sub	Subcooled	-
sup	Superheated	-
TW	Tap water	-
W	Water	-

Abbreviations

CFC	Chlorofluorocarbons	-
CHP	Combined heat and power	-
CIP	Clean-in-place	-
COP	Coefficient of performance	-
CVC	Closed vapour compression	-
DH	Direct heating	-
EES	Engineering Equation Solver	-
GWP	Global warming potential	-
HCFC	Hydrochlorofluorocarbons	-
HEN	Heat exchanger network	-
HFC	Hydrofluorocarbons	-
HP	Heat pump	-
HVAC	Heating, ventilation and air conditioning	-
HX	Heat exchanger	-
MER	Minimum energy requirement	-
MVR	Mechanical vapour recompression	-

ODP	Ozone depleting potential	-
TVR	Thermal vapour recompression	-
UHT	Ultra-high-temperature	-
VHC	Volumetric heating capacity	-

1 Introduction

1.1 Background

The need for innovation and new solutions in the industry are massive due to heat energy waste. Numbers from Wolf et al. (2012) indicated that 14% of the industrial heat demand in Germany can be covered by recycling of heat energy with present technology, whereas this number could rise to 32% with new developments for higher temperatures. Energy efficient equipment are needed for heat waste recovery to be economical sustainable. Proven methods and technology for such use are already commercialized, but the demand for new technology to exploit a larger portion of the industry is evident.

An industry associated with large heat energy waste is the dairy industry. The energy consumption in the Norwegian dairy industry were in 2007 between 510 and 580 GWh per year (Helgerud et al., 2007). A large amount of this energy is utilized in heating treatment of the dairy products. Even though many of Norwegian factories have energy efficient heat treatment equipment, which recycles large quantities of heat, are the potential for more such equipment present. The recycling of heat is mostly done through heat exchanger networks, which is a commonly used method in all industries. However, when the recycled heat temperatures are too low to be utilized effectively, are other equipment more suitable.

An efficient way to recycle heat energy at high temperatures is the use of heat pumps. Commercial heat pumps are mainly used in refrigerators and in the HVAC market, but are also highly used in the industrial sector. The performance of industrial heat pumps is high on moderate temperatures, but they struggle when the recovered heat needs to be at high temperatures. The result of this is that industry usually does not recover heat on such temperatures.

In the present time has water emerged as a heat pump working fluid with high potential in high temperature applications. The properties of water create however difficulties for the heat pump compressors. Piston and screw compressors are the most common compressor types used in commercial heat pumps, but experience problems in large water based systems on high temperatures. Other compressor types can therefore be relevant for such systems.

Turbo compressors are an interesting option, and are already commonly used commercial in some industrial heat pump applications were usually water is utilizes as working fluid. These systems are often restricted to applications that require temperature lifts of maximum 30 °C, because sufficient performances cannot be achieved with higher temperature lifts. New developments in turbo machinery could however make it feasible to exploit applications with high temperature lifts.

1.2 Objective

The objective of this master thesis is to perform an energy analysis on an existing UHT plant for heat treatment of milk, and propose modifications which can make the plant more energy efficient. Several pinch analysis are to be done to the plant with and without modifications. These modifications can consist of both new heat exchanger networks and heat pump implementations. An energy balance will be done on the existing system and systems with modifications to compare the energy efficiency.

All implemented heat pumps are working on high temperatures and with water as working fluid. The applications of such heat pumps in a heat treatment process of milk is evaluated, and the energy efficiency is compared to other solutions. Additionally is the technological readiness level of turbo compressors for direct steam compressions evaluated.

1.3 Outline of Thesis

Chapter 2 contains a literature survey on heat treatment in the dairy industry, where ultra-high-temperature (UHT) treatment is emphasized.

Chapter 3 contains a literature survey on how heat pumps can be integrated in an industrial plant.

Chapter 4 contains a literature survey on industrial heat pumps with a focus on turbo compressors with water as refrigerant.

Chapter 5 presents the selected case for the thesis.

Chapter 6 presents the simulation model to the respective case.

Chapter 7 represents the results of the simulation model and evaluates the results through discussion.

Chapter 8 contains the conclusion from the results and suggestion for further work.

1.4 Delimitations

The Tine Aalesund factory logs both heat and power consumption in the factory. Since this thesis focuses around only one of several plants in the factory is the consumption of this specific plant impossible to obtain form the logging device. The heat and power consumption of utilities to the the specific plant are therefore calculated through several assumptions.

Because the temperatures of the plant are located from different sources, are the temperatures associated with some uncertainty. The specific heat capacity of all process streams are retrieved by assuming an average between inlet and outlet temperatures of the respective streams. Because some mass flow rates are retrieved from these temperatures and specific heat capacities, are also these rates uncertain.

There exist no available information of the vacuum tank in the UHT plant, which means that all existing processes within this tank are assumed. It is also assumed that an evaporating stream from this tank can be utilized in all new modifications, and if this is not possible are most results wasted.

2 UHT treatment in the Dairy Industry

All manufacturing of dairy products include heating processes. These processes are utilized to kill microorganisms, inactivate enzymes or sometimes to achieve other chemical changes. The results depend on the duration of heating and the temperature, which is defined as the intensity of the treatment. The main reason for the heat treatment is to warrant the safety of consumers, to increase the keeping quality and to establish specific product properties (Walstra et al., 1999, p. 189-190). The different heating processes in the dairy industry are:

- Thermalization
- Low pasteurization
- High pasteurization
- Sterilization

Thermalization is the treatment with lowest intensity, usually 20 seconds at 60-69°C. Low pasteurization follows with 30 minutes at 63°C or 15 seconds at 72°C, while high pasteurization is a more intensive treatment with 20 seconds at 85-100°C. The most intensive treatment is sterilization, which can be done by 30 minutes heating at 110°C, 30 seconds at 130°C or 1 second at 145°C. The latter two are examples of ultra-high-temperature (UHT) treatment (Walstra et al., 1999, p. 208-209). Because UHT treatment is emphasised in this thesis, will only this treatment be covered more thoroughly in this chapter.

The purpose of an UHT sterilizing plant is to heat dairy products to a temperature between 135 and 150°C and hold it there for a few seconds, before cooling it down to a much lower temperature. Chemical changes are minimized by utilizing the highest possible temperature for the shortest time possible. This means that the best processing effects on the product is obtained with the fastest rate of heating to the sterilizing temperature, and the fastest rate of cooling after the holding period. However, the industry also has to consider other variables as well, and especially economic factors like capital costs, operating costs and cleaning are important (Burton, 1994, p. 77).

UHT processes is either done with direct or indirect heating systems. The dairy products are heated indirectly if the process is done by a refrigerant in a normal indirect heat exchanger, which means that the heat is transferred by conduction through a surface that is separating the refrigerant and the dairy products. The refrigerant is usually either water liquid or steam. The dairy products are heated directly if steam and dairy products are mixed directly under pressure, where heat is transferred almost instant when steam is condensing. An important attribute with both heating systems is regeneration, where incoming dairy products are heated by outgoing dairy products. The principle of regeneration is shown in figure 2.1. Regeneration can only cover the first part of the heating process, while either indirect or direct heating covers the upper temperature levels.

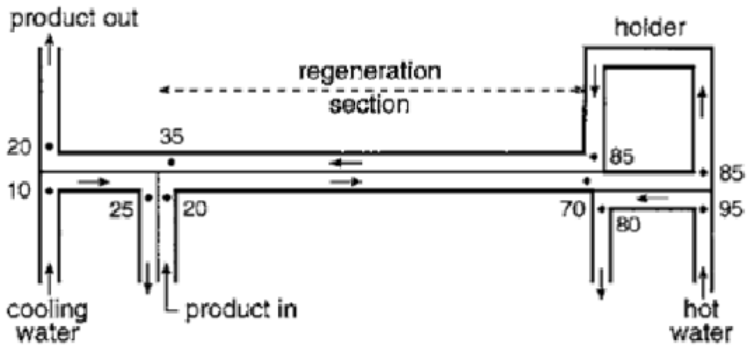

Figure 2.1: Simple illustration of a heat exchanger with regneration in a pasteurization process (Walstra et al., 1999, p. 228).

2.1 Indirect heating system

The plate heat exchanger is often used in indirect heating systems, and is advantageous because of its large specific heating surface, which results in a small temperature difference between the dairy product and the refrigerant. Another advantage is the low energy consumption due to the possibility for more regeneration than with other methods. A drawback with plate heat exchangers are their vulnerability when high pressure is applied. Cracks can be formed in the plates, which can result in leakage. Heating above 100°C requires pressures over 1 bar and this requires special construction of the plate heat exchanger (Walstra et al., 1999, p. 228-229). Plate heat exchangers for UHT processing are therefore developments from those which have been used at lower temperatures, where those used for UHT processes are implemented with more sophisticated and expensive gasket material to withstand higher processing temperatures and pressures (Burton, 1994, p. 81). Plate heat exchangers are only used in UHT processes when the products have a low viscosity, due to high possibilities for fouling when viscous products are processed.

There are some cases where plate heat exchangers are not suitable, where a tubular heat exchanger is a better option. Tubular heat exchangers have a larger flow channel than the plate type, which means that they can handle viscous products or products that contains high levels of pulp, fibers or particulate solids (Lewis and Heppel, 2000, p. 72). They have a smaller specific heating surface than plate heat exchangers, and the temperature between the milk and the refrigerant will therefore be greater and more heat exchanger losses will occur. High pressure needs to be applied in this type of heat exchanger to avoid fouling and to improve the heat transfer. This causes no problems for tubular heat exchanger, which means that very high temperatures can be applied. They are therefore excellently fit for indirect UHT treatment with temperatures up to 150 °C (Walstra et al., 1999, p. 230).

2.2 Direct heating system

UHT treatment is often done with direct heat exchangers, where the steam is injected directly into the milk or the other way around. An almost instantaneous heating to the desired temperature occurs. The water steam, which needs to be of high purity, will directly condense, and maintained in the milk for a few seconds, before it is discharged in a vessel which reduces the pressure. An almost instantaneous evaporation of water occurs causing rapid cooling. The amount of water evaporating should be the same amount as the condensing water in the heating process. Direct heating is normally just applied as the final stage of the heating. Typically, regeneration heats the products up to around 80°C and direct heating from 80°C to the sterilizing temperature. Direct heating can either be done by injection, also called steam-into-product type, or by infusion, also called product-into-steam type (Datta et al., 2002).

The injection or steam-into-product system includes an injector which is the heart of this system. Steam is here pressurized and injected into the product stream through a suitable nozzle. The most important requirement is the fast condensation of steam, to give rapid heating, and to prevent bubbles of uncondensed steam into the holding area. If a passage of bubbles appears in the holding tube, this would reduce the holding time at the sterilization temperature by displacing liquid and reducing the effective volume of the tube. Rapid condensation is achieved by introducing the steam into the liquid in the form of small bubbles or the form of a thin sheet. Another requirement when this method is used, is to limit the operation costs by having a low pressure difference between the liquid and the steam. This will give lower costs linked to steam generation and heat losses. A phenomenon which typically occurs in injectors when bubbles are formed, is rapid collapse of these bubbles during condensation, which causes local changes in pressure in the liquid and can create high noises. Some form of sound insulation can therefore be appropriate (Burton, 1994, p. 105).

The infusion or product-into-steam system is similar to the injection system in all aspects except how the mixing of product with steam is done. There is a steam pressure vessel with a conical base into which the heated product falls and from which it passes to the holding tube. The size of these vessels and the methods for product distribution can vary a lot in the industry, but the product heating time is rarely more than half a second, making the size and distribution factors negligible on the sterilizing effect. Because of the infuser shape, the volume of a bottom pool of products adds time to the holding period, and will be determined by the flow rate through the holding tube. The volume of this pool must therefore be minimized by careful level control (Burton, 1994, p. 111-113).

The heating process of an infuser is generally gentler than for injectors since infusers are not associated with bubble condensation and high noise level. Infusers have always a larger size than the equivalent injectors. The dairy product distribution systems in an infuser are mechanically equal to a steam injector, but since an infuser also consist of a pressure vessel, the capital costs are generally larger for infusers. The heating time for injectors is typically just under a second, while infuser as mentioned earlier normally heats the product in half a second. The heating time for both systems appear to be approximately similar, and is moreover short compared to the holding time. This holding time is more predictable for injector types than for infuser types. More factors play a role in the holding time when infusers are used, including the pool of product at the bottom of the vessel (Burton, 1994, p. 123-125). Figure 2.2 illustrates simple operation differences between an injector and an infuser.

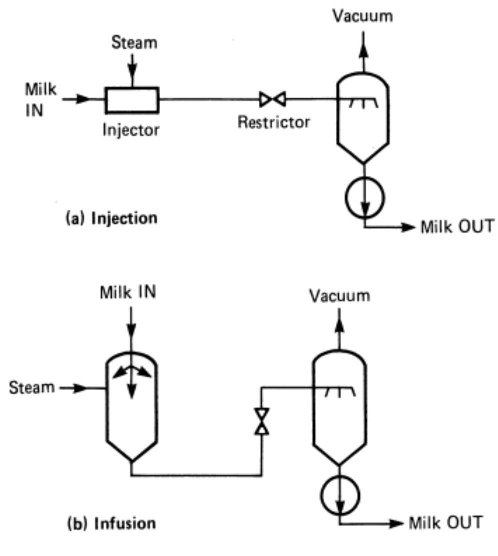

Figure 2.2: Simple illustration of (a) injection and (b) infusion systems (Burton, 1994, p. 104)

2.3 Comparison of indirect and direct heating systems

The two types of system differs especially with direct systems capability of high rates of heating and cooling. UHT processing, where high temperature is supposed to be held for short times, is for that reason simpler to obtain with direct systems. There are also product quality benefits with direct heating, but economic disadvantages as well, which is the most influential factor in choosing between the two systems (Lewis and Heppel, 2000, p. 101).

Direct heating is relatively complex, and is associated with both higher investment and operation costs than indirect heating. A number of components are customized and specially fabricated, unlike indirect systems with standard heat exchanger plates and tubes. Direct systems are also in need of more equipment than indirect systems, such as expansion vessels and condenser in addition to more pumps and control systems. In 1994 were a direct plant associated with twice the capital cost as an equivalent indirect plant with a plate heat exchanger (Burton, 1994, p. 124). The operation costs in a direct system is higher than an indirect system because of the restricted possibilities of heat regeneration. A direct plant can typically regenerate 50-60%, while this number is around 90% for indirect plants (Lewis and Heppel, 2000, p. 102). The higher complexity and costs associated with a direct system makes an indirect system superior in most cases. A direct system can only be justified in special cases, where product characteristics and quality make its use desirable, which is especially common for viscous products.

3 Heat pump Integration in Dairy Industry

For new installations of energy efficient equipment there will be a variety of feasible solutions and the best solution for a specific plant will depend on a number of factors. This is true for any sector of process industry, but especially in the food industry were the number of factors to be balanced is more than ordinary process plants. Nicol et al. (2005) categorize in particularly four interacting factors to select best practice technology. These factors and what they include are represented in table 3.1.

Table 3.1: Interacting factors for best practice technology (Nicol et al., 2005)

Product	Operational	Capital	Energy
Food safety	Flexibility	Capital cost	Energy cost
Product quality	Scale	Cost of control technology	Relative cost of fuels
Product functionality	Product changes	Existing capital infrastructure	Peak demand charges
-	Cleaning	-	Supply infrastructure

For instance could product changes interact with the energy efficiency in a plant. If energy efficient equipment is customized for a special operation state, it will perform worse if the operation is changed with product changes. Another example is the dependency capital costs and scale of new equipment has on energy efficiency. Small plants with less space and capital are less likely to invest in new costly energy efficient equipment, whereas big plants with large capital would have the opportunity to do so. A typical measure for increasing energy efficiency in the industry, and thereby also operation costs, is recycling of heat. This is normally done with a methodology called process integration, and the result of this methodology would decide which new equipment that fits the specific plant. Examples of such new equipment as a result of process integration are heat exchanger networks, CHP units and heat pumps.

The energy factor from table 2.1 is especially important in Norwegian industry. Greenhouse gas emissions taxes are high compared to other countries (Bruvoll and Larsen, 2004), which implies higher costs of fossil fuels. Norway is blessed with hydro-electric opportunities and the electric energy costs are generally low and stable. This reduces the electricity-to-fuel price ratio, and makes industrial heat pumps an interesting choice for energy efficient purchases. It is said that if the electricity-to-fuel price ratio is above three no general applicability of electrically driven compression heat pumps is expected (Wallin et al., 1990). Industrial heat pumps are associated with high capital costs, and a minimization of operational costs is therefore vital to obtain a low return of investment.

Heat pump systems are rarely used in the dairy industry. In 2004 had in fact just Canada, Norway, the Netherlands and the US installed heat pumps in dairy plants. Moreover, the number of plants with installed heat pump systems in these respective countries is relatively low. A more common solution in the industrial sector is the installation of heat exchangers to utilize exhausted process streams, which is a solution associated with less investment cost and overall risk than heat pumps. It is well known that the industry is slow to invest in new equipment when it already exists a profitable solution in the specific plant (Ozyurt et al., 2004). However, energy saving measures are nowadays vital for industrial plants to be competitive, and the continuously discovery of more efficient and liable high temperature heat pump technology makes it an interesting option for heating processes in the dairy industry.

3.1 Process Integration - Pinch Analysis

Where there are many process streams at differing temperature levels it will be appropriate to use process integration. Process integration involves a method called pinch analysis, which is highly used in the industry. The pinch method is a simple methodology, which do not require complex mathematical calculations. It gains energy advantages from improving the integration of processes, often by developing simpler and more elegant heat recovery systems. Improvements can be found not only from heat recovery projects, but also from changing process conditions, improving operability and more effective interfacing with utility systems (Kemp, 2007, p. 2-6).

The first step when using the pinch analysis is to locate the hot and the cold streams. Hot streams are usually process streams with an unused heat potential exiting an industrial process or streams within a process that requires cooling, whereas cold streams are usually streams that requires a temperature lift before entering or within an industrial process. The second step of the Pinch analysis is to figure out the temperature levels of the respective streams in both entering and exiting state, in addition to the specific heat flow rates. An example of how hot and cold streams are typically categorized and the required information to perform calculations is shown in table 3.2. For simplicity are only two hot and two cold streams included. ΔT

is the temperature range of the streams, C is the heat capacity flow rate of the streams, whereas Q_{load} represents the heat load of each stream.

Table 3.2: Typical representation of hot and cold streams.

Hot/Cold	ΔT [°C]	C [kW/K]	Q_{load} [kW]
Hot	$40 \leftarrow 120$	-2	-160
Hot	$90 \leftarrow 150$	-2.6	-156
Cold	$10 \rightarrow 110$	1	100
Cold	$70 \rightarrow 140$	3	210

Before going any further, a minimum temperature difference ΔT_{min} is chosen, which is included in the analysis to ensure that within any interval, hot streams and cold streams are at least ΔT_{min} apart. This value is decided through a cost minimization of capital costs and operational costs. Operational costs is smaller with a low ΔT_{min} because this gives a larger amount of heat transfer, whereas capital costs is decided from the heat exchanger area, which again is inversely proportional to ΔT_{min}.

After this optimization is done, the hot and cold composite curves can be plotted and the temperature interval method could be used to find pinch temperature (Kemp, 2007, p. 19-22). An illustration of how the composite curves typically can be arranged is shown in figure 3.1, where a ΔT_{min} of 10 kelvin is applied. Table 3.3 is a representation of the temperature interval method for a system consisting of the streams from table 3.2. T^* represents the temperatures that define the intervals, Q_I is the heat surplus or deficit in a specific interval I, ΔQ_I is the cumulative heat surplus or deficit from interval 1 to interval I, while $\Delta Q_I{}^*$ is a variable created to identify the pinch points, in addition to hot and cold utilities.

By studying the composite curves in figure 3.1 it is evident that the hot streams transfers all its heat to the cold streams over the pinch point, making only heating required in this region. The opposite phenomenon occurs under the pinch point. Table 3.3 contains two important values, i.e. the minimum required heat demand over the pinch Q_{Hmin} and the minimum required cooling demand under the pinch Q_{Cmin}. These values illustrates the heat demand and cooling demand not covered by process integration. The pinch point can also be located in table 3.3, and it can simply be said that this point divides the process into two parts – a heat sink above it and a heat source below it. The total pinch analysis gives three rules to achieve minimum utility targets, which is described by Kemp (2007, p. 27) as:

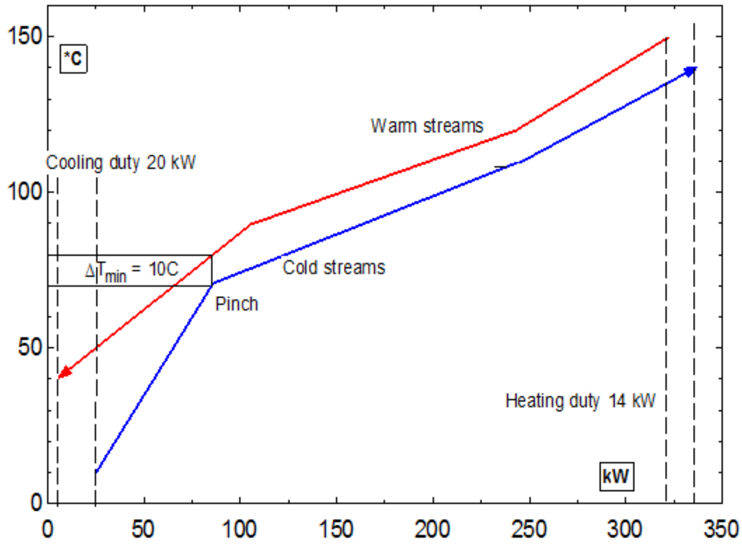

Figure 3.1: Illustration of the composite curves from the example in 3.2.

Table 3.3: Temperature interval method for the streams in table 3.2

I	T^* [°C]	C_I [kW/K]	Q_I [kW]	ΔQ_I [kW]	$\Delta Q_I{}^*$ [kW]
				0	14 (Q_{Hmin})
	145				
1		0.4	12	12	2
	115				
2		-0.6	-18	-6	20
	85				
3		2	20	14	0 (PINCH)
	75				
4		-1	-40	-26	40
	35				
5		1	20	-6	20 (Q_{Cmin})
	15				

- Do not transfer heat across the pinch.
- Do not use cold utilities above the pinch.
- Do not use hot utilities below the pinch.

With the knowledge from the first steps of the pinch analysis and the rules to achieve minimum utility targets, a heat exchanger network (HEN) can be designed. In this process is it necessary to divide the problem at the pinch, and design the two parts separately. Another necessity is to start the design at the pinch and moving away from it, because the streams from each side of the pinch should be be brought to pinch by other streams and not external utilities. The last requirement yields that if one hot stream with C_H shall bring a cold stream with C_C from somewhere below the pinch and up to the pinch point through a heat exchanger, must C_H be greater or equal to C_C. The opposite will always apply above the pinch point. If this is not the case, will this heat exchanger always break the criteria of ΔT_{min}. Splitting of streams can be necessary if this criteria is not met (Linnhoff and Hindmarsh, 1983).

The capital cost of heat exchangers is dependent on the surface area and the amount of heat exchangers. All though it is possible to design several different systems, the energy recovery should theoretically be the same in each case, making the surface area equal in all cases. In the early stages of the design is therefore a minimization of units of utmost important, since more units require more expenses in pipelines, foundations and maintenance (Linnhoff and Hindmarsh, 1983). Figure 3.2 is a typical HEN solution of a system consisting of two hot and two cold streams represented in a grid diagram. The temperatures of all streams are placed on each side of the respective stream or between heat exchangers, while the heat transferred in each heat exchanger is located next to the respective heat exchanger. The heat capacity flow rate $C_{H/K,i}$ of each stream is located on the right hand side in the diagram.

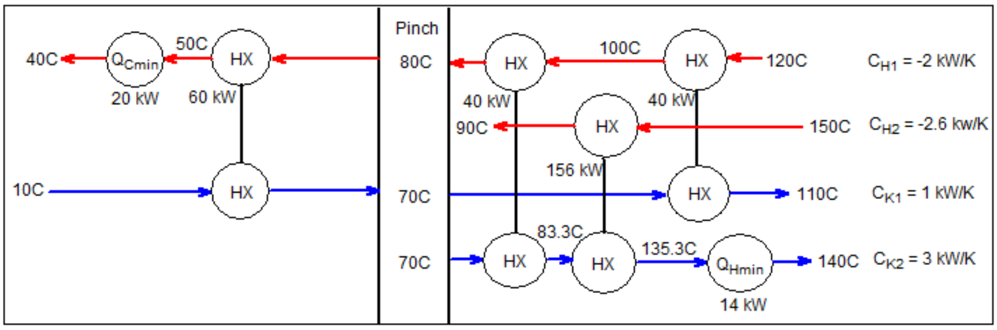

Figure 3.2: A typical HEN of a system consisting of two hot and two cold streams.

In this solution is the minimum energy requirement and the maximum energy recovery achieved, which make this solution a MER network. However, the network design is rarely so easy, and trade-offs are usually made to reduce the number of heat exchangers at the expense of increasing some utility load (Kemp, 2007, p. 34). Moreover, limitations in the specific plant could also deny the possibility of a MER network, such as limiting space and reassurance of product quality.

There can occur special cases where the pinch method is shown to be a poor methodology, like cases where thresholds problems occur. A threshold problem occurs when either the external heat input or the external heat output disappears. This is a situation where the energy need is not affected by the position of the composite curves. However, the exergy need is strongly affected, due to different temperature levels. An exergy analysis can in this case give an insight, which is lost in the pinch method (Wall and Gong, 1996).

Usage of an exergy analysis could also be valuable in a general case. The pinch method centers primarily on maximizing the internal heat transfer with the choice of appropriate ΔT_{min}. A proposed extension of the pinch method takes into account the complete heat transfer exergy losses, the pressure drop exergy losses and the exergy associated with manufacturing of the heat exchangers. Such an extended exergy synthesis results in an improved and more coherent exergy balance for comparing exergy recovery schemes. It offers a new insight and permits the identification of solutions which are more stable in time and fairly independent of changing economic conditions (Staine and Favrat, 1996).

3.2 Heat pump Installation through Process Integration

After a pinch analysis of the existing process streams is done, an appropriate placement of heat pump can be conducted. There are four methods to integrate a heat pump in industrial plants, and there exists different rules for integrating the different methods. The four methods are defined by Ranade (1988) as:

- Utility to Utility
- Process to Utility
- Utility to Process
- Process to Process

Utility to utility heat pumps are normally used to create levels of steam from existing lower level of steam. They are most likely to be appropriate in situations where an existing facility is purchasing a certain level of steam, but through process changes a demand of slightly higher level of steam occurs. Process to utility heat pumps should take the surplus heat from below the pinch and use it to generate steam, whereas a utility to process heat pump could be used above the pinch to reach higher temperatures from an existing utility. The most common solution, which is emphasized in this thesis, is the process to process heat pump, where energy savings can only be realized by placing the heat pump across the pinch. This means that a heat pump can take heat from below the pinch and reject it at a temperature above the pinch temperature. Surplus heat available below the pinch will then be made to supply heating required above (Townsend and Linnhoff, 1983). However, there has been studies on integrated heat pumps that not crosses the pinch point, which have concluded that this could be economically interesting when heat exchange gets expensive due to large distances between streams (Berntsson, 2002).

To decide if a heat pump should be implemented in an industrial plant, there needs to be an assurance that it will decrease the annual costs in the plant. The potential for heat pump integration can be analysed from the composite curves. If these are open, the potential is large for an efficient heat pump implementation, whereas closed curves would make the installation less obvious. Open curves are typically characterized by one or more evaporating or condensing stream, which is also the case in the open curve in figure 3.3. One case of open curves and one of closed curves are here illustrated, where the difference in energy savings by a heat pump implementation clearly can be observed. Other factors, which affects the potential of heat pump installations, are the electricity-to-fuel ratio, the annuity factor, the absolute level of energy prices and the specific heat pump cost (Wallin et al., 1990).

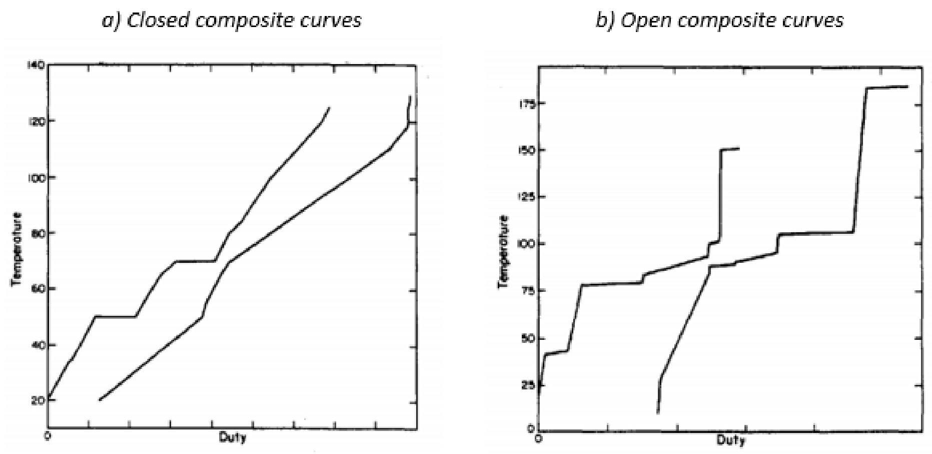

Figure 3.3: Difference between heat pump potential for respectively open and closed composite curves (Wallin et al., 1990)

The composite curves restrict the heat loads to be extracted and delivered by the heat pump. The limit is located where the utilized heat makes a new pinch point.

When this limit is exceeded, the composite curves are forced apart to maintain ΔT_{min}. ΔT_{min} at the original pinch point is now higher than it originally was, and a new pinch point or points are denoted by the heat pump. The original pinch point is however still the limit where no heat is allowed to be transferred, due to the fact that this pinch is made by the system itself and not by the heat pump. Another important aspect, is the change in heat exchanger network, which the implementation of a heat pump creates. The amount of heat, which is extracted from streams and delivered to other streams by the heat pump, is no longer available for heat exchange with other process streams (Wallin et al., 1990).

When a heat pump is integrated through pinch analysis occurs a question about the ideal heat pump temperature lift. A normal approach is to determine the temperature lift regarding COP, but research concludes that COP of a heat pump alone cannot be the sole criteria. In most cases is it not energy savings, but the reduction costs associated with energy that is the goal for process integration and heat pump implementation. A method is therefore suggested, which limits the temperature range within which the optimum heat pump must lie based on an economic approach (Ranade, 1988).

3.3 Characteristics of Process Integration in Dairy Industry

There are several characteristic aspects for process integration in the dairy industry, which are caused by product requirements like hygiene, product safety, nutritional value and product quality. The non- or semi-continuous operation of most dairy plants creates an additional constraint, which again makes a normal Pinch analysis challenging. Process integration in dairy plants can therefore be approached in a number of ways, and several novel pinch methods are developed to serve for different constrains (Atkins and Walmsley, 2013).

One of the required measures making dairy plants non- or semi-continuous is cleaning. CIP or cleaning-in-place is vital in a dairy plant, mostly because of product safety. The intervals between each wash are vital to a number of factors, like production rates, energy and water consumption, and are typically 6 to 12 hours for a pasteurization process. In plants with several individual processes, the cleaning schedules are normally following production, and for that reason is there no scheduled planning of CIPs between processes to improve energy efficiency. To integrate separate processes is therefore difficult because the processes will not always coincide with each other. Location could be another limitation in the integration process. There is not always space for pipelines across a factory, and if these pipes are long, there could be another feasible solution which would function better in practice (Atkins and Walmsley, 2013).

Process integration for plants with a number of semi-continuous processes can be done with the pinch method, but some adjustment needs to be done. For instance are the heat flow rates of the streams in this case the average rates over a period of time, and these values could therefore vary some within this interval. All though the composition curves can be made from the pinch analysis, it is still unknown which streams that can be connected due to the semi-continuous operation. An approach by Walmsley et al. (2013) suggests the use of a heat recovery loop instead of a heat exchanger network, which means that excess heat from processes can be delivered to other processes by a thermal storage. The thermal storage consists of an intermediate medium that acts like a heat transfer medium that is stored and recirculated as needed. This medium is normally water at low temperatures or oil at higher temperatures. The approach was tested in a case study with a dairy factory containing several heat treatment processes, and proved large annual savings.

Non- or semi-continuous processes create problems for heat pump installation in dairy plants. It is obvious that heat pumps cannot perform if the processes delivering heat and the processes receiving heat are not operating simultaneous. Atkins and Walmsley (2013) wrote that a heat pump installation within one specific individual process would not include this problem, in addition to having a lower threshold to overcome in order to be economic. The heat treatment process of dairy products is a typical process were there is possibilities for a heat pump installation due to the requirement of both heating and cooling. The heat pump could be placed as a cooling device for the dairy product stream exiting either the heating process or the regeneration process and deliver heat as a part of the heating process. Ozyurt et al. (2004) tried such an approach in a pasteurization process by using the heat pump evaporator to cool the dairy products to 34°C and the heat pump condenser to heat the products to 72°C. The system showed sufficient performance with a primary-energy-saving of 66%.

There are in dairy plants typically a number of process streams that contains water at different temperature levels. The water consumption in a dairy factory is huge, and a cost reduction can be obtained by minimizing this consumption. It is therefore been developed methodologies that combine heat integration and mass integration. Such methods were used by Becker et al. (2009) and a calculated reduction of 25 % water consumption were obtained. Despite these promising calculations, there has to this date been no published studies of this method being applied in the dairy industry. There could be various reasons for this, like the complexity of the method, but it is certain that the industry requires case studies before applying the methodology (Atkins and Walmsley, 2013).

4 High Temperature Heat Pumps

High temperature heat pumps are mostly used in the industry, where high temperature processes can be used as heat source for a heat pump system. Most of the industrial heat pumps are working on moderate temperature levels, but there are huge potential in utilizing processes on higher temperature levels. These industrial heat pump systems are categorized in different types, depending on how they operate. Several different types of systems exist, but the most used types are categorized by Wolf et al. (2012) as:

- Closed vapor compression systems (CVC)
- Mechanical vapor recompression systems (MVR)
- Absorption cycles
- Thermal vapor recompression (TVR)

MVR systems represented in 1997 50 % of all installed industrial heat pumps, while CVC applications represented 45 % (Berntsson and Franck, 1997, p. 16). CVC and MVR systems are most relevant for dairy treatment applications. and will for this reason be discussed further in this report.

4.1 Closed Vapor Compression System (CVC)

The easiest CVC system consists of four different components, which are an evaporator, a compressor, a condenser, and a throttling valve. In an industrial application, the system utilizes energy in an industrial process to evaporate the refrigerant flowing in the system before the refrigerant pressure increases through the compressor. The refrigerant enters then the condenser were it condenses on a higher temperature level, and the cycle ends with the expansion through the throttling valve, and the refrigerant is again at the lower pressure level. Figure 4.1 illustrates a simple CVC system.

The system used in applications in the industry today are usually more complicated than figure 4.1 to optimize the cycle performance. Several stages of compression and throttling are commonly used, in addition to either intercooling components or internal heat exchangers depending on the working fluid. A cascade heat pump system is also an option for high temperature lifts, where two or more CVC cycles are connected by using the heat energy from the condenser in a low temperature cycle to evaporate the fluid in a higher temperature cycle. Another option is a trans-critical system, where the refrigerant is compressed up to trans-critical levels and the refrigerant releases its energy through a gas cooler instead of condensing.

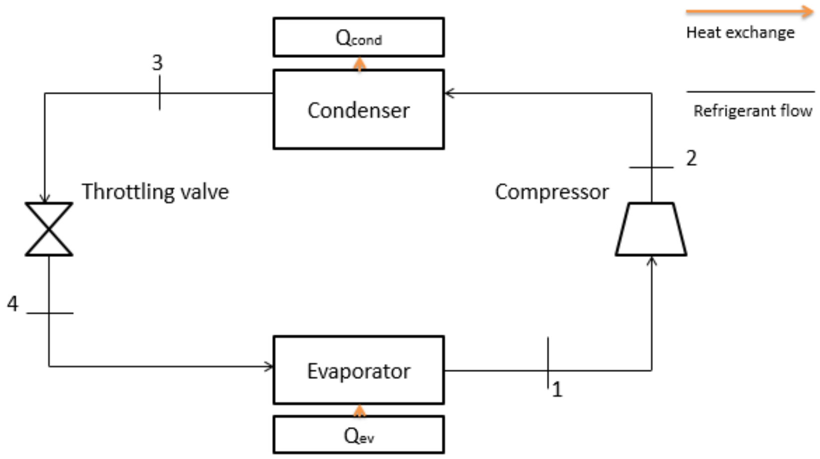

Figure 4.1: Simple closed vapor compression cycle

Most application of closed industrial heat pumps are working on temperature levels less than 50°C. The closed system had in 1988 a maximum temperature level at 120-140°C due to thermal degradation of oil lubrication in compressor and limitation of the refrigerants (Gluckman, 1988, p.64-65).

4.1.1 High temperature refrigerants

When high-temperature refrigerants are compared to each other, there are some properties that are especially important to consider due to their impact on how the heat pump system will operate. The critical pressure P_{crit} is important, since the superheating out of the compression process is proven directly dependent on it. If P_{crit} is high, the fluid has a tendency to exit the compressor in high superheated state, while with low P_{crit} the fluid needs to enter the compressor in superheated state to avoid liquid in the compressor. Another important property is the critical temperature T_{crit} since the COP tends to increase with higher T_{crit}. The volumetric heating capacity VHC is important for the compressor design, since low VHC gives a high volumetric flow rate, which again requires larger compressor sizes. The value of VHC is strongly dependent on the normal boiling point T_{bp}. A high T_{bp} implies a low saturation pressure at a given temperature and therefore a low vapor density, which again gives a low VHC (Bertinat, 1988).

The first parameters used to exclude not-optional fluids are the Ozone Depletion Potential (ODP) and the Global Warming Potential (GWP), which are used to measure the threat a fluid oppose to the ozone layer and to global warming. After the decisions of phasing out refrigerants with too high GWP and ODP, many refrigerants with capable thermodynamic properties are or will in the future be unusable in heat pump systems. This includes especially CFC, HFCF and HFC fluids. The decision of phasing out CFC fluids due to high ODP values was an important part of the Montreal Protocol from 1989. HCFC fluids were suggested to replace CFC fluids, but the restriction of HCFC usage started in 1996 and the plan is to phase them out before 2030 (Chamoun et al., 2012). HFC fluids were the solution to the out phasing of HCFC fluids and were seen as a long-term solution with their low ODP values (Calm and Hourahan, 2007). The awareness of global warming became a huge political discussion in the late 1990s, and with the Kyoto Protocol in 1997 the GWP values of refrigerants became an important issue and restricted the usage of HFC fluids, although it is still highly in use. Since the global warming focus is just getting larger, more restrictions on HFC fluids are counted as a future certainty.

The natural fluids used in the first mechanical vapor compression cycles, like ammonia, carbon dioxide, hydro carbons and water, are nowadays considered as substitutes for fluorochemical compounds (Calm and Hourahan, 2007). Especially water (R718) and some hydrocarbons like pentane (R601) or isopenthane (R601a) are showing potential for applications at high temperature levels. The advantages with water as refrigerant are many, like the GWP and ODP values, it is not flammable, no disposal problems after use, it can work with direct heat exchangers and it has a high latent enthalpy (Karazhi et al., 2004). Water is especially showing potential in heat pumps at very high condensing temperatures over 150°C, where it is one of few available refrigerants that actually can be utilized due to its high critical temperature. Usage of water steam has shown to give high COP, but it is far from ideal due to its low VHC value and extreme superheating through the compressor, which increases compressor size (Bertinat, 1988). It is identified that water has approximately 200 times higher volumetric flow rate than traditional refrigerants for the same application at moderate temperature levels (Müller, 2001), but this factor will decrease for higher temperatures. Usage of hydrocarbons is also counted as not ideal, since they all are in some ways flammable, which require implementation of different security measures. Hydrocarbons like pentane and isopentane have incidentally the opposite problem as water with superheating, which means that they need to be superheated before entering the compressor.

4.1.2 Application

Industrial heat pumps are used in a large range of branches, and the most common ones are the petroleum industy, chemical industry, wood drying industry and the food and beverage industry. The most common application is dehumidification drying of wood, which usually are small CVC systems (McMullan, 2013).

4.2 Mechanical Vapor Recompression (MVR)

MVR systems are similar to the CVC system, but some differences still exist. The most obvious one is the fact that a MVR system is open, which means that it has a mass flow of refrigerant in and out of the system because it uses the exhausted process fluid as the refrigerant in the system. This exhausted process fluid is usually vapor entering the MVR system, which means that it directly goes through a compression process without needing the evaporator. The fluid then transfers heat while condensing and then leaves the system (Wolf et al., 2012). Figure 4.2 illustrates a typical MVR system.

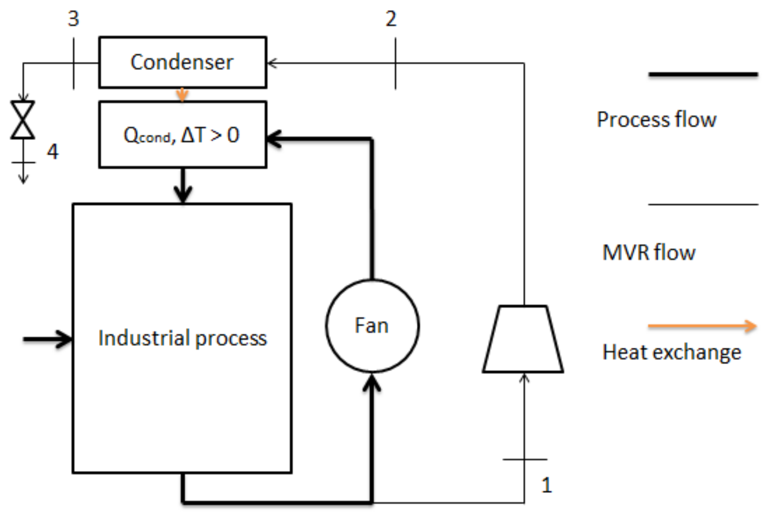

Figure 4.2: Simple mechanical vapor recompression system.

MVR systems are preferred when it is possible because of high performance compared to CVC systems and other solutions. It exists multiple reasons for this, but it is clear that losses in the extra components and mainly the extra heat exchanger in CVC systems have an impact (Bantle et al., 2015). In addition to this is the MVR system cheaper than a CVC system due to fewer components. The problem with a MVR system is that its application is seen as much more restricted than other systems (Gluckman, 1988, p. 64-65).

4.2.1 Refrigerants

The refrigerant in MVR systems is usually water, because it is the fluid that most often is available in the industry as exhausted process gas. Water steam between 70-100°C is compressed and usually condenses at temperatures around 20-30°C higher than before compression. It is also possible to achieve higher temperature levels, but the COP will decrease and it requires higher investment costs in the sense of technological modifications and extra components.

4.2.2 Application

The main application for MVR systems are vapor compression evaporation, distillation, evaporative crystallization and evaporative drying. The reason for that is the high energy consumption these processes are requiring, and a reduction in this consumption, which can be achieved by implementing a MVR system, can lower the operation costs substantially (Turbovap, n.d.). For vapor compression evaporation application, which is a process were heat is recovered by transferring the heat from the condensing process to the evaporation process, are MVR systems especially in highly use and are easy to implement (McMullan, 2013).

The temperature lifts for these different applications can vary a lot. For usage in evaporation plants the temperature lift is 10 to 20 kelvin, in distillation columns 30 to 60 kelvin, in drying processes 50 to 70 kelvin and in process integration 20 to 70 kelvin (Madsboell et al., 2015).

4.3 Compressor

The choice of compressor type and design in an industrial heat pump system is vital to obtain not just a low system investment cost, but also a low operation cost in form of an optimal COP for the system. The most common compressor types in industrial heat pumps are reciprocation compressors, screw compressors and turbo compressors. There are some other compression technologies available in industrial application, for instance centrifugal fans and roots air blowers, which are used for smaller pressure ratios.

The choice of compressor type is depended on different variables, including the size and temperature levels of the system. The competitive capacity ranges for different compressor types have been changing a lot over the years. The present opinion is that the range is decreasing from the higher to the lower speed machines, which means that reciprocation compressors normally handles the lowest capacities, the turbo compressors handle the highest capacities, while the screw compressor range is in between the other compressor types. Typical values are maximal 500 kW heat output in systems with reciprocation compressors and 5 MW for screw compressors, while turbo compressors are used in systems with heat output from 2 MW and higher (LeonardoEnergy, 2007). As screw compressors have been developed to handle higher capacities, the usage has penetrated the traditional capacity range of turbo compressors, making turbo compressors less competitive on medium-high flow rates (Soumerai, n.d.).

The physical properties of the working fluid, in particular the specific heat of condensation and the specific volume at the inlet of the compressor affects the size of the compressor, and is therefore important factors in choosing and designing the compressor in a heat pump system. Compressor efficiency is another important factor, and typical total efficiency values for different compressor types were in 1997 0.48 for reciprocation compressors, 0.55 for screw compressors and 0.64 for turbo compressors (Berntsson and Franck, 1997, p. 21). These values will differ some for different working fluids, but it is still an indication of typical compressor type performances.

4.3.1 Turbo compressor

It exists two different kinds of turbo compressors, the axial compressor and the centrifugal compressor. Axial compressors are preferred in applications like gas turbines and high speed ship engines, while centrifugal compressors are often used in automotive engines and in natural gas pipelines. The advantage of the axial compressor is a more compact design and high efficiency, while the centrifugal compressor is cheaper and can manage relatively higher pressure ratios per stage of compression (Madsboell et al., 2015). The centrifugal compressor is the most natural choice for heat pump applications, mostly due to the investment costs.

The main task for compressors is to increase the working fluids pressure. A centrifugal turbo compressor causes that to happen when the working fluid is entering the compressor axially, and its kinetic energy increases by the compressors rotating impeller. After leaving the impeller, the fluid passes through a diffuser, where the high speed flow is slowed down which leads to a pressure increase before it exits the compressor in radial direction. This pressure increase is dependent on the impeller peripheral speed, the fluid properties, the flow field geometry, the degree to which the compressor flow field have been optimal designed and the operation conditions of the compressor (Šarevski and Šarevski, 2014). An illustration of a typical centrifugal compressor design can be found in figure 4.3.

It is said that the most serious limitation of a turbo compressor is the lack of flexibility, in the sense that it cannot operate at a constant speed over a wide range of conditions. Moreover, screw compressors allow higher compression ratios and typically higher COP in CVC systems (Berntsson and Franck, 1997, p. 5-6). The result of this is that turbo compressors are for most applications just competitive at

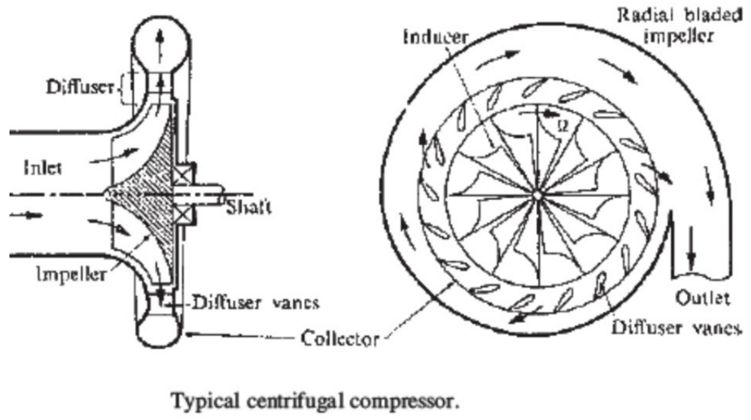

Figure 4.3: Typical centrifugal compressor design (Roy and Pradeep, n.d.)

very high capacities or in special cases where competing compressor types are not sufficient.

4.3.2 Centrifugal compressors in water based heat pumps

The centrifugal turbo compressor is often used in MVR systems and large CVC systems. The usage of turbo compressors in such systems can be explained by the use of water vapor as refrigerant, and its extremely low density compared to other refrigerants, which results in high volumetric flow rates. The pressure ratio in turbo compressors used in MVR systems is normally around 2, while the isentropic efficiencies varies from 0.7 to 0.8 (Berntsson and Franck, 1997, p. 26). The turbo compressor can produce higher pressure ratios with use of advance materials and multiple stages, like the customized turbo compressor offered by the Danish company DryingMate (n.d.), which can manage a temperature lift of 50 kelvin.

The newest commercial centrifugal compressors are able to handle extreme volumetric flow rates, but have a rather restricted temperature lift per stage. The Chinese enterprise Turbovap (n.d.) have compressors available with a maximal temperature increase of 20 kelvin per stage and can maximal manage 100 ton water per hour. The low temperature increase for each stage is a result of waters properties, and especially its high critical pressure, which makes it highly superheated after each compressor stage (Bertinat, 1988). The consequence of this is that the pressure ratio must be as high as possible, but is limited by waters low molecular mass, forcing an extreme impeller peripheral velocity. The material restricts this velocity to maximal

500 to 600 m/s (Šarevski and Šarevski, 2012a).

Normal commercial centrifugal compressors have problems with the rotational speed requirements in applications with usage of water as working fluid. A novel compressor with a special planetary gear from an automobile supercharger is newly developed, giving the centrifugal compressor a rotational speed range from 100 000 to 300 000 rpm. The cost of manufacturing a traditional gear for centrifugal compressors to operate more than 50 000 rpm is huge. The planetary gear can therefore be an extremely cost efficient component in turbo compressor used with water vapour (Madsboell et al., 2015). This is a promising development of the technology, considering that the research paper from Lachner et al. (2007) concluded that heat pump systems using water as refrigerant needed a breakthrough in compressor technology due to the high compressor costs.

This development is now patented and commercialized by the Danish company Rotrex A/S, which delivers relatively cheap turbochargers. Their largest turbochargers can handle a rotational speed of 90 000 rpm, whereas the smaller ones handles rotational speeds up to 220 000 rpm, and are used in the automobile industry (RotrexA/S, n.d.). However, a cooperation project between Rotrex A/S and another Danish company, Weel & Sandvig, investigates the possibility for utilization of such compressor technology for water based heat pumps at temperatures between 70 and 150 °C. The concept is to connect compressors together both in parallel and series in the heat pump system, such that flexibility in volumetric capacity and temperature lift are achieved (Weel et al., 2012).

An advantageous attribute with centrifugal compressors in industrial heat pumps, is the possibility for usage without oil lubrication. Among several possibilities are magnetic bearings and dry gas seal bearings the most common technology in oil free compressors (Stahley, 2005). This technology leads to not only less maintenance costs, but will also in heat pump applications lead to no losses due to oil contamination on heat transfer surfaces. Moreover, avoidance of oil circulation in heat pumps is especially vital in applications where products are vulnerable for small leakages in heat exchangers, such as in the food industry. The planetary gear used in the novel compressors from Rotrex A/S is however in need of oil lubrication, but non of this oil is in contact with the working fluid. Although no leakage of oil to the compressed water vapour were detected when this compressor technology were demonstrated (Weel et al., 2012), are there no assurances of this over a long period of time, and can therefore be problematic in applications with fragile products.

For centrifugal compressor using water as refrigerant is multi-stage compression often required due to high pressure ratio and high volumetric flow rate (Lachner et al., 2007). To split the compression in stages means in this case simply to connect two or more separate compressors. The COP of a two-stage R718 centrifgual compressor system with an intercooler-economizer between the stages is proven to be constant at around 9 % higher than a single stage for different temperature lifts (Šarevski and Šarevski, 2012b). For systems that requires lifts more than 45 kelvin is a three-stage compression process needed, whereas the maximum three-stage temperature lift is approximately 75 kelvin (Madsboell et al., 2015). The compressor stages are usually limited to three, and as illustrated in figure 4.4 by the trends for flashed intercooled and indirect intercooled compressors, the increase in COP by expanding the system from 3 to 4 stages is usually not high enough to be financial advisable.

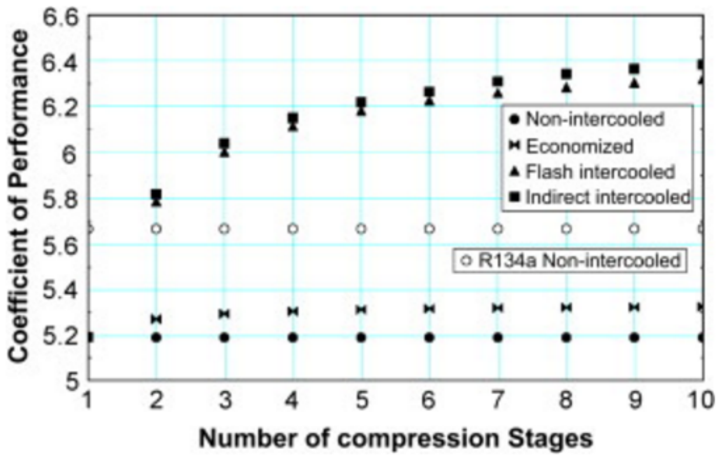

Figure 4.4: COP for different number of stages for a variety of intercooling methods (Lachner et al., 2007).

In figure 4.4 are different methods to cool the superheated water steam between compression stages mentioned. The most efficient methods are flash intercooling, which is a method of direct intercooling, and indirect intercooling. These two methods are illustrated in figure 4.5. Flash intercooling is done by cooling the working fluid in a flash tank, also called pressure vessel, between the compression stages, whereas indirect intercooling is done by cooling the working fluid with an external fluid in a heat exchanger. Flash intercooling and indirect intercooling have, as figure 4.4 shows, approximately the same efficiency, but flash intercooling is considered a cheaper option and is therefore preferred in most cases. However, it is not possible to implement flash intercooling for all applications, and indirect intercooling is rather applied in such cases (LeonardoEnergy, 2007).

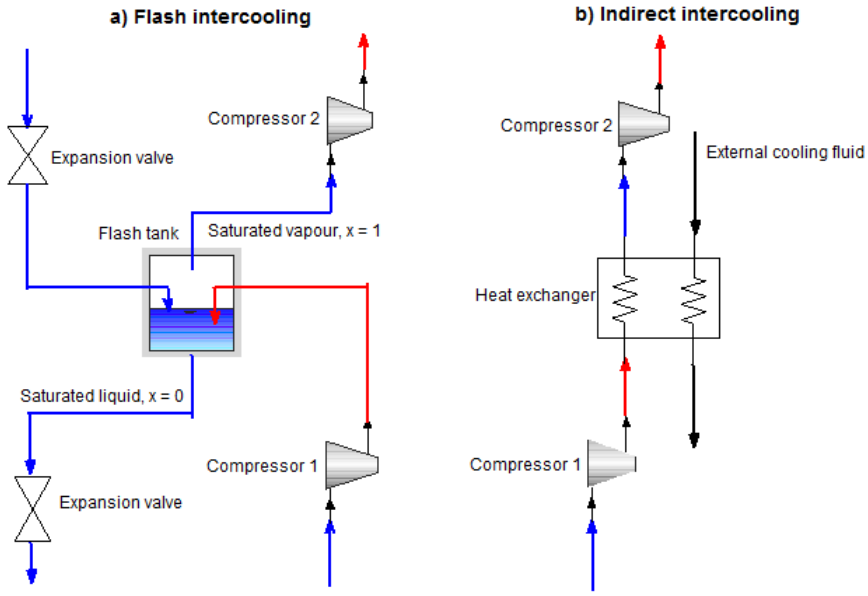

Figure 4.5: Different intercooling methods.

5 Case study: UHT Plant in Tine Aalesund

Tine Aalesund is a dairy factory in Norway, which operates several heat treatment plants including four UHT plants and several pasteurization plants. The pasteurization plants are extremely energy efficient, with around 90% regeneration, but this is not the case for the UHT plants. One of these UHT plants is a model called Thermen, which is emphasised in this case study. A flowchart and additional information of the plant is attached in appendix A, and a simple illustration can be found in figure 5.1.

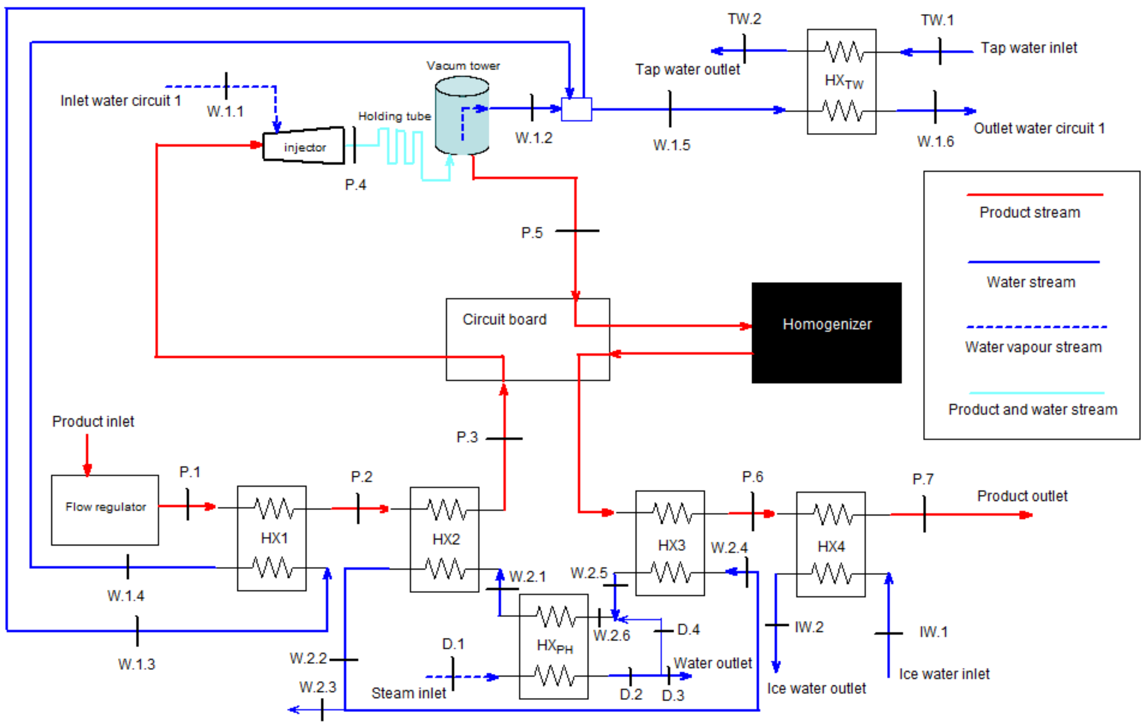

Figure 5.1: Simple flowchart of Thermen.

Thermen is a typical UHT plant with direct heating, in addition to external preheating and cooling of products. The plant runs at a variety of operating conditions dependent on the product specifications. The reason for this is that some products have higher viscosity than others, which means that these viscous products leave the system at high temperatures to secure the correct flow rate. Hence, these products only requires a small amount of cooling after the UHT treatment. On the other hand are less viscous products in need of a large amount of cooling after the UHT treatment. Moreover, these less viscous products have a larger customer demand,

and are produced more frequently than the viscous products. For this reason is only the production of whole milk, which is one of these less viscous products, further analysed in this thesis. Whole milk contains 88.7 % water, 3.5 % fat, 4.5 % carbohydrates and 3.3 % protein.

6240 kg/hour of products enter the system in P.1 at 4 °C and are pumped to the first heat exchanger for preheating, named HX1. However, this heat exchanger is not utilized for products with high cooling demand. The outlet temperature at P.2 is therefore still 4 °C. The products are then entering a second heat exchanger for preheating, HX2, and are heated to 75°C in P.3. The products are now pumped to the direct heating system consisting of an injector, a holding tube and a vacuum tank. 784 kg/hour water steam at 6 bar is injected into the products. The products are almost instantly heated to 143°C in P.4 while the water steam is condensing. The products are kept at a temperature of 143 °C and a pressure of 4.5 bar for 7-9 seconds in the holding tube, before it is expanded in the vacuum tank at 0.7 bar. The expansion leads to evaporation of exactly 784 kg/hour of water steam, and an almost instantly cooling of the products to 76°C in P.5. The water evaporating temperature at 0.7 bar is 90°C, which means that some additional cooling must be done in the vacuum tank. There is no information available of the vacuum tank content, and the additional cooling is therefore assumed as free cooling.

The products are then entering the homogenizer, and exit at approximately the same temperature. The rest of the product cooling is done with two heat exchangers. The first heat exchanger for cooling, named HX3, cools the products down to 23 °C in P.6. For later calculations is this temperature corrected to 24.8 °C to withstand a criteria of 5 kelvin temperature difference in heat exchangers. The second heat exchanger for cooling, named HX4, cools the products down to the final temperature of 8 °C before exiting the plant.

The four different heat exchangers for product heating before direct heating (HX1 and HX2) and cooling after direct heating (HX3 and HX4) are all using water as heating or cooling fluid. HX1, which is not utilized, is available to use parts of the condensate from the exhausted water vapour exiting the vacuum tower as heating fluid. This is shown in figure 5.1 by point W.1.3 and W.1.4. HX2 for preheating and HX3 for cooling are connected within the same water circuit, which means that these two heat exchangers are dependent on each other. However, the mass flow rate in HX2 is greater than in HX3. The water circuit is made for flexible operation, but this analysis assumes constant operation, such that 893 kg/hour water is drained in W.2.3 before entering HX3, whereas the same amount is added in D.4. 6790 kg/hour water is cooled in HX2 from 81.1 °C in W.2.1 to 19.8 °C in W.2.2.

5897 kg/hour of water enters HX3 at 19.8 °C in W.2.4 and is heated to 70.7 °C in W.2.5. To achieve the required mass flow rate and temperatures in HX2, is an additional heat exchanger HX_{PH} connected to the circuit, which acquires energy from steam production. Water is here heated from 70.7 °C in W.2.6 to 81.1 °C in W.2.1. Because HX3 is not sufficient for the whole cooling demand of the products, is ice water utilized in HX4. 6548 kg/hour of ice water enters HX4 at 2 °C in IW.1 and leaves at 17 °C in IW.2.

The factory utilizes two gas boilers of 8 and 4 tons and one electricity boiler of 3 tons to produce water vapour at 10 bar. The gas boilers have an efficiency of 86%, while the electric boiler has an efficiency of 92%. This steam is expanded to 6 bar and utilized both for the injector and for preheating of the products in the UHT plants. For Thermen is this steam utilization shown in figure 5.1 before the injector in W.1.1 and before heat exchanger HX_{PH} in D.1. The ice water production in the factory is done in a 80 000 liter tank, which is cooled by the evaporator of an ammonia refrigeration system. The water temperature entering the tank is dependent on the outdoor temperature, while the water exiting the tank is between 1°C and 2°C. Due to the varying temperature entering the tank is it sufficient for calculations to apply an approximate average yearly temperature of 6 °C. The temperatures, T_P and T_W, and mass flow rates, $\dot{m}_P$ and $\dot{m}_W$, of all known process streams are listed in table 5.1.

5.1 System 1: Existing Plant

The temperatures ΔT, mass flow rate $\dot{m}$, specific heat capacity Cp and heat loads Q of the process streams that are included in the energy analysis are listed in table 5.2. The streams are divided into cold streams (K) and hot streams (H). The streams included are:

- K1, which contains the preheating of the product stream from point P.1 to P.3.
- K2, which contains the direct heating of the product stream from point P.3 to P.4.
- K3, which contains the water side of heat exchanger HX3 from point W.2.4 to W.2.5.
- K4, which contains the water side of heat exchanger HX_{PH} from point W.2.6 to W.2.1
- H1, which contains the cooling of the product stream from point P.5 to P.7.

Table 5.1: Mass flow rates and temperatures at specific points in Thermen.

Products	$\dot{m}_P$ [kg/h]	T_P [°C]		Water	$\dot{m}_W$ [kg/h]	T_W [°C]
P.1	6240	4.0		W.1.1	784.4	159.0$^{(2)}$
P.2	6240	4.0		W.1.2	784.4	76.0
P.3	6240	75.0		W.1.3	0	-
P.4	7024.4	143.0		W.1.4	0	-
P.5	6240	76.0		W.1.5	784.4	76.0
P.6	6240	24.8$^{(1)}$		W.1.6	784.4	33.3
P.7	6240	8.0		W.2.1	6789.6	81.1$^{(3)}$
				W.2.2	6789.6	19.8$^{(3)}$
				W.2.3	892.8	19.8$^{(3)}$
				W.2.4	5896.8	19.8$^{(3)}$
				W.2.5	5896.8	70.7$^{(3)}$
				W.2.6	6789.6	70.7$^{(3)}$
				IW.1	6548.4	2.0
				IW.2	6548.4	17.0

(1) *This temperature is corrected from 23°C to 24.8°C to withstand a criteria of minimum 5 kelvin between fluids in heat exchangers.* (2) *Only the pressure of this stream is known, and saturated vapour is assumed to obtain this temperature.* (3) *These temperatures are collected from when the actual plant was running.*

- H2, which contains the exhausted condensate utilized for hot tap water production from point W.1.5 to W.1.6.
- H3, which contains the water side of heat exchanger HX2 from point W.2.1 to W.2.2.

It can be observed that both K1 and K2, and K3 and K4 actually are the same streams. However, these are divided into separate process streams to make calculations possible. K1 and K2 are divided because the direct heating is forced in the system and cannot be replaced by heat exchanger integration due to product quality reasons. K3 and K4 are divided because the mass flow rate of the streams are different.

The water vapour utilized in the direct heating DH and in HX_{PH} are categorized as hot utilities, while the ice water stream in HX4 and the tap water stream in HX_{TW} are categorized as cold utilities. Regarding the cold utilities is it important to notice that power is needed to produce the ice water stream, whereas hot tap water is produced in HX_{TW}, and is therefore counted as useful surplus heat from the system.

Table 5.2: Hot and cold process streams for system 1.

Hot/Cold	ΔT [°C]	$\dot{m}$ [kg/h]	Cp [kJ/kg*K]	Q [kW]
K1	$4 \rightarrow 75$	6240	3.936	484.36
K2	$75 \rightarrow 143$	6240	3.983	469.47
K3	$19.8 \rightarrow 70.7$	5896.8	4.186	349.02
K4	$70.7 \rightarrow 81.1$	6789.6	4.194	82.25
H1	$8 \leftarrow 76$	6240	-3.933	-463.49
H2	$33.3 \leftarrow 76$	784.4	-4.186	-38.94
H3	$19.8 \leftarrow 81.1$	6789.6	-4.190	-484.36

From the information in table 5.2 can a grid diagram of the existing HEN system be drawn, as done in figure 5.2. The temperatures of all streams are placed on each side of the respective stream or between heat exchangers, while the energy transferred in each heat exchanger is located next to the respective heat exchanger. The heat capacity flow rate $C_{H/K,i}$ of each stream is placed on the right hand side in the diagram.

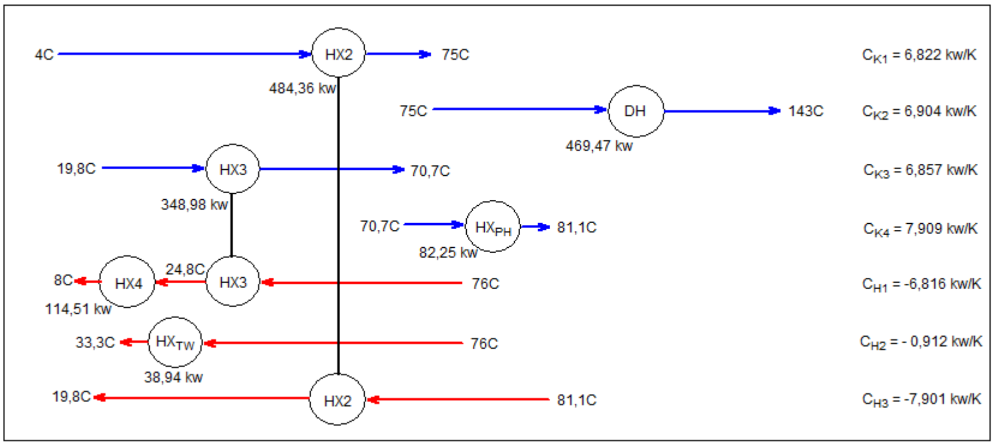

Figure 5.2: Grid diagram of the heat exchanger network in system 1.

Figure 5.2 shows that the existing plant uses 6 heat exchangers in the current operation, but has 7 heat exchangers available if HX1 is accounted for. This operation requires 530.54 kW of hot utilities, in the form of DH and HX_{PH}, and 147.56 kW in cold utilities, in the form of HX4 and HX_{TW}. This however does not reflect the heat and power consumption required by the UHT plant. The reason for this is the different attributes of the utilities. DH and HX_{PH} utilizes steam from boilers with efficiencies lower than unity, and the heat and power consumption will be distinctly higher than what is transferred in the heat exchange processes. HX4 utilizes ice water, which is produced by an energy efficient ammonia refrigeration system, and the power demand will be distinctly lower than what is transferred in the heat exchange process. Only the heat transferred in HX_{TW} is approximately identical to what is accounted for in the energy balance for the whole system.

5.2 System 2: Existing Plant with Modifications

From figure 5.1 can it be observed that there must occur a water vapour stream in the vacuum tank, which condenses trough the tank at 90°C and cools down to 76°C before it exits and is utilized afterwards for hot tap water production (stream H2 in system 1). In this system is it assumed that the water vapour stream can be utilized in a condensation process, which is called stream H4. It is also assumed that the latent heat from cooling of this stream from 90°C to 33.3°C can be utilized, which is this system is called H2. Thus, this is the same stream as H2 from system 1, only with greater temperature range. Whether it is possible to utilize the condensation process without affecting the product processing is uncertain, but the option is nevertheless interesting from an energy efficiency perspective. The water vapour, which occurs in the vacuum tank, is the same amount of water as the amount injected into the product in the direct heating process. The result of this is that stream H2 and the new condensing stream H4 are dependent on the hot utility in form of direct heating on stream K2, which means that this utility is forced in this system. The streams in this system and system 1 are otherwise identical. Table 5.3 includes the temperatures ΔT, mass flow rate $\dot{m}$, specific heat capacity Cp and heat load Q of the included process streams in this system.

Table 5.3: Hot and cold process streams for system 2.

Hot/Cold	ΔT [°C]	$\dot{m}$ [kg/h]	Cp [kJ/kg*K]	Q [kW]
K1	$4 \rightarrow 75$	6240	3.936	484.36
K2	$75 \rightarrow 143$	6240	3.983	469.47
K3	$19.8 \rightarrow 70.7$	5896.8	4.186	349.02
K4	$70.7 \rightarrow 81.1$	6789.6	4.194	82.25
H1	$8 \leftarrow 76$	6240	-3.933	-463.49
H2	$33.3 \leftarrow 90$	784.4	-4.186	-51.71
H3	$19.8 \leftarrow 81.1$	6789.6	-4.190	-484.36
H4	$90 \leftarrow 90$	784.4	-	-497.44

From table 5.3 can an unfinished grid diagram of system 2 be drawn, as shown in figure 5.3. A finished diagram of HEN as the one in figure 5.2 cannot be done with this system yet, since no calculations are done. However, the forced hot utility, DH, has to be operating in order to create stream H2 and H4, which can be included in the diagram. The temperatures of all streams are placed on each side of the respective stream, while the energy transferred in each heat exchanger is located next to the respective heat exchanger. The heat capacity flow rate $C_{H/K,i}$ of each stream is placed on the right hand side in the diagram.

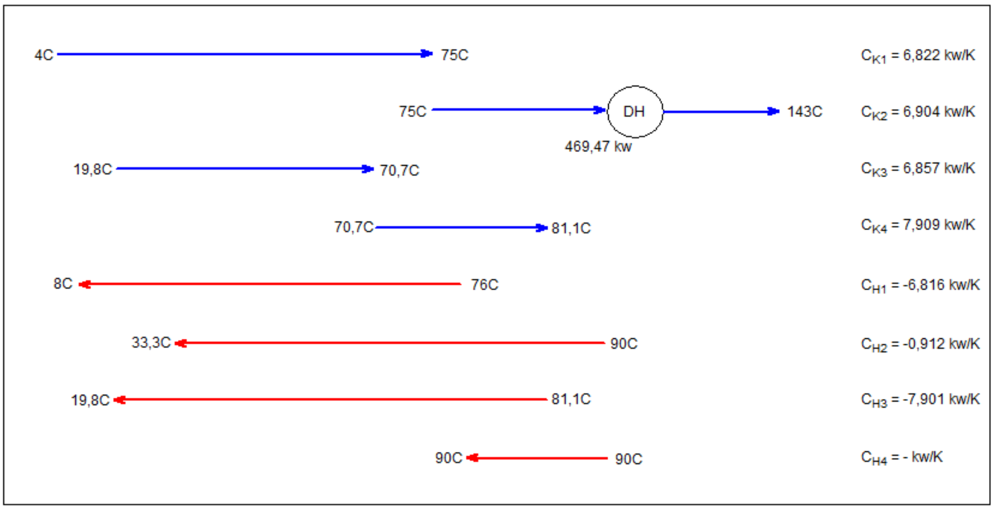

Figure 5.3: Simple illustration of the unfinished grid diagram of system 2.

5.3 System 3: Existing Plant with Modifications

In this system are the condensing stream H4 and the latent heat of H2 utilized, as for the previous system. However, a clear difference between this system and system 2 occurs because the entire water circuit W.2 from figure 5.1 is removed. This means that stream K3, K4 and H3 are not included in this stream. This is done to test the potential of doing preheating and cooling of products otherwise. The W.2 circuit is not practically removed, but rather operates as utilities. The argument for doing this is that the circuits heat demand is large, and reducing this heat load by integrating heat exchangers between other streams could mean massive energy savings. Whether these heat exchanger integrations are possible is still questionable, since product quality needs to be secured and space needs to be available, but the option is still interesting from an energy efficiency perspective. The process streams in this system are otherwise identical to system 2, and listed in table 5.4.

Table 5.4: Hot and cold process streams for system 3.

Hot/Cold	ΔT [°C]	$\dot{m}$ [kg/h]	Cp [kJ/kg*K]	Q [kW]
K1	$4 \rightarrow 75$	6240	3.936	484.36
K2	$75 \rightarrow 143$	6240	3.983	469.47
H1	$8 \leftarrow 76$	6240	-3.933	-463.49
H2	$33.3 \leftarrow 90$	784.4	-4.186	-51.71
H4	$90 \leftarrow 90$	784.4	-	-497.44

Since H4 and H2 from system 2 are also included in this system, is the forced direct heating DH also included in this system. From this information and table 5.4 can the unfinished grid diagram be drawn, as illustrated in figure 5.4. The temperatures of each stream is placed on each side of the stream, while the energy transferred in each heat exchanger is located next to the respective heat exchanger. The heat capacity flow rate $C_{H/K,i}$ of each stream is placed on the right hand side in the diagram.

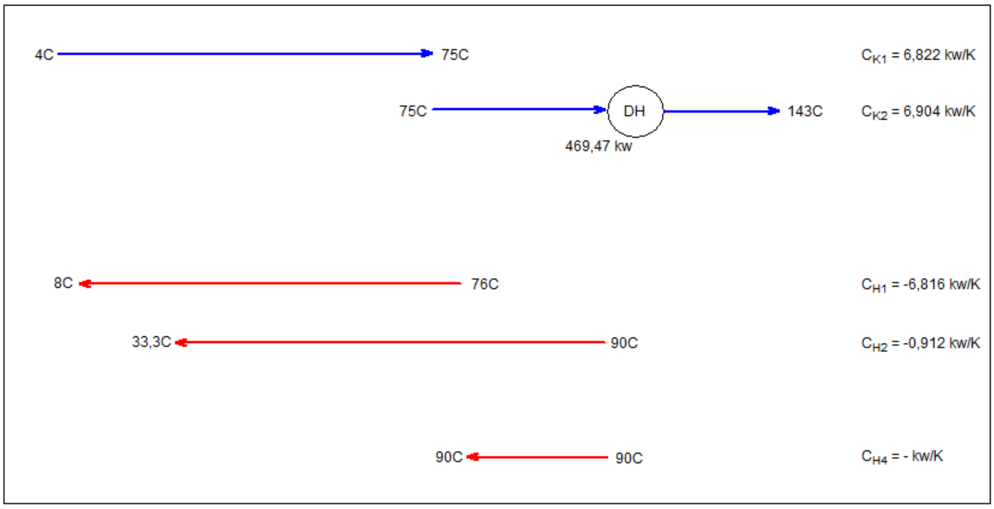

Figure 5.4: Simple illustration of the unfinished grid diagram of system 3.

5.4 System 4: System 2 with Heat Pump Implementation

System 4 is identical to system 2, but with one new modification. The point of this system is to look at the potential for utilization of the new condensing stream H4 from system 2 in a heat pump, such that the heat in this stream can be recycled on a higher temperature level. In the previous systems is it clear that most of the heat consumption is realised through steam production for the direct heating process. If this heat rather could be delivered by a heat pump with high net power factor, could this save a huge amount of energy.

The chosen heat pump solution for this application uses the steam from the vacuum tank directly as working fluid and compresses it up to 6 bar, which is the same pressure level as the steam utilized in the existing system for direct heating. The steam from the compression process is then utilized in the direct heating process instead of steam produced in steam boilers. The heat pump must therefore be able to cover the entire heat demand required in the direct heating process. The heat pump will continuously recycle water from the vacuum tank, and release it in the injector, which makes this heat pump a closed system. Because no investments in evaporator or condenser are required, and no heat exchanger losses will occur, is this solution considered the cheapest and most suitable option for this specific application.

To be able to implement this heat pump, there are some challenges that must be solved. There is absolutely no literature studies on heat pumps for this specific application, and its potential is therefore uncertain. There are several reasons for this. Firstly, the product quality must be maintained with this solution, which means that the water vapour must be sterile. To accomplish this must the heat pump be equipped with a filter system, oil free compressors and possibly more such equipment to secure no contamination of the products. This problem could be avoided by using a CVC solution. However, another challenge with the heat pump is the high required temperature lift of 70 kelvin, which would be even higher for a CVC system due to extra heat exchanger losses. Even a temperature lift of 70 kelvin creates difficulties for the chosen system, because the water vapour exiting the compression process is highly superheated. This forces a multistage compression process with intercooling between the stages to avoid extreme temperatures. Turbo compressors are commonly utilized in such arrangements with water as working fluid, and can be developed with oil-free bearings. A multistage turbo compressor solution is therefore chosen for the compression process in the heat pump.

Another requirement is the pinch point temperature. For the heat pump implementation to be rational, it must be placed around the pinch, since this would be a process-to-process heat pump. This means that the pinch point must be located at a temperature level above the condensation temperature of stream H4 at 90°C for the heat pump implementation to be advisable. The pinch, which here is spoken about, is the pinch point from a pinch analysis on system 2. This is because such a pinch analysis is done on the system with the original streams without any impact from the heat pump. To create a heat exchanger network for this system must a new pinch analysis be done with the heat pump system implemented.

The portfolio of streams for the new pinch analysis, which includes the heat pump implementation, are almost identical to the stream portfolio of system 2 in table 5.3. However, since the heat pump is a closed system, the stream H2 must be removed because no water liquid from the vacuum tower can be utilized in the heat exchanger network. Otherwise is the stream H4 named HP in this system, and changed according to how the heat pump approximately must function in this system. The mass flow rate of stream HP $\dot{m}_{HP}$ is unknown, but must be lower than the mass flow rate of stream H4, since superheat from the heat pump compression is utilized in the direct heating. The superheat temperature out of last compression step is unknown as well. The streams are shown in table 5.5.

Table 5.5: Hot and cold process streams for system 4.

Hot/Cold	ΔT [°C]	$\dot{m}$ [kg/h]	Cp [kJ/kg*K]	Q [kW]
K1	$4 \rightarrow 75$	6240	3.936	484.36
K2	$75 \rightarrow 143$	6240	3.983	469.47
K3	$19.8 \rightarrow 70.7$	5896.8	4.186	349.02
K4	$70.7 \rightarrow 81.1$	6789.6	4.194	82.25
H1	$8 \leftarrow 76$	6240	-3.933	-463.49
H3	$19.8 \leftarrow 81.1$	6789.6	-4.190	-484.36
HP	$143 \leftarrow ?$	>784.4	-	-469.47

Since the heat pump is implemented with stream HP as its heat source and stream K2 as its heat sink, this connection is forced in this system. All other streams are free, and this is shown in figure 5.5. The temperatures of all streams are placed on each side the respective stream or between heat exchangers, while the energy transferred in each heat exchanger is located next to the respective heat exchanger. Since the heat pump performance needs to be simulated, the power consumption of the heat pump (HP in figure 5.5) is unknown. The heat capacity flow rate $C_{H/K,i}$ of each stream is placed on the right hand side in the diagram.

The heat pump itself, is a MVR system with 3-stage compression, and can be observed in figure 5.6. The steam exiting the vacuum tank (1) is heated 5 kelvin to avoid liquid compression, before it enters the first compressor step (2) and exits it at the first intermediate pressure level (3). Since the stream exits the first compression step highly superheated is a heat exchanger placed between first and second compressor step for intercooling purposes. Here will the gas be cooled down to 5

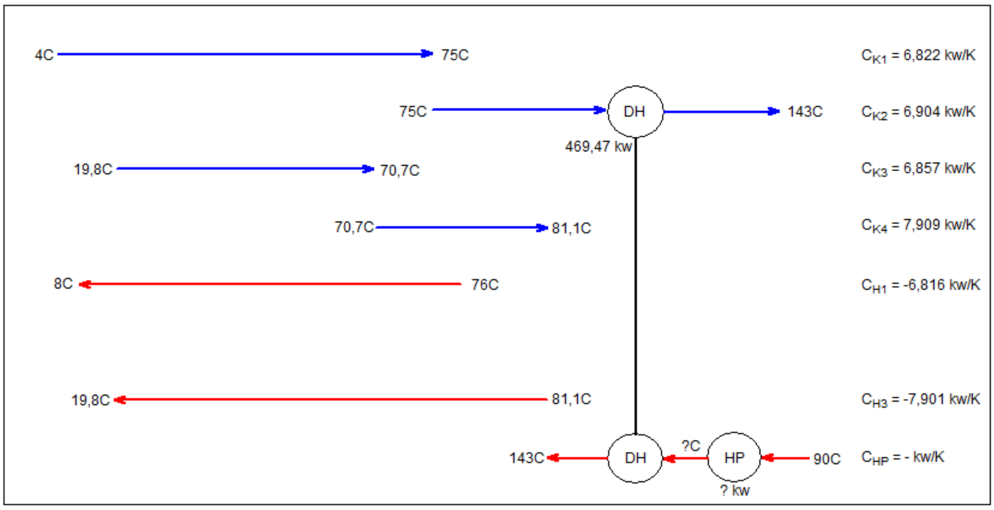

Figure 5.5: Simple illustration of the unfinished grid diagram of system 4.

kelvin above the dew point line (4) before it enters the second compression step. The stream reaches the second intermediate pressure level at the compressor exit (5) and enters the second intercooling heat exchanger highly superheated. It is also in this heat exchanger cooled down to 5 kelvin above the dew point line (6) before it enters the third compression step and is compressed up to the top pressure level (7). The steam is at this point superheated while it enters the injector and is then mixed with products while it cools down and condenses in the direct heating process. The mixed products and water liquid stays in the holding tube for approximately 8 seconds before it expands in the vacuum tank, and the water evaporates and can be utilized again in the heat pump.

The compressor steps are organized such that the first compressor step is three turbo compressors in parallel, the second step is two turbo compressors in parallel, whereas the third step is one single turbo compressor. Such an arrangement is typical for multistage compression with turbo compressors in water based systems with large heat capacity. The reason for this is the density difference of the water vapour entering the different compressor steps. In the first compressor step is low pressure steam entering, which implies a low density. This gives a large volumetric flow rate, and more turbo compressors in parallel are therefore required. The entering stream in compressor step 2 has a higher pressure, which gives a lower volumetric flow rate, and requires therefore only two parallel turbo compressors. The entering stream in the third compressor step has an even higher pressure, which gives an even lower volumetric flow rate, and only one turbo compressor is required.

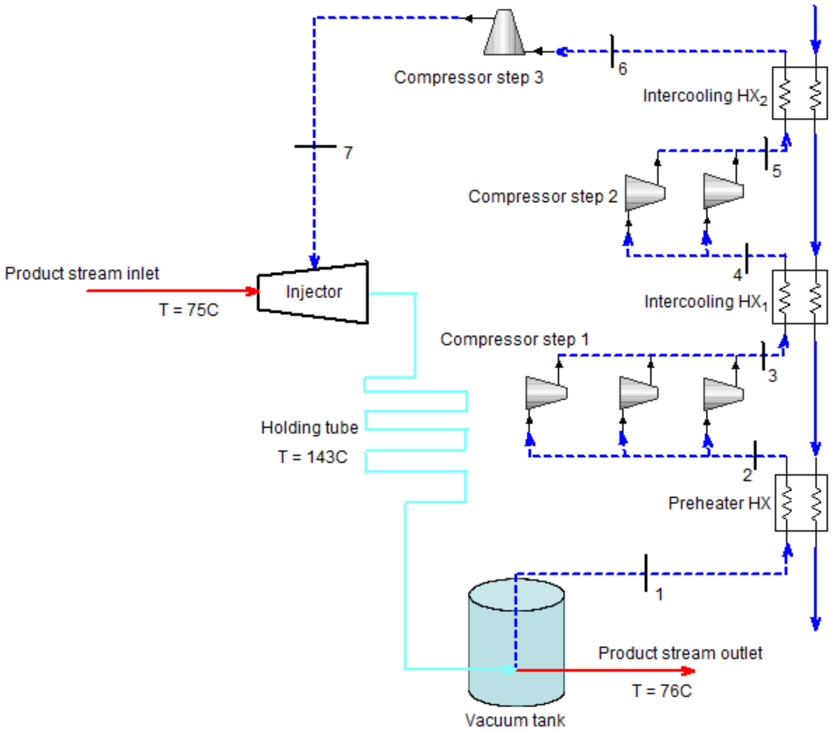

Figure 5.6: Illustration of the heat pump solution for the UHT plant.

Regarding the intercooling between the compression steps would usage of intermediate pressure vessels have been the ideal option, as the literature claims. However, this is not possible in this heat pump system. The reason for this is that the stream after the third compression step is utilized in the direct heating, such that there is no stream being expanded as there would in a normal heat pump system. Indirect heat exchangers are therefore conducting the intercooling. A water circuit is made to do the cooling in the heat exchangers. As the water is heated on the water circuit side of the intercooling heat exchangers, this circuit can also possibly be utilized in a heat exchanger for preheating of the vapour entering the first compressor step to avoid liquid compression.

5.5 System 5: System 3 with Heat Pump Implementation

System 5 is identical to system 3, but with one new modification. The point of this system is to look at the potential for utilization of the condensing stream H4 from system 3 in a heat pump, such that the heat in this stream can be recycled on a higher temperature level. The heat pump implementation in this system is identical to how it is done in system 4, which means that the same requirements are associated with this system. These requirements can be summarized as:

- The water must be sterile when it leaves the heat pump. A filter system, oil free compressors and possibly more such equipments are required.

- High temperature lift is required by the heat pump. A multistage turbo compressor solution with intercooling between the stages is chosen to cope with this.
- The heat source stream H4 must be located under the pinch.

Regarding the latter requirement is the pinch, which here is spoken about, the pinch point from a pinch analysis on system 3. This is because this pinch analysis is done on the system with the original streams without any impact from the heat pump. To create a heat exchanger network for this system must another pinch analysis be done with the heat pump system implemented.

The portfolio of streams for this pinch analysis is almost identical to the streams in table 5.4 from system 3. However, since the heat pump is a closed system, the stream H2 must be removed because no water liquid from the vacuum tank can be utilized in the heat exchanger network. Otherwise is stream H4 named HP in this system, and changed according to how the heat pump approximately must function in this system. The mass flow rate of stream HP is unknown, but must be lower than the mass flow rate of stream H4, since superheat from the heat pump compression can be utilized in the direct heating. The superheat temperature out of last compression step is unknown as well. The streams are shown in table 5.6.

Table 5.6: Hot and cold process streams for system 5.

Hot/Cold	ΔT [°C]	$\dot{m}$ [kg/h]	Cp [kJ/kg*K]	Q [kW]
K1	$4 \rightarrow 75$	6240	3.936	484.36
K2	$75 \rightarrow 143$	6240	3.983	469.47
H1	$8 \leftarrow 76$	6240	-3.933	-463.49
HP	$143 \leftarrow ?$	>784.4	-	-469.47

Since the heat pump is implemented with HP as its heat source and K2 as its heat sink, this connection is forced in this system. All other streams are free, and this is shown in figure 5.7. The temperatures of each stream is placed on each side of the stream heat exchangers, while the heat transferred in each heat exchanger is located next to the respective heat exchanger. Since the heat pump performance needs to be simulated, the power consumption of the heat pump (HP in figure 5.7) is unknown. The heat capacity flow rate $C_{H/K,i}$ of each stream is placed on the right hand side in the diagram.

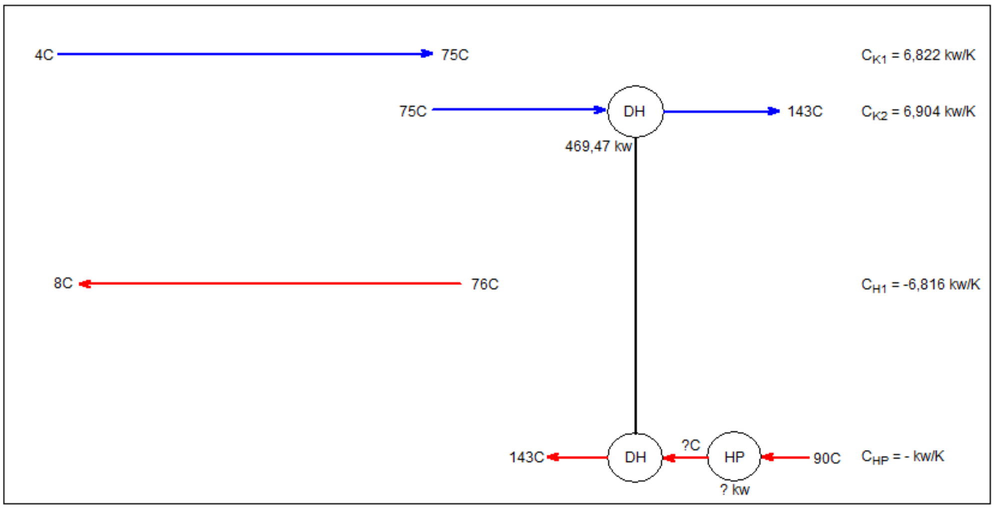

Figure 5.7: Simple illustration of the unfinished grid diagram of system 5.

The heat pump itself is in this system identical to the heat pump from system 4, since it serves the same purpose and the same heat demand is required. Figure 5.6 from chapter 5.4 yields therefore also to this heat pump solution.

6 Simulation Models

In order to compare the system performances are pinch analysis carried out. Through these analysis are minimal required hot and cold utilities, in addition to potential heat exchanger networks identified. Different limitations in possibilities for heat exchanging between streams are taken into account, such that new systems with modifications can be created. By creating these systems are the actual hot and cold utilities for each system identified, and the heat/power consumption and usable surplus heat are simulated. For systems with a heat pump solution is a heat pump model carried out. Calculation of the five systems are computed in Engineering Equation Solver (EES). These codes are attached in Appendix C.

6.1 Pinch Analysis Model

The following assumptions were made to achieve the thermodynamic parameters from the pinch analysis:

- No heat loss to the surroundings.
- No pressure loss in pipes.
- All specific heat capacities Cp of process streams are assumed as average between the inlet and outlet states of the respective stream.
- All heat exchangers are countercurrent and have a thermal efficiency of 100%.
- A minimum temperature difference ΔT_{min} of 5 kelvin is used for all systems.

Table 6.1 lists the input and output variables of the pinch analysis model.

Table 6.1: Inputs and outputs of the pinch analysis.

Inputs	Outputs
Number of streams	Temperatures of pinch point
Mass flow rates	Minimum hot utilities
Inlet and outlet temperatures	Minimum cold utilities
Specific heat capacities	Possible heat integrations of streams
Minimum temperature difference	MER system
Additional criterion	"Common sense" HEN system

6.1.1 Thermodynamic state properties

Each liquid streams specific heat load is calculated by temperature T [°C] of two points and the average specific heat capacity Cp [kJ/kg*K] of the stream between the two points. The Cp values are for milk calculated by the knowledge of the mass fraction of fat y_{fat}, protein $y_{protein}$, carbohydrates y_{carbs}, water y_{water} and the Cp values of these substances at specific temperatures. The calculation of Cp_{milk} can be found in equation 6.1.

$$Cp_{milk} = y_{fat} * Cp_{fat} + y_{protein} * Cp_{protein} + y_{carbs} * Cp_{carbs} + y_{water} * Cp_{water} \tag{6.1}$$

The Cp of all the substances except from water are calculated with equations 6.2, 6.3 and 6.4, which are retrieved from Choi and Okos (1986). Cp_{water} is retrieved from either a thermodynamic table for saturated water liquid (EngineeringToolbox, n.d.) or from a thermodynamic table of water at atmospheric pressure (ThermExcel, 2003).

$$Cp_{fat} = 1.9842 + 1.4733 * 10^{-3} * T - 4.8008 * 10^{-6} * T^2 \tag{6.2}$$

$$Cp_{protein} = 2.0082 + 1.2089 * 10^{-3} * T - 1.3129 * 10^{-6} * T^2 \tag{6.3}$$

$$Cp_{carbs} = 1.5488 + 1.9625 * 10^{-3} * T - 5.9399 * 10^{-6} * T^2 \tag{6.4}$$

For evaporating or condensing water streams are each point calculated by knowing the value of two thermodynamic properties. These properties can be pressure P [bar], temperature T [°C], enthalpy h [kJ/kg*K] or vapour quality x [-]. The unknown properties are retrieved by the thermophysical property functions in EES.

6.1.2 Temperature interval method

To retrieve the pinch point temperature and the minimum hot and cold utilities are the temperature interval method utilized. The input and output of the method are listed in table 6.2.

The first aspect of this method is to locate the hot and cold streams in the system and their mass flow rate $\dot{m}$, specific heat capacity Cp and temperature range T. This is done by taking the existing plant and look deeper at the specific processes. In this analysis are all process streams, which are essential for heating or cooling

Table 6.2: Inputs and outputs of the temperature interval method.

Inputs	Outputs
Number of streams	Pinch point temperature
Mass flow rates	Minimum hot utilities
Inlet and outlet temperatures	Minimum cold utilities
Specific heat capacities	
Minimum temperature difference	

processes, included. These are in the specific plant relatively easy identified. However, there are some streams, which can be characterized as both process streams and utilities. Here are simple logic taken into account. Whether these streams are characterized as process streams or utilities will nevertheless not influence the total energy balance. Each streams Cp are retrieved as mentioned in chapter 5.1.1. Each streams T are identified from either product specifications in Appendix A, measured when the plant was running or retrieved from employees at the factory. Each streams $\dot{m}$ are identified either from product specifications in Appendix A or calculated by an energy balance in heat exchangers.

With the knowledge of the specific stream inlet and outlet temperatures and the minimum temperature difference ΔT_{min} allowed in the heat exchangers, can temperature intervals be made. The inlet and outlet temperature of each stream are corrected by ΔT_{min} to avoid too low temperature difference in heat exchangers. This process are different for hot and cold streams, and the corrections can be found in equation 6.5 and 6.6. Here is $T_{C/H}{}^*$ the pinch corrected temperature, while $T_{C/H}$ is the real temperature. The first temperature interval will be located between the highest corrected inlet or outlet temperature in the system and the second highest corrected inlet or outlet temperature in the system. The intervals will follow this trend down to the last temperature interval, which will be located between the second lowest corrected inlet or outlet temperature in the system and the lowest corrected inlet or outlet temperature.

$$T_C^* = T_C + \frac{\Delta T_{min}}{2} \tag{6.5}$$

$$T_H^* = T_H - \frac{\Delta T_{min}}{2} \tag{6.6}$$

The next step of the temperature interval method is to locate the sum of heat capacity flow rate of each temperature interval $C_{I,sum}$. The heat capacity flow rate of the specific stream C is the product of the stream mass flow rate $\dot{m}$ and the specific heat capacity Cp, as illustrated in equation 6.7. On a side note is it important to know that the heat capacity flow rate is written with positive sign for cold streams and with negative sign for hot streams. If the heat capacity flow rate of each stream is known and the knowledge of which streams that are located in the different temperature intervals is obtained, can the sum of heat capacity flow rate in each temperature interval be calculated from equation 6.8. Here is I a specific interval, N represents the number of cold streams in interval I, M represents the number of hot streams in interval I, i represents the specific cold streams that occur in interval I, j represents the specific hot streams that occur in interval I, while R represents the amount of intervals.

$$C = Cp * \dot{m} \tag{6.7}$$

$$C_{I,sum} = \sum_{i=1}^{N} C_{i,C} + \sum_{j=1}^{M} C_{j,H}, \qquad I = 1,2,3...R \tag{6.8}$$

With the information of the heat capacity flow in each interval in addition to the corrected temperatures, which defines the different intervals, can the surplus or deficit of heat in each interval Q_I be calculated. This is the product of the heat capacity flow $C_{I,sum}$ and the temperature difference in the specific interval, as illustrated in equation 6.9. From the surplus or deficit of heat in each interval, is it possible to calculate the cumulative surplus or deficit of energy flow from the first interval to interval I ΔQ_I, which is shown in equation 6.10. ΔQ_1 is equal Q_1 in the first interval.

$$Q_I = C_{I,sum} * (T^*_{I,inlet} - T^*_{I,outlet}), \qquad I = 1,2,3...R \tag{6.9}$$

$$\Delta Q_I = \Delta Q_{I-1} + Q_I, \qquad I = 1,2,3...R \tag{6.10}$$

The pinch point can be observed from the values of ΔQ. The specific interval I with the greatest value of ΔQ defines the pinch point. This pinch temperature T^*_{pinch} is then the lowest temperature of interval I. Since the temperatures of the cold and hot streams are corrected by ΔT_{min} must the temperatures be corrected back. This means that the the pinch point temperature of cold streams is defined by equation 6.11. and the pinch point temperature of hot streams is defined by equation 6.12.

$$T_{pinch,C} = T^*_{pinch} - \frac{\Delta T_{min}}{2} \tag{6.11}$$

$$T_{pinch,H} = T^*_{pinch} + \frac{\Delta T_{min}}{2} \tag{6.12}$$

Not only the pinch is defined from the ΔQ with the highest value. Since this in fact must be the point of the system which the highest deficit of heat occurs, is this value defining the minimum hot utilities Q_{Hmin} required in the system. This value combined with the surplus or deficit of energy flow in each interval Q_I can be utilized to identify the minimum cold utilities required by the system. A new variable ΔQ^*_I is made to identify this value. This variable starts at ΔQ^*_0 equal the minimum hot utilities Q_{Hmin}, as shown in equation 6.13 and the calculation is further done similar to the calculations of ΔQ_I. The calculation of the rest of ΔQ^*_I is illustrated in equation 6.14. The ΔQ^*_R in the last interval defines the minimum cold utilities Q_{Cmin}, as illustrated in equation 6.15. Moreover, the ΔQ^* at the pinch temperature T^*_{pinch} is always zero.

$$\Delta Q^*_0 = Q_{Hmin} \tag{6.13}$$

$$\Delta Q^*_I = \Delta Q^*_{I-1} - Q_I, \qquad I = 1,2,3...R \tag{6.14}$$

$$\Delta Q^*_R = Q_{Cmin} \tag{6.15}$$

The results from the temperature interval method can be used to create the hot and cold composite curves. The composite curves are simply a plot of the temperature of hot and cold streams dependent on the heat load $Q_{H/C}$. As the temperature interval method, is these curves also made by creating temperature intervals. However, the methods differ because these curves are divided into hot and cold curves and the temperatures are not corrected by ΔT_{min}. By setting $Q_{0,C}$ equal to Q_{Cmin} at the lowest cold stream temperature and $Q_{0,H}$ equal to zero on the lowest hot stream temperature, can the intervals of the composite curves be created by equation 6.16 for cold streams and equation 6.17 for hot streams. Here is G the amount of temperature intervals for the cold streams and F the amount of temperature intervals for the hot streams. After creating these curves is it evident that the pinch point is located where the curves are closest to each other. This distance is exactly ΔT_{min} kelvin. Q_{Hmin} is also located in the plot, and is the horizontal distance between where the cold curve ends and the hot curve ends.

$$Q_{I,C} = Q_{I-1,C} + (\sum_{i=1}^{N} C_{i,C}) * (T_{I,C} - T_{I-1,C}), \qquad I = 1,2,3...G \tag{6.16}$$

$$Q_{I,H} = Q_{I-1,H} + (\sum_{i=1}^{M} C_{j,H}) * (T_{I,H} - T_{I-1,H}), \qquad I = 1,2,3...F \tag{6.17}$$

6.1.3 Heat exchanger network design

With the knowledge of where the pinch point is located, is it possible to create HEN systems. The inputs and outputs of such a design is listed in table 6.3.

Table 6.3: Inputs and outputs of heat exchanger design process.

Inputs	Outputs
Minimum temperature difference ΔT_{min}	MER system
Temperatures of pinch point	"Common sense" HEN system
Minimum hot utilities	
Minimum cold utilities	
Possible heat integration of stream	
Additional criterion	

There is no blueprint of HEN design appearances. However, to obtain a system of maximum energy recovery or minimum energy requirement (MER), there are guidelines to follow. First of all, the design of the systems must be divided at the pinch point into two regions that are designed separately. The heat exchange connections must first be designed close to the pinch and then gradually design connections further away from the pinch point. If this is not done, there will be no assurance that the streams are brought to the pinch point by the maximum amount of internal heat exchange. For heat exchanger connections between one cold stream with C_C and one hot stream with C_H, which leads the streams to or from the pinch point, can there be difficulties with the criteria of ΔT_{min}. If the constraint in equation 6.18 below the pinch or equation 6.19 above the pinch is not held within a heat exchanger, will the criteria of ΔT_{min} always be broken. However, in this analysis are connections, which break these constraints, made if $C_H \approx C_C$, because the temperature difference in the specific heat exchanger $\Delta T \approx \Delta T_{min}$.

$$C_H \geqslant C_C \tag{6.18}$$

$$C_H \leqslant C_C \tag{6.19}$$

By following the guidelines above can a MER system be created. However, a MER system is rarely the most sustainable design, and additional criterion in table 6.3 are therefore included. The MER system is in this analysis not considered the best design when:

- The MER system utilizes several heat exchangers that only recovers a small amount of heat. In such cases are other designs considered more suitable because of the investment costs.
- Parts of the forced hot utility for direct heating is located below the pinch. A MER system will always replace this part with heat exchanger integration. This is not allowed in the analysis, and other designs are rather considered
- Water streams, which only are included in the system for product heating or cooling purposes, are utilized in the MER system for other purposes. This is considered as an irrational design, and other designs are rather considered.

When some of the above incidents occur is rather a "common sense" HEN system chosen. These systems utilizes more external utilities than the MER system, and will usually obtain higher operation costs. However, they will serve the purpose of obtaining the required product quality and they will usually require less total costs than a MER system.

6.2 Heat pump Model

The heat pump can be modelled like most other closed vapour compression heat pumps. The difference is that no heat exchanger losses will occur from a condenser or an evaporator since evaporation and condensing are done directly. Water is otherwise used as refrigerant. The following assumptions were made to achieve the thermodynamic parameters in the heat pump model:

- No heat loss to the surroundings.
- No pressure loss in heat exchangers.
- All heat exchangers are countercurrent and have a thermal efficiency of 100%.
- Compressor isentropic efficiency is assumed a variable depending only on pressure ratio.

- Simple calculations of pressure losses in compressor steps in the system are taken into account.

Table 6.4 lists the input and output variables of the heat pump model.

Table 6.4: Inputs and outputs of the heat pump model.

Inputs	Outputs
Evaporation temperature	Intermediate pressure levels
Condensing temperature	Mass flow rate
Refrigerant	Thermodynamic state properties
Heat load	Compressor power consumption

6.2.1 Thermodynamic state properties

All different states in the system require two thermodynamic properties and the refrigerant to be known to calculate the other state properties. The different thermodynamic properties evaluated in the model are pressure P [kPa], temperature T [°C], enthalpy h [kJ/kg*K], density ρ [kg/m^3], and the vapour quality x [-]. The unknown properties are retrieved by the thermophysical property functions in EES.

6.2.2 Compressor

The performance of the turbo compressor steps i calculated from the available constant isentropic efficiency compressor component in the component library in EES. The input and output variables of this component can be obtained from table 6.5.

Table 6.5: Inputs and outputs of the compressor model.

Inputs	Outputs
Refrigerant	Outlet specific enthalpy
Inlet specific enthalpy	Power consumption
Inlet pressure	
Outlet pressure	
Mass flow rate	
Isentropic efficiency	

The high and low pressure levels, respectively the pressure in the vacuum tank and the pressure level of water entering the injector, are known. With this knowledge can the two intermediate pressure levels be calculated through equation 6.20. This means that all these pressure levels can be utilized as input variables in the compressor model.

$$P_{intermediate} = \sqrt{P_{high} * P_{low}} \tag{6.20}$$

All compressor steps are modelled with some pressure loss. Because the dimensions of the heat pump are not known, is this loss calculation simplified. Each compressor step is modelled with a pressure loss equal to one kelvin temperature loss in the two phase region with constant enthalpy. This is done in both the compressor suction pipe and discharge pipe. This calculation is done with equation 6.21, 6.22 and 6.23. $T_{corr,loss}$ is the temperature used for correcting the pressure level, T_{new} is the temperature after loss calculations, h_{actual} is the constant specific enthalpy, x is the vapour quality, whereas P_{old} and P_{new} are respectively the pressure before and after loss calculations

$$T_{corr,loss} = f(x = 1, P = P_{old}) - 1 \tag{6.21}$$

$$P_{new} = f(x = 1, T = T_{corr,loss}) \tag{6.22}$$

$$T_{new} = f(P = P_{new}, h = h_{actual}) \tag{6.23}$$

By the knowledge of the high, low and intermediate pressure levels, in addition to the pressure losses in the system, can the pressure ratios π in each compressor step be calculated. With this information can an isentropic efficiency η_{is} be roughly estimated. This is done with the performance graph of existing equipment from the company Rotrex, which can be illustrated in figure 6.1. The graph includes two different set of curves, where η_{is} represents number next to ascending curves and the descending curves are speed lines. The mass flow rate and speed lines in this figure are only valid for water vapour at a pressure of 0.5 bar and a temperature of 82 °C. However, it is assumed that the compressors for this application are customized, such that the operation of all compressors are optimized for their respective pressure ratio. This means that η_{is} is assumed independent of mass flow rate. The speed lines are not included in the analysis.

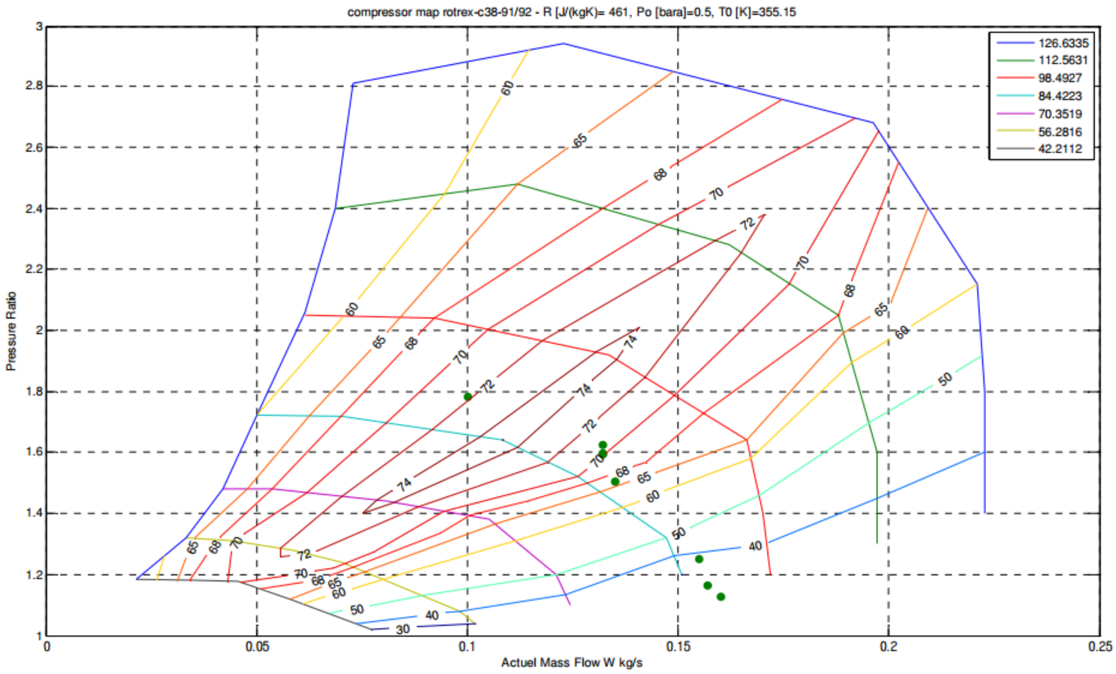

Figure 6.1: Performance graph of the turbocharger model Rotrex C38-91/92 (Weel et al., 2012).

With the compressor performances known, is the outlet specific enthalpy $h_{inj,inlet}$ of the last compressor step known. Because the heat load of the direct heating process Q_{demand} and the specific enthalpy of the water after direct heating $h_{inj,outlet}$ are known as well, can the mass flow rate of the heat pump $\dot{m}_{HP}$ be calculated by applying equation 6.24.

$$\dot{m}_{HP} = \frac{Q_{demand}}{h_{inj,inlet} - h_{inj,outlet}} \tag{6.24}$$

6.2.3 Heat exchangers

Both direct heat exchange and indirect heat exchange are included in the heat pump model, and can be modelled with equation 6.25, where i represent the two heat exchanging fluids. Direct heat exchange occurs in both the injector and the vacuum tank, whereas indirect heat exchange occurs in the heat pump preheating and intercooling processes. The purpose of the indirect heat exchangers are to either cool or heat the water vapour to a state of 5 kelvin superheat before the compression processes. This must be done by external heating or cooling. The model assumes in this case that cooling of the working fluid can be done with free cooling, such that no external utilities are accounted for in these processes. It also assumes that the fluid used for free cooling is heated sufficiently such that it also can be utilized for preheating of working fluid before the first compression step.

$$\sum_{i=1}^{2} \dot{m}_{i,inlet} * h_{i,inlet} - \sum_{i=1}^{2} \dot{m}_{i,outlet} * h_{i,outlet} = 0 \tag{6.25}$$

6.3 Heat and Power Consumption Model

There are three different utilities included in the analysis. These are:

- Steam production
- Ice water production
- Hot tap water production

The power consumption of pumps, sensor systems and other such required equipment is thereby not included. Steam and ice water are produced from respectively steam boilers and an ammonia refrigeration system, whereas hot tap water production acquires surplus heat from the UHT-plant. Since heat and power input in the UHT-plant are defined with positive sign, are steam production and ice water production positive, while hot water production is negative in this perspective.

The steam production is carried out by two gas boilers and one electric boiler. The gas boiler has an efficiency η_{el} and the electric boiler has an efficiency η_{gas}. The heat/power consumption of these two boilers over a year, Y_{el} and Y_{gas}, is known, such that an overall efficiency of the steam production η_{steam} can be calculated, which is done with equation 6.26.

$$\eta_{steam} = \frac{Y_{el} * \eta_{el} + Y_{gas} * \eta_{gas}}{Y_{el} + Y_{gas}} \tag{6.26}$$

For the calculation of the combined heat and power consumption for steam production is there done a simplification, by assuming that the heat released in the direct heating process and in heat exchanger HX_{PH}, Q_{DH} and Q_{PH} is approximately the same as the amount of heat absorbed by the steam in the boilers. When η_{steam} is taken into account, is the calculation of the heat/power consumption by the steam boilers E_{steam} represented in equation 6.27.

$$E_{steam} = \frac{Q_{DH} + Q_{PH}}{\eta_{steam}} \tag{6.27}$$

To be able to calculate the power consumption for the ice water production is knowledge of the ice water mass flow rate $\dot{m}_{IW}$ in the UHT plant a necessity. This is done by using the energy balance in heat exchanger HX4 from figure 5.1 with equation 6.28. $\dot{m}_p$ is the mass flow rate of the products, Cp_p and Cp_{IW} are the specific heat capacity of respectively the products and the ice water, $T_{p,inlet}$ and $T_{p,outlet}$ are the inlet and outlet temperature of the products in HX4, while $T_{IW,inlet}$ and $T_{IW,outlet}$ are the inlet and outlet temperatures of the ice water in HX4.

$$\dot{m}_{IW} = \frac{\dot{m}_p * Cp_p * (T_{p,inlet} - T_{p,outlet})}{Cp_{IW} * (T_{IW,outlet} - T_{IW,inlet})} \tag{6.28}$$

When this is known can the heat transferred to the evaporator in the ammonia refrigeration system from the amount of ice water used in the UHT plant be calculated. Additional information for this calculation are the inlet temperature $T_{IWP,inlet}$ and the outlet temperature $T_{IWP,outlet}$ of the ice water in the ammonia evaporator. The heat transfer to the ammonia evaporator $Q_{ammonia_evap}$ is then calculated by equation 6.29.

$$Q_{ammonia,evap} = \dot{m}_{IW} * Cp_{IW} * (T_{IWP,inlet} - T_{IWP,outlet}) \tag{6.29}$$

There is no information of the performance of the ammonia refrigeration system for ice water production. However, the condensing and evaporation temperatures T_{cond} and T_{evap} are known, such that the COP_{carnot} can be calculated. This is done in equation 6.30. Normally is the COP of such refrigeration systems approximately half the value of the COP_{carnot}, and this simplification is done to be able to calculate the power consumption of the ice water production E_{IW}. Equation 6.31 is utilized to calculate this value.

$$COP_{carnot} = \frac{T_{evap}}{T_{condens} - T_{evap}} \tag{6.30}$$

$$E_{IW} = \frac{Q_{ammonia_evap}}{0.5 * COP_{carnot}} \tag{6.31}$$

The hot tap water production is done in a heat exchanger, where surplus heat from the UHT plant is utilized to heat tap water. The heat exchanger is modelled as earlier with 100 % thermal efficiency and with countercurrent flows. The heat absorbed by hot tap water E_{TW} is calculated by equation 6.32. Here is $\dot{m}_{surplus}$ the mass flow rate of the surplus stream from the UHT plant, which in all systems are equal for condensing and liquid surplus streams. $Cp_{surplus}$ is the specific heat capacity of a liquid surplus stream, $T_{surplus,inlet}$ and $T_{surplus,outlet}$ are the temperatures of the liquid stream at the inlet and outlet of HX_{TW}, whereas $h_{g,inlet}$ and $h_{l,outlet}$ are respectively the specific enthalpy of saturated gas and liquid for condensing streams.

$$E_{TW} = \dot{m}_{surplus} * [Cp_{surplus} * (T_{surplus,inlet} - T_{surplus,outlet}) + (h_{g,inlet} - h_{l,outlet})] \tag{6.32}$$

7 Results and Discussion

7.1 Energy Analysis of System 1

Table 7.1 represents the heat transfer of the hot and cold utilities Q_{hot} and Q_{cold}, and the heat and power consumption of the utility production $E_{utility}$ in the existing plant. With heat and power consumption for steam production to direct heating and preheating, power consumption for ice water production to HX4 and heat recycled by hot water production, will the energy balance give a net heat and power deficit $E_{utility,sum}$ of 607.51 kW for the existing system.

Table 7.1: Heat flow in utilities and heat/power flow in utility production in existing plant

Utility	Q_{hot} [kW]	Q_{cold} [kW]	$E_{utility}$ [kW]
DH	469.47	-	541.74
HX_{PH}	82.25	-	94.91
HX4	-	-114.51	9.34
HX_{TW}	-	-38.94	-38.94
Sum	551.72	-153.45	607.51

A pinch analysis is utilized to identify a minimum hot utility Q_{Hmin} of 536.36 kW, a minimum cold utility Q_{Cmin} of 138.08 kW, and a pinch point temperature T^*_{pinch} of 73.5 °C for system 1. The streams from table 5.2 are used and a ΔT_{min} of 5 kelvin is applied. Figure 7.1 illustrates the hot and cold composite curves of system 1, whereas the results from the temperature interval method are attached in appendix B.1.

From the composite curves in figure 7.1 is a MER system created, which only utilizes Q_{Hmin} and Q_{Cmin} as external utilities. The MER system is illustrated in figure 7.2. This system would indeed require less external utility than the existing system, but it would unfortunately not be logical to do so. The whole function of the system is to heat and cool the product stream. In this system is the whole water circuit including stream K3, K4 and H3, which function in the existing system is to support the products by preheating in HX2 and cooling in HX3, only supporting the products with a fraction of its potential. These streams are mostly utilized to transfer heat between each other, which makes this water circuit wasted. Moreover, all capital costs of the new heat exchangers HX_{new1} and HX_{new2}, which are two

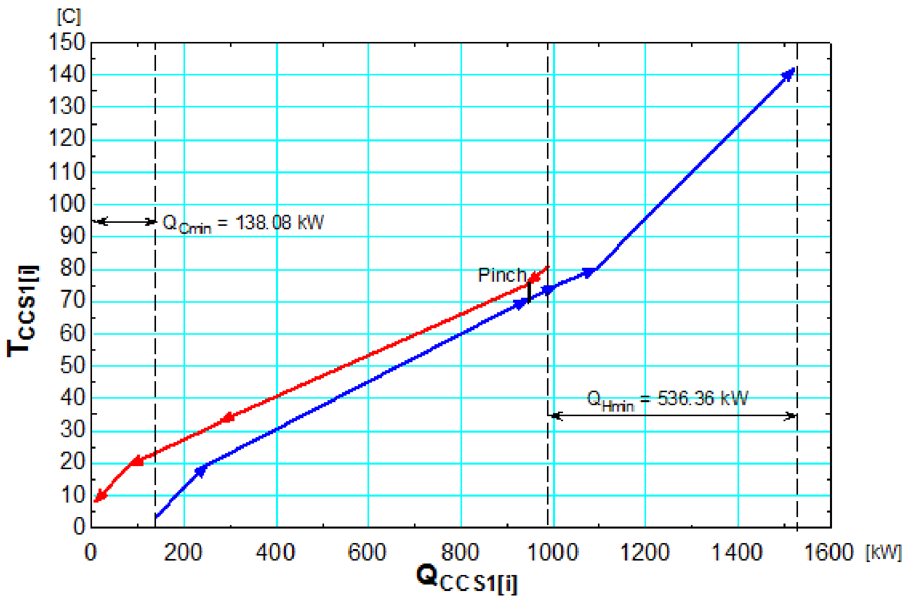

Figure 7.1: The hot and cold composite curves in system 1.

unnecessary heat exchangers, make this system insufficient. The preheating and cooling of products are mostly done by regeneration in HX_{new3}. This system proves thereby that regeneration in the UHT plant can potentially cover just below 50 % of the heat demand, and that the water circuit consisting of stream K3, K4 and H3 is inefficient with these operating conditions.

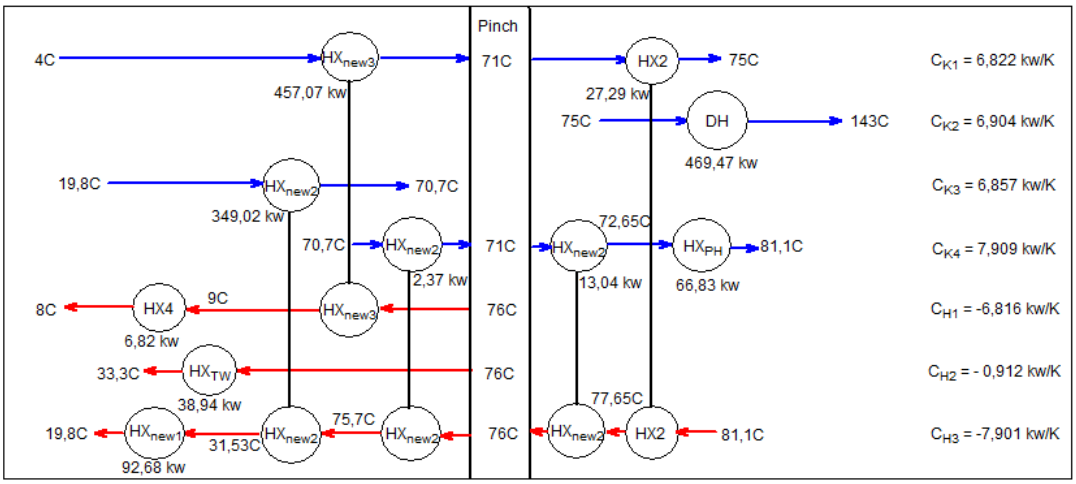

Figure 7.2: An example of a MER network of system 1.

The MER system in figure 7.2 is as mentioned above a poor exploitation of the available streams in this system. The existing system on the other hand, exploits the streams as they originally were made to. The heat exchanger network in figure 5.2 is elegantly designed. However, there is one change that can make the system operate more energy efficient than the existing one without any additional capital costs. Due to the criteria of a minimum temperature difference in the heat exchangers of 5 kelvin, is it unnecessary to heat stream K4 to 81.1°C when 80°C is sufficient, because the stream H3 can start the preheating of K1 at 80°C. The heating of K4 in heat exchanger HX_{PH} is done by water steam produced by the steam boilers. Because the boilers have an efficiency below unity, is integration of process streams more energy efficient. By lowering the start temperature of H3 means that this stream cannot do the whole preheating of stream K1. This can be solved by utilization of stream H2 in the already existing heat exchanger HX1, which is not in use in the existing system. Less energy is then available for hot tap water production in HX_{TW}, but net energy savings will be achieved. The new HEN system of system 1 is shown in figure 7.3. The heat exchanger HX_{theo} is just a theoretical heat exchanger, which is not real, but just included in the system to show that the change is possible with the existing streams.

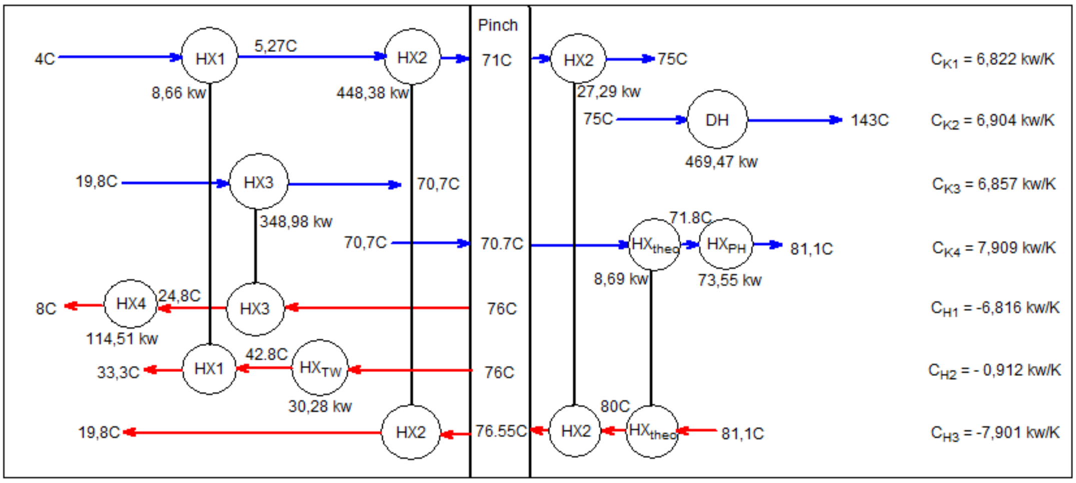

Figure 7.3: An optimal new HEN system for system 1.

Table 7.2 illustrates the heat and power required in the production of each utility and the net heat and power deficit of the three system options. As observed in the table is the MER system the most energy efficient system, but the small amount of energy savings are not convincing enough to defend the expensive and irrational new heat exchangers. The new HEN system is also more energy efficient than the existing one, with a net heat and power deficit $E_{utility,sum}$ of 605.66 kW. Since the usable surplus heat of 38.95 kW is identical to the existing system, is these savings a result of less combined heat and power consumption. This is nevertheless not dramatic, since no real investments is required by this system. This is only the most effective operation of the existing plant under these specific product qualifications. The table shows an extremely low difference in $E_{utility,sum}$ of 1.38 kW between the new HEN system and the existing one.

Table 7.2: Heat/power required in utility production in the existing plant, a MER system for system 1 and a new HEN system for system 1.

System	E_{DH} [kW]	E_{PH} [kW]	E_{IW} [kW]	E_{TW} [kW]	Sum [kW]
Existing system	541.74	94.91	9.34	-38.95	607.04
MER system	541.74	77.11	8.11	-38.95	588.01
HEN system	541.74	84.87	9.34	-30.29	605.66

7.2 Energy Analysis of System 2

A pinch analysis is utilized to identify a minimum hot utility Q_{Hmin} of 400.44 kW, a minimum cold utility Q_{Cmin} of 512.38 kW, and a pinch point temperature T^*_{pinch} of 87.5 for system 2. The streams from table 5.3 are used and a ΔT_{min} of 5 kelvin is applied. Figure 7.4 illustrates the hot and cold composite curves of system 2, whereas the results from the temperature interval method are attached in appendix B.2.

From the composite curves in figure 7.4 is a MER system created, which only utilizes Q_{Hmin} and Q_{Cmin} as external utilities. An example of a MER system is illustrated in 7.5. This system would indeed be the most energy efficient option of HEN for this system, but it is unfortunately not possible to carry it out. The reason for this is that the direct heating is forced in the system firstly to maintain the product quality, but it is also forced since the streams H2 and H4 are products of the steam production for direct heating purposes. A MER system would always break this criteria since the direct heating starts at a temperature lower than the pinch point temperature, which means that parts of the hot utility DH always are located below the pinch point.

The HEN system for system 2 is absolutely bound to the direct heating, which implies 469.47 kW of external hot utility in the system. 69.04 kW of this hot utility is required under the pinch, which means that this amount of heat is also added to the cold external utilities. However, this additional cold utility can be utilized in

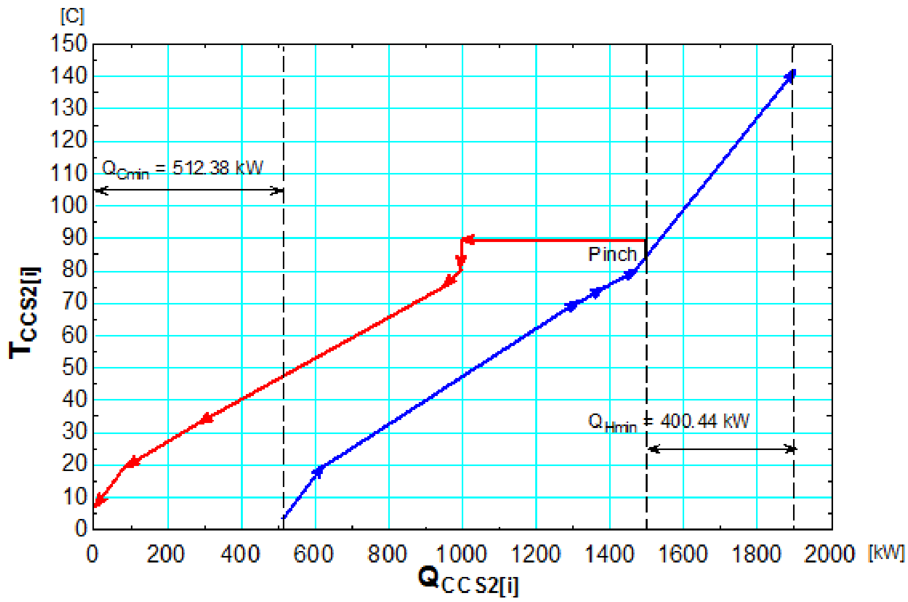

Figure 7.4: The hot and cold composite curves in system 2.

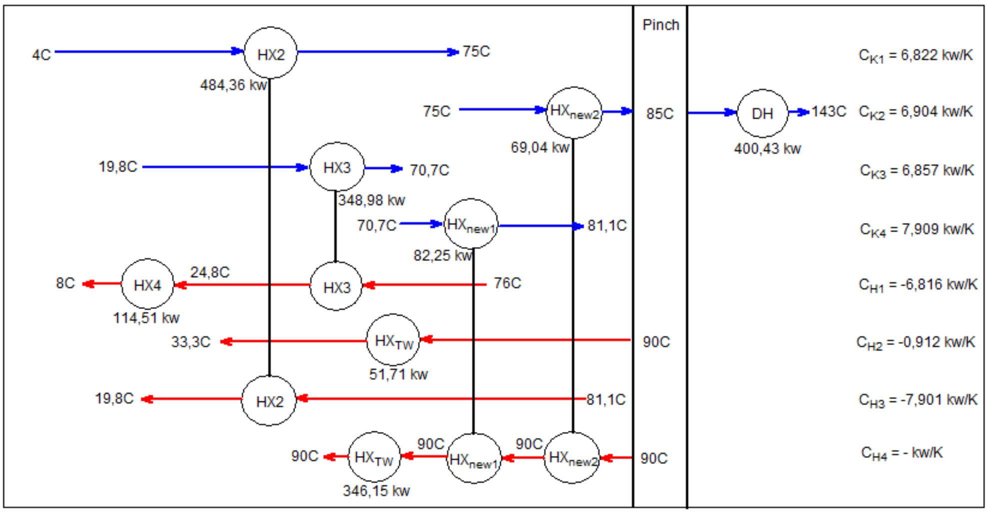

Figure 7.5: An example of a MER network of system 2.

hot tap water production, such that it is not completely lost. The HEN network of system 2 is illustrated in figure 7.6.

As figure 7.6 shows, there is no external hot utility for stream K4. However, the water circuit consisting of K3, K4 and H3 still requires some additional mass flow input. To avoid additional steam production for this purpose can this external mass flow input come from the condensing stream H4, such that this stream practically replaces steam from the steam boilers in HX_{PH}. This heat exchanger is in figure 7.6 called HX_{new1}. Table 7.3 lists the heat transfer of the hot and cold utilities Q_{hot}

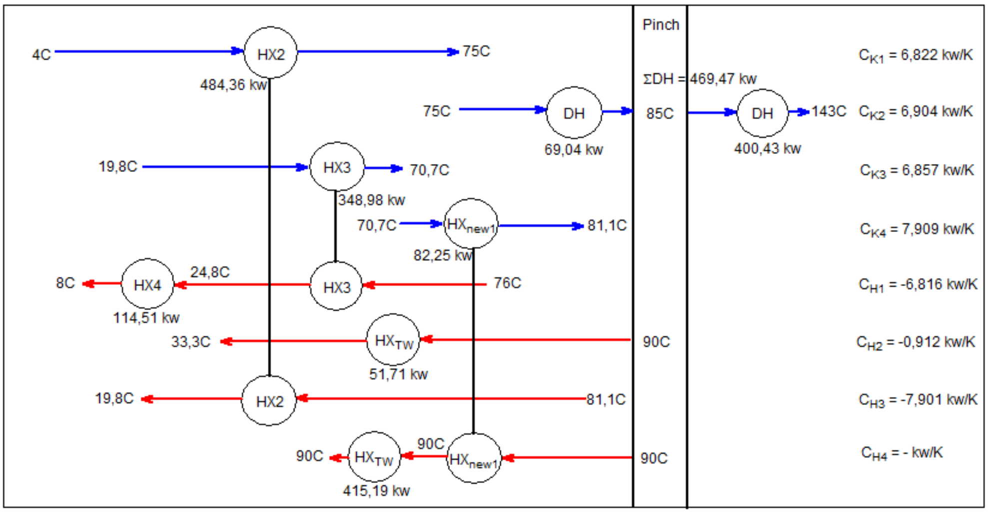

Figure 7.6: An optimal HEN system of system 2.

and Q_{cold}, and the heat and power consumption of the utility production $E_{utility}$ in the new HEN system. This system has a combined heat and power consumption of 551.08 kW and a usable surplus heat of 466.90 kW, making the net heat and power deficit $E_{utility,sum}$ only 84.18 kW. The large amount of surplus heat shows the potential of utilizing the exiting water vapour stream from the vacuum tank (stream H4). However, most of this potential are not required within the system, and must be utilized for purposes such as hot tap water production, space heating or similar demands in the factory. Whether the entire surplus heat potential can be utilized or not, is not known.

Table 7.3: Heat flow in utilities and heat/power flow in utility production in system 2.

Utility	Q_{hot} [kW]	Q_{cold} [kW]	$E_{utility}$ [kW]
DH	469.47	-	541.74
HX_{PH}	0	-	0
HX4	-	-114.51	9.34
HX_{TW}	-	-466.90	-466.90
Sum	469.47	-581.41	84.18

7.3 Energy Analysis of System 3

A pinch analysis is utilized to identify a minimum hot utility Q_{Hmin} of 400.44 kW, a minimum cold utility Q_{Cmin} of 459.29 kW, and a pinch point temperature T^*_{pinch} of 87.5 °C in system 3. The streams from figure 5.4 are used and a ΔT_{min} of 5 kelvin is applied. Figure 7.7 illustrates the hot and cold composite curves of system 3, whereas the results from the temperature interval method are attached in appendix B.3.

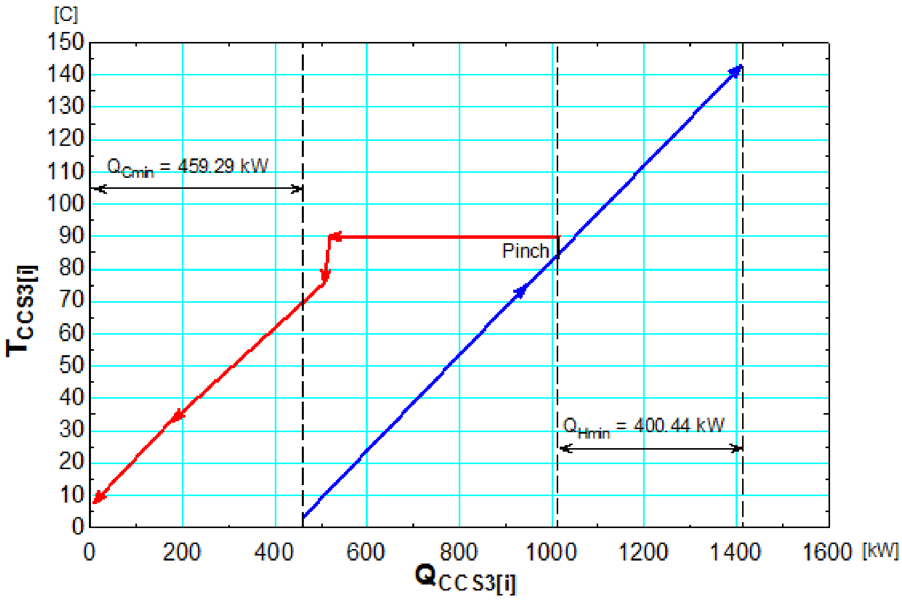

Figure 7.7: The hot and cold composite curves in system 3.

From the composite curves in figure 7.7 is a MER system created, which only utilizes Q_{Hmin} and Q_{Cmin} as external utility. An example of a MER system is illustrated in figure 7.8. This system would indeed be the most energy efficient option of HEN for this system, but it is unfortunately not possible to carry it out. The reason for this is that the direct heating is forced in the system firstly to maintain the product quality, but it is also forced since the streams H2 and H4 are products of the steam production to direct heating purposes. A MER system would always break this criteria since the direct heating starts at a temperature lower than the pinch point temperature, which means that parts of the hot utility DH always is located below the pinch point.

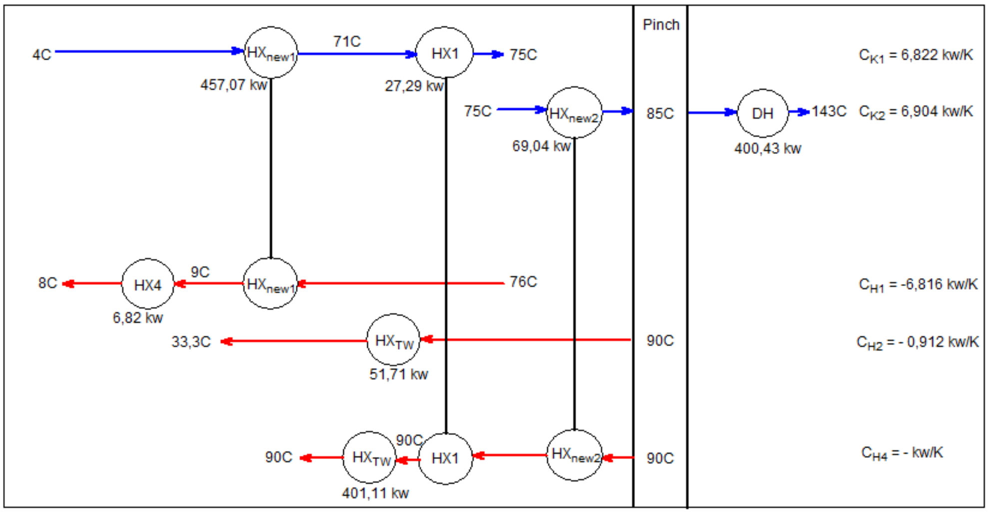

Figure 7.8: An example of a MER network of system 3.

The HEN system for system 3 is absolutely bound to the direct heating, which implies 469.47 kW of external hot utility in the system. 69.04 kW of this hot utility is required under the pinch, which means that this amount is also added to the cold external utilities. However, this additional cold utility can be utilized in hot tap water production, such that it is not completely lost. The HEN network of system 3 is illustrated in figure 7.9.

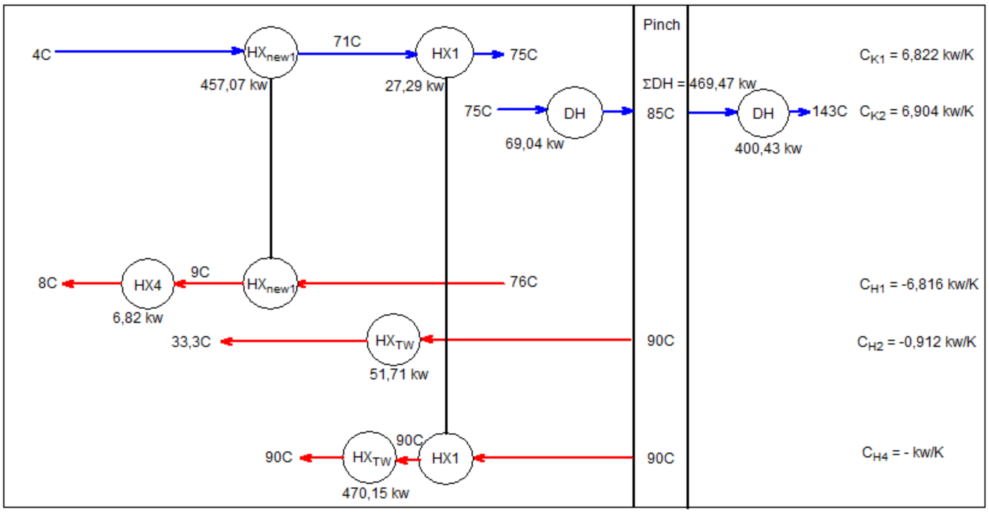

Figure 7.9: An optimal HEN network of system 3.

As figure 7.9 shows, most of the preheating and cooling is done by regeneration in the new heat exchanger HX_{new1}, which would be able to perform 457.07 kW of regeneration. The potential of this application is already proven in the MER system from system 1, but when it did no make sense to apply it there, this system would be much more effective. By removing the water circuit consisting of stream K3, K4 and H3, regeneration emerges as a better solution for preheating and cooling of products.

Because of the temperature ranges of stream K1 and H1 is it obvious that the whole preheating and cooling cannot be done by regeneration. Surplus heat from the condensing stream H4 is utilized to bring stream K1 up to its final temperature of 75 °C in HX1, whereas ice water must bring stream H1 down to its final temperature of 8°C in HX4. To meet the criteria of cooling, the heat exchanger HX4 will only be needed for 6.82 kW of ice water to cool stream H1 from 9°C to 8°C. The electric power required to produce this amount of ice water is close to negligible in the energy balance for the whole system. The use of heat exchanger HX4 could also be avoided either by letting the product stream leave the system at 9°C, or by setting the minimum temperature difference ΔT_{min} to 4 kelvin. However, the established boundaries counts for all systems to be able to compare their respective performance, and for this reason are the options above not considered further.

Except from bringing stream K1 up to its final temperature in HX1, can the condensing stream H4 and the latent heat of H2 only be utilized as usable system surplus heat. With this surplus heat of 521.86 kW and a combined heat and power consumption of 541.90, is the net heat and power deficit $E_{utility,sum}$ in this system only 24.04 kW, which is shown in table 7.4. This table lists the heat transfer of the hot and cold utilities Q_{hot} and Q_{cold}, and the heat and power consumption of the utility production $E_{utility}$. The large amount of surplus heat shows the potential of utilizing the exiting water vapour stream from the vacuum tank (stream H4). However, most of this potential are not required within the system, and must be utilized for purposes such as hot tap water production, space heating or similar demands in the factory. Whether the entire surplus heat potential can be utilized or not, is not known.

Table 7.4: Heat flow in utilities and heat/power flow in utility production in system 3.

Utility	Q_{hot} [kW]	Q_{cold} [kW]	$E_{utility}$ [kW]
DH	469.47	-	541.74
HX_{PH}	0	-	0
HX4	-	-6.82	4.16
HX_{TW}	-	-521.86	-521.86
Sum	469.47	-528.68	24.04

7.4 Energy Analysis of System 4

The pinch analysis on system 2 shows that the condensing stream H4 from the vacuum tank is located under the system pinch temperature, which can be observed either in figure 7.5 or 7.6. This criteria for implementing a heat pump will therefore stand. Additionally is the heat source, stream H4, a condensing stream with a large potential heat output, and the composite curves from figure 7.4 are therefore certainly open. With this information is a heat pump installed to serve the direct heating process. Before the performance of the heat pump can be calculated, is an isentropic efficiency η_{is} of 0.73 for all three compressor steps roughly estimated. The estimation process is illustrated in figure 7.10. The pressure ratios π are respectively 2.181, 2.163 and 2.145 for compressor step 1, 2 and 3, and varies because of the loss calculation utilized in the heat pump model. However, because the variation is so small, is a common π of 2.16 utilized for the η_{is} estimation.

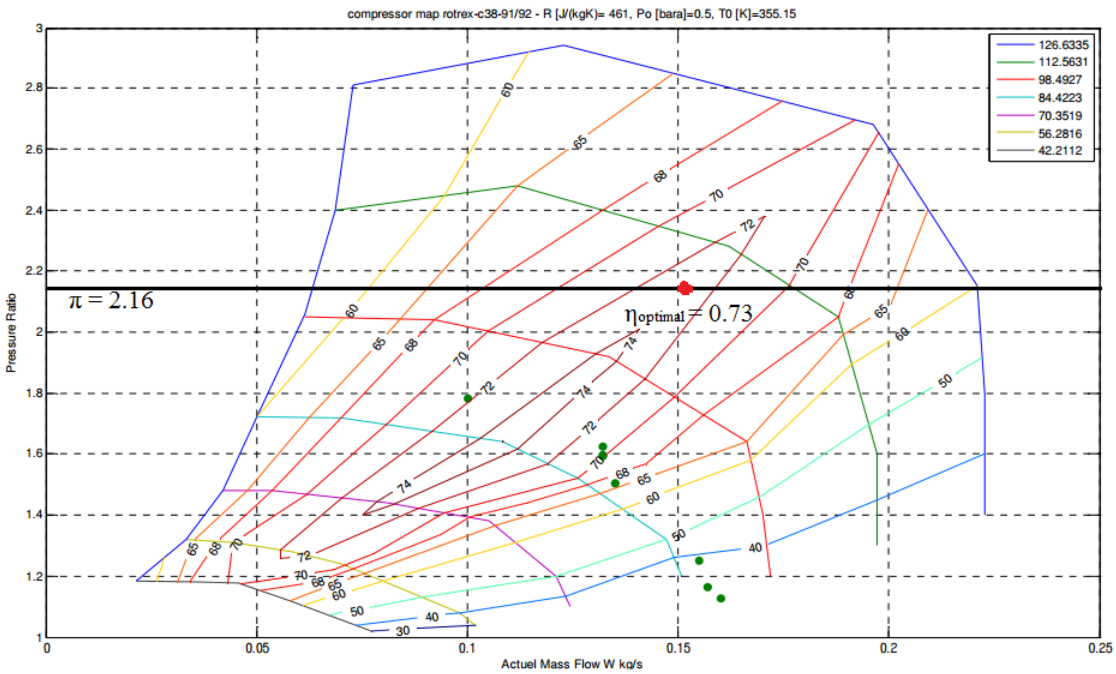

Figure 7.10: Rough estimation of the isentropic efficiency in the compressors by using existing equipment.

A refrigerant mass flow rate $\dot{m}_{HP}$ of 721.44 kg/h is calculated from a direct heating heat demand Q_{DH} of 469.47 kW, a superheated temperature T_{sup} of 244.42 °C out of the last compressor step, a condensation temperature T_{cond} of 158 °C and a subcooling temperature T_{sub} of 143 °C. With $\dot{m}$, π and η_{is} known, is an overall compressor power $W_{compr,sum}$ of 122.70 kW achieved. This gives a COP of 3.826. A summary of important values from the heat pump is listed in table 7.5. T_{in} and T_{out}

represent respectively the suction and discharge temperatures of each compressor step.

Table 7.5: Heat pump performance.

Compressor	$\dot{m}_{HP}$ [kg/h]	π	T_{in} [°C]	T_{out} [°C]	W_{compr} [kW]	Q_{DH} [kW]	COP
Step 1	721.44	2.181	95.00	197.16	39.57	-	-
Step 2	721.44	2.163	114.82	219.47	40.88	-	-
Step 3	721.44	2.145	137.29	244.42	42.25	-	-
Sum	-	-	-	-	122.70	469.47	3.826

After the heat pump implementation is a new pinch analysis done to identify a minimum hot utility Q_{Hmin} of 66.87 kW, a minimum cold utility Q_{Cmin} of 99.14 kW, and two pinch point temperatures T^*_{pinch} of 73.5 °C and 73.2 °C. The streams from table 5.5 are used and a ΔT_{min} of 5 kelvin is applied. Figure 7.11 illustrates the hot and cold composite curves of system 4, whereas the results from the temperature interval method are attached in appendix B.4.

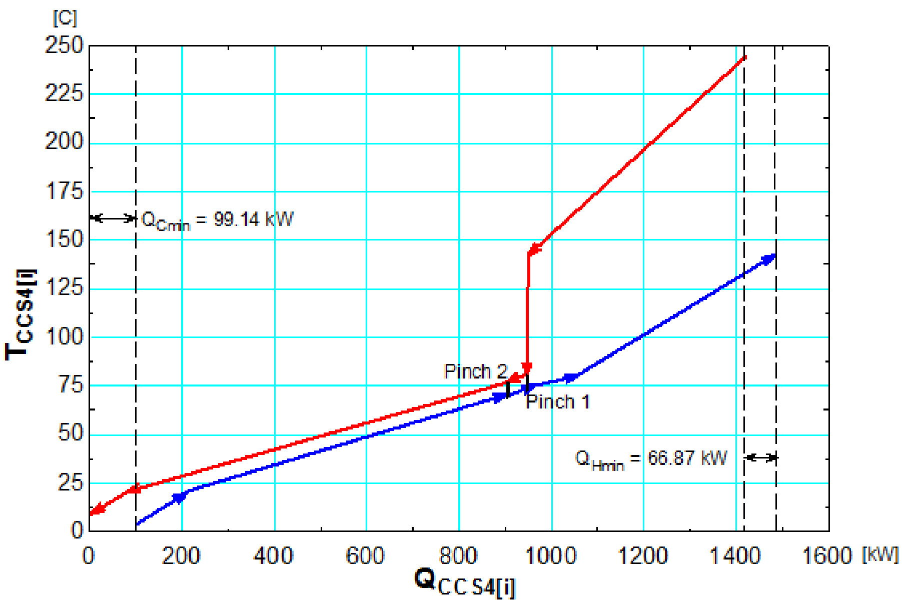

Figure 7.11: The hot and cold composite curves in system 4.

From the composite curves in figure 7.11 is a MER system created, which only utilizes Q_{Hmin} and Q_{Cmin} as external utilities. An example of a MER system is illustrated in figure 7.12. This system would indeed require the minimum amount of utilites, but it would unfortunately not be logical to do so. The whole function of the system is to heat and cool the product stream. In this system is the whole water circuit including stream K3, K4 and H3, which function in the existing system is to support the products by preheating in HX2 and cooling in HX3, only supporting the products with a fraction of its potential. These streams are mostly utilized to transfer heat between each other, which makes this water circuit wasted. Moreover, the capital costs of the new heat exchanger HX_{new2}, which is an unnecessary heat exchanger, makes this system insufficient. The preheating and cooling of products are mostly done by regeneration in HX_{new1}.

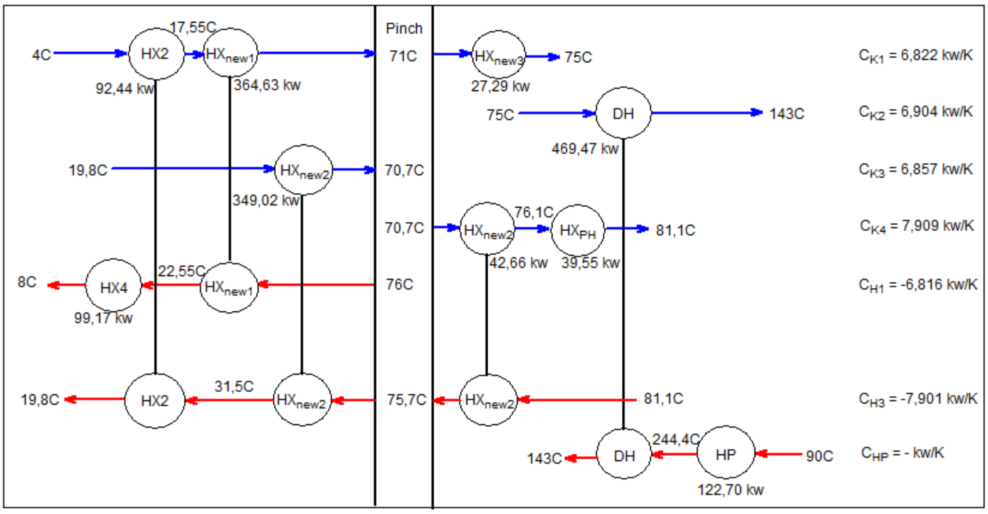

Figure 7.12: An example of a MER network of system 4.

A new HEN system which utilizes the streams more elegantly is made, and can be observed in figure 7.13. Here is stream K3 and stream H3 utilized for preheating and cooling of products, which they are made for originally. This gives a small amount of extra hot utility consumption in HX_{PH}, but avoids investments of several new heat exchangers. The only difference between this HEN system and the existing one, is the direct heating process with use of the heat pump and the removal of hot tap water production in HX_{TW}.

Table 7.3 lists the heat transfer of the hot and cold utilities Q_{hot} and Q_{cold}, and the heat and power consumption of the utility production $E_{utility}$. With a power consumption of 122.70 kW, are the energy savings associated with utilizing a heat

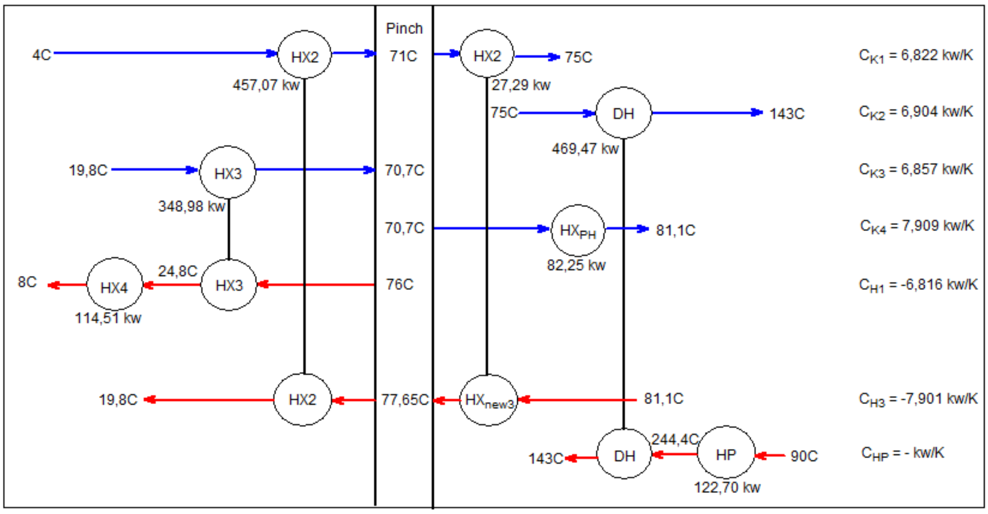

Figure 7.13: An optimal HEN network of system 4.

pump in the direct heating process 419.04 kW. The heat pump utilizes the whole stream HP, and because of the heat pump is the stream H2 from the previous systems unavailable for utilization in this system, which means that no surplus heat is available. Otherwise is the combined heat and power consumption 226.25 kW, which again gives a net heat and power deficit $E_{utility,sum}$ of 226.25 kW in this system.

Table 7.6: Heat flow in utilities and heat/power flow in utility production in system 4.

Utility	Q_{hot} [kW]	Q_{cold} [kW]	$E_{utility}$ [kW]
DH	469.47	-	122.70
HX_{PH}	82.25	-	94.91
HX4	-	-114.51	9.34
HX_{TW}	-	0	0
Sum	551.72	-114.51	226.95

7.5 Energy Analysis of System 5

The pinch analysis on system 3 shows that the condensing stream H4 from the vacuum tank is located under the system pinch temperature, which can be observed in figure 7.8 or 7.9. This criteria for implementing a heat pump will therefore stand.

Additionally is the heat source, stream H4, a condensing stream with a large potential heat output, and the composite curves from figure 7.4 are therefore certainly open. With this knowledge is a three stage heat pump made to serve the direct heating process. Since the heat pump in this system serves the same purpose as the one in system 4, in addition to be of equal design, will a η_{is} of 0.73, a $\dot{m}_{HP}$ of 721.44 kg/h, a $W_{compr,sum}$ of 122.70 kW, a Q_{DH} of 469.47 kW and a COP of 3.826 also yield in this system. A summary of important values associated with the heat pump can be found in table 7.5 from section 7.4.

After the heat pump implementation is a new pinch analysis done to identify a minimum hot utility Q_{Hmin} of 27.65 kW, a minimum cold utility of Q_{Cmin} of 6.82 kW and a new pinch temperature T^*_{pinch} of 6.5 °C. The streams from figure 5.7 are used and a ΔT_{min} of 5 kelvin is applied. Figure 7.14 illustrates the hot and cold composite curves of system 5, whereas the results from the temperature interval method are attached in appendix B.5.

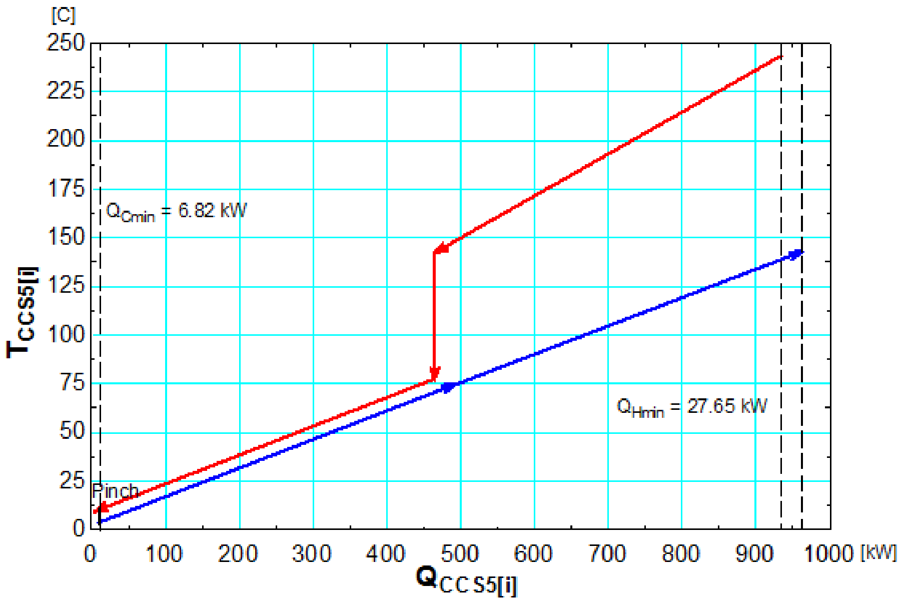

Figure 7.14: The hot and cold composite curves in system 5.

From the composite curves in figure 7.14 is a MER system created, which only utilizes Q_{Hmin} and Q_{Cmin} as external utilities. An example of a MER system is illustrated in figure 7.12. Since this system only includes four process streams and the direct heating is done by the heat pump, are the options for heat integration limited to integration between stream K1 and H1. A "common sense" HEN system will regenerate the exact same amount of heat as the MER system in HX_{new1} of 457.07 kW, which make these two systems identical. This regeneration is extremely effective under this operation, and close to 50 % of the total heat demand is achieved due to this implementation. The only external utilities are steam to fulfil the last part of preheating in heat exchanger HX_{new2} of 27.29 kW, in addition to a small amount of ice water cooling for the last part of the cooling of products in HX4 of 6.82 kW.

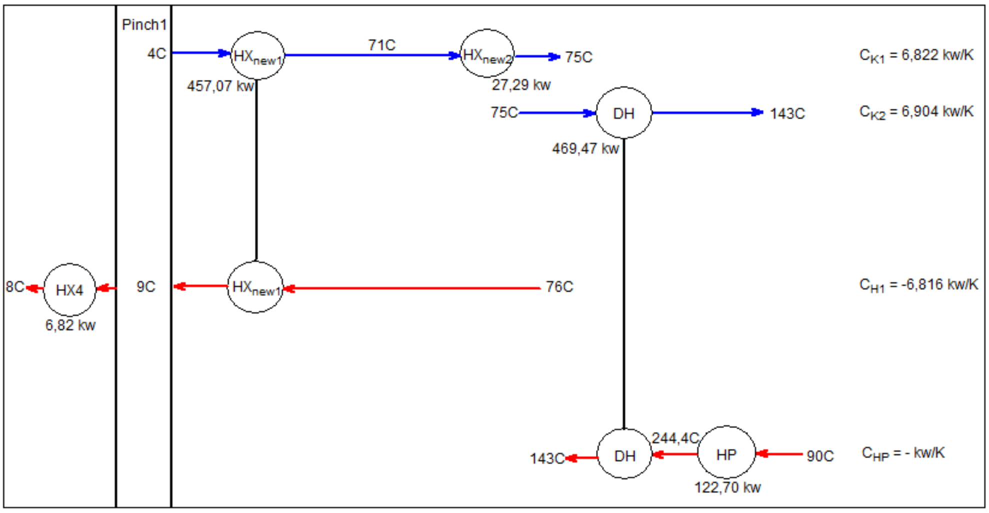

Figure 7.15: An optimal HEN system of system 5.

Table 7.7 lists the heat transfer of the hot and cold utilities Q_{hot} and Q_{cold}, and the heat and power consumption of the utility production $E_{utility}$. With a power consumption of 122.70 kW, are the energy savings associated with utilizing a heat pump in the direct heating process 419.04 kW. The heat pump utilizes the whole stream HP, and because of the heat pump is stream H2 from the previous systems unavailable in this system, which means that no surplus heat is available. Otherwise is the combined heat and power consumption 158.35 kW, which again gives a net heat and power deficit $E_{utility,sum}$ of 158.35 kW.

Table 7.7: Heat flow in utilities and heat/power flow in utility production in system 5.

Utility	Q_{hot} [kW]	Q_{cold} [kW]	$E_{utility}$ [kW]
DH	469.47	-	122.70
HX_{PH}	27.29	-	31.49
HX4	-	-6.82	4.16
HX_{TW}	-	0	0
Sum	496.76	-6.82	158.35

7.6 Discussion and Comparison of all Systems

The most important results from the simulations are listed in table 7.8. These results include the heat and power consumption E_{DH}, E_{PH} and E_{IW} for utility production, the surplus heat E_{TW} and the net heat and power deficit $E_{utility,sum}$ for all systems. Only the new HEN systems, which were created on common sense terms, are included in the table. Hence, the MER systems, which included irrational heat exchanger networks, are not further analysed.

Table 7.8: Heat and power consumption, surplus heat and net heat and power deficit in all systems.

System	E_{DH} [kW]	E_{PH} [kW]	E_{IW} [kW]	E_{TW} [kW]	$E_{utility,sum}$ [kW]
Existing system	541.74	94.91	9.34	-38.95	607.04
HEN system 1	541.74	84.87	9.34	-30.29	605.66
HEN system 2	541.74	0	9.34	-466.90	84.18
HEN system 3	541.74	0	4.16	-521.86	24.04
HEN system 4	122.70	94.91	9.34	0	226.95
HEN system 5	122.70	31.49	4.16	0	158.35

From these results in addition to the heat exchanger networks from each system, can advantages and disadvantages of each system be identified. To analyse these advantages and disadvantages are several factors vital. These factors are:

- Combined heat and power consumption for utility production in the system.
- Usable surplus heat in the system.
- Net heat and power deficit in the system.

- Sustainability of system modifications.
- System flexibility.
- Potential capital costs of new equipment.

The combined heat and power consumption in all systems is mainly affected by two factors; how the direct heating is done and how the preheating is done. Regarding the direct heating, is this done by steam boilers in system 1, 2 and 3, and by a heat pump in system 4 and 5. The heat pump implementation in system 4 and 5 saves 419.04 kW compared to steam generation for the same purpose, making system 4 and 5 superior in this perspective. This is because the steam generation is done originally by inefficient steam boilers, whereas the heat pump COP is 3.826. Regarding the preheating, is this mainly done either by the water stream H3 in system 1, 2 and 4, or by regeneration in system 3 and 5. The regeneration saves around 60 kW compared to utilization of stream H3 for the same purpose, making system 3 and 5 superior in this perspective. This is because regeneration is associated with lower heat exchanger losses. Otherwise is the power consumption from ice water production in all systems only a small portion of the total heat and power consumption, due to an energy efficient refrigeration system.

The surplus heat in each system is utilized only by stream H2 or H4 in the heat exchanger called HX_{TW}. This heat exchanger is originally used for hot tap water production, but is in system 2 and 3 used as a collective term for surplus heat at a sufficient temperature level for hot tap water production, space heating or other such demands. Because system 2 and 3 utilize the full potential of stream H2 and H4, are these two systems superior in this perspective with respectively 466.90 kW and 521.86 kW of surplus heat. Because the existing system and HEN system 1 are not modified such that the condensing stream from the vacuum tank (stream H4) is utilized, are these two systems associated with only a fraction of the surplus heat of system 2 and 3. Due to the included heat pump solution, which denies access to surplus heat from stream H2 and H4, is no system surplus heat available in system 4 and 5.

Of the systems with modifications are system 2 and 3 associated with the lowest net heat and power deficit of respectively 84.18 kW and 24.04 kW compared to system 4 and 5 with a net heat and power deficit of 226.95 kW and 158.35 kW. The main reason for this is that the surplus heat in system 2 and 3 is greater than the net recycled heat from the heat pump system. Otherwise are system 3 and 5 using regeneration as preheating and cooling of products, which explains why system 3 has a better performance than system 2 and system 5 has a better performance than system 4. The existing system and HEN system 1 have a distinctly higher net heat and power deficit than the systems with modifications, which proves that all modifications potentially are associated with large energy savings.

The question of sustainable modifications are relevant for all new HEN solutions, except from HEN system 1. The main challenge for all new modifications, is the exploitation of the occurring water vapour in the vacuum tank. Whether or not an utilization of this stream affects the direct heating process is not known, and must be further investigated. System 2 and 3 are both dependent on the possibility of exploiting the system surplus heat. First of all must there be enough energy demand present in the factory. Secondly must there be possible to implement the new required equipments for utilization of this surplus heat. The surplus heat is most relevant for hot tap water production in the factory, because the demand is approximately constant over the year, and the production of hot tap water can be done in intervals. This is relevant due to the semi-continuous operation of the UHT-plant. However, this creates a challenge for utilization of the surplus heat for space heating purposes or similar demands, which requires more continuous operation. If this is the case cannot the whole surplus heat be exploited, and parts of the stream must rather be drained.

For system 4 and 5 must the heat pump be able to release sterile water into the injector. Whether this can be done without contamination are dependent on the compressor technology and a filter system. The compressors should be oil-free, which should not be a problem with turbo compressors. However, because of the properties of water will normal turbo compressors have difficulties with the high required rotational speed for the temperature lift in the heat pump. To obtain this rotational speed with normal turbo compressor technology, are the compressor capital costs large. To avoid such capital costs, are turbo compressors with a planetary gear desired. These compressors are however only delivered with oil lubrication. On the other hand, since turbo compressors can be produced without oil lubrication, are there possibilities for development of oil-free compressors with a planetary gear in the close future.

This analysis has only focused on the operation of the products which have the highest cooling demands. The new HEN systems for system 3 and 5 utilizes regeneration in stead of the existing water circuit for this purpose. The performance of these two systems will for treatment of products with lower cooling demands be worse than this analysis concludes with, because less regeneration is possible with such operating conditions. On the other hand will the remaining systems, which include the water circuit from the existing system, experience approximately no change in performance when the operating conditions are changed. The water circuit is created for flexibility of the UHT-plant operation, which means that if the plant operation continues as it does today will system 1, 2 and 4 have a flexibility advantage.

The capital cost of each system is a vital factor when factories invest in new equipment. System 4 and 5 are most likely associated with the highest capital costs, due to the heat pump installation. The capital costs of system 2 and 3 are highly dependent on how much of the surplus heat that can be utilized. A great amount of capital costs are required if the whole potential is to be exploited. How high these costs are compared to other systems are dependent on which purpose this surplus heat shall serve, and therefore impossible to estimate. Otherwise are the existing system and the new HEN system 1 associated with no capital costs.

8 Conclusion and Suggestion for Further Work

8.1 Conclusion

The main results of the energy analysis are:

- The existing system has a net heat and power deficit of 607.44 kW. HEN system 1, an optimized version of the existing without any modifications, obtained a net heat and power deficit of 605.66 kW. Moreover, the combined heat and power consumption in both these systems exceeds 630 kW
- System 2 and 3 obtain the lowest net heat and power deficit of respectively 84.18 kW and 24.04 kW when the full surplus heat potential of a vapour water stream exiting the vacuum tank is exploited. However, the possibilities for utilizing the full surplus heat potential is questionable.
- The same water vapour steam can be utilized in a high temperature heat pump to carry out the full direct heating process in the UHT plant. In system 4 and 5 is this done, which gave a net heat and power deficit of 226.95 kW for system 4 and 158.35 kW for system 5. Moreover, 419.04 kW of energy savings for direct heating purposes is achieved by heat pump implementation.
- System 3 and 5 benefit in an energy efficiency perspective from the substitution of a water circuit from the existing system with regeneration for the purpose of preheating and cooling of products. However, by this modification are these systems less flexible and more vulnerable for change of UHT-plant operating conditions.
- In system 2, 3, 4 and 5 is it assumed that a water vapour stream from the vacuum tower in the UHT plant can be exploited. Whether this modification is possible or not is unknown, but the result from this thesis shows that an utilization of this stream can result in large energy savings.

8.2 Suggestion for Further Work

Based on the knowledge from this report, the following is suggested to be investigated in future work:

- Perform an energy analysis on the UHT plant with varying operating conditions.
- Investigate how the new systems proposed in this thesis will perform with varying operating conditions.

- Investigate which processes that occur in the vacuum tank, and if the steam from this tank can be utilized without affecting product quality.
- Development of oil-free water based turbo compressor with a planetary gear for this application.
- Perform a more accurate novel pinch analysis on the UHT plant, which could take into account more limitations associated with UHT plants, such as semi-continuous operation and water consumption.

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A Appendix - Information about the UHT plant

This information is locked for confidentiality reasons. If authorized, it can be retrieved at the Department of Energy and Process Engineering at NTNU.

B Appendix - Temperature interval method results

B.1 Results of temperature interval method - System 1

i	T* [°C]	C_i [kW/K]	Q_i [kW]	ΔQ_i [kW]	ΔQ_i* [kW]
					536.36 (Q_{Hmin})
	145.5				
1		6.904	427.37	427.37	108.99
	83.6				
2		14.813	74.04	501.44	34.92
	78.6				
3		6.912	7.60	509.04	27.32
	77.5				
4		6.829	27.32	536.36	0 (PINCH)
	73.5				
5		-0.899	-0.27	536.09	0.27
	73.2				
6		-1.952	-82.74	453.34	83.01
	30.8				
7		-1,039	-8.83	444.51	91.85
	22.3				
8		-7.896	-39.48	405.03	131.33
	17.3				
9		0.005	0.06	405.09	131.27
	6.5				
10		-6.816	-6.82	398.27	138.08 (Q_{Cmin})
	5.5				

B.2 Results of temperature interval method - System 2

i	T* [°C]	C_i [kW/K]	Q_i [kW]	ΔQ_i [kW]	ΔQ_i* [kW]
					400.44 (Q_{Hmin})
	145.5				
1		6.904	400.44	400.44	0 (PINCH)
	87.5				
2		-	-497.44	-97.00	497.44
	87.5				
3		5.992	23.37	-73.63	474.07
	83.6				
4		13.901	69.50	-4.12	404.57
	78.6				
5		6.000	6.60	2.47	397.97
	77.5				
6		5.917	23.67	26.14	374.30
	73.5				
7		-0.899	-0.27	25.87	374.57
	73.2				
8		-1.952	-82.74	-56.87	457.31
	30.8				
9		-1.031	-8.83	-65.70	466.15
	22.3				
10		-7.896	-39.48	-105.18	505.63
	17.3				
11		0.005	0.06	-105.12	505.57
	6.5				
12		-6.816	-6.82	-111.94	512.38 (Q_{Cmin})
	5.5				

B.3 Results of temperature interval method - System 3

i	T* [°C]	C_i [kW/K]	Q_i [kW]	ΔQ_i [kW]	ΔQ_i* [kW]
					400.44 (Q_{Hmin})
	145.5				
1		6.904	400.44	400.44	0 (PINCH)
	87.5				
2		-	-497.44	-97.00	497.44
	87.5				
3		5.992	59.92	-37.08	437.52
	77.5				
4		5.910	23.64	-13.44	413.88
	73.5				
5		-0.907	-38.72	-52.16	452.60
	30.8				
6		0.005	0.13	-52.02	452.47
	6.5				
7		-6.816	-6.82	-58.84	459.29 (Q_{Cmin})
	5.5				

B.4 Results of temperature interval method - System 4

i	T* [°C]	C_i [kW/K]	Q_i [kW]	ΔQ_i [kW]	ΔQ_i* [kW]
					66.87 (Q_{Hmin})
	246.9				
1		-	-469.49	-469.49	536.36
	145.5				
2		6.904	427.37	-42.12	108.99
	83.6				
3		14.813	74.07	31.95	34.92
	78.6				
4		6.912	7.60	39.55	27.32
	77.5				
5		6.829	27.32	66.87	0 (PINCH)
	73.5				
6		0.013	0	66.87	0 (PINCH)
	73.2				
7		-1.039	-52.90	13.97	52.90
	22.3				
8		-7.896	-39.48	-25.51	92.38
	17.3				
9		0.005	0.06	-25.45	92.32
	6.5				
10		-6.816	-6.82	-32.26	99.14 (Q_{Cmin})
	5.5				

B.5 Results of temperature interval method - System 5

i	T* [°C]	C_i [kW/K]	Q_i [kW]	ΔQ_i [kW]	ΔQ_i* [kW]
					27.65 (Q_{Hmin})
	246.9				
1		-	-469.49	-469.49	497.14
	145.5				
2		6.904	469.49	0	27.65
	77.5				
3		6.822	27.29	27.29	0.37
	73.5				
4		0.005	0.37	27.65	0 (PINCH)
	6.5				
5		-6.816	-6.82	20.84	6.82 (Q_{Cmin})
	5.5				

C Appendix - EES Program Codes

```
File:Complete script of master thesis milk cp.EES                                    18.06.2016 11.44.52  Page 1
     EES Ver. 9.935: #3812: For use only by students and faculty Dept. of Energy and Process Engineering, NTNU, NORWAY

"COMPLETE SCRIPT OF ALL PROGRAMMING IN MASTER THESIS"

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"MODEL OF SPECIFIC HEAT CAPACITY CALCULATIONS OF MILK FOR FURTHER USE"

"MILK CONSTISTS MOSTLY OF WATER, FAT, PROTEIN AND CARBOHYDRATES. ONLY THESE COMPOUNDS ARE
INCLUDED"
"THE OPERATION OF THE SYSTEM WHEN WHOLE MILK IS PRODUCED IS USED IN ALL CALCULATIONS. THIS
IMPLIES THESE MASS FRACTIONS"
x_fat = 0,035 "Mass fraction of fat in whole milk"
x_protein = 0,033 "Mass fraction of protein in whole milk"
x_carbs = 0,045 "Mass fraction of carbohydrates in whole milk"
x_water = 1 - (x_fat+x_protein+x_carbs) "Approx mass fraction of water in whole milk"

"Cp_fat = 1,9842 + (1,4733/1000)*T - (4,8008/1000000)*T^2"
"Cp_protein = 2,0082 + (1,2089/1000)*T - (1,3129/1000000)*T^2"
"Cp_carbs = 1,5488 + (1,9625/1000)*T - (5,9399/1000000)*T^2"
"Cp_water = table for specific heat capacity at either atmospheric pressure or saturated states are used"

"Specific heat capacity of milk at 4*C"
Cp_fat4C = 1,9842 + (1,4733/1000)*T_exp1 - (4,8008/1000000)*(T_exp1)^2 "Specific heat capacity of fat at 4*C"
Cp_protein4C =  2,0082 + (1,2089/1000)*T_exp1 - (1,3129/1000000)*(T_exp1)^2 "Specific heat capacity of protein at 4C"
Cp_carbs4C = 1,5488 + (1,9625/1000)*T_exp1 - (5,9399/1000000)*(T_exp1)^2 "Specific heat capacity of carbohydrates
at 4C"
Cp_water4C = 4,205 "From table at atmospheric pressure"
Cp_milk4C = x_fat*Cp_fat4C + x_protein*Cp_protein4C + x_carbs*Cp_carbs4C + x_water*Cp_water4C "Specific heat
capacity of milk at 4C"

"Specific heat capacity of milk at 75*C"
Cp_fat75C = 1,9842 + (1,4733/1000)*T_exp3 - (4,8008/1000000)*(T_exp3)^2 "Specific heat capacity of fat at 75*C"
Cp_protein75C =  2,0082 + (1,2089/1000)*T_exp3 - (1,3129/1000000)*(T_exp3)^2 "Specific heat capacity of protein at
75C"
Cp_carbs75C = 1,5488 + (1,9625/1000)*T_exp3 - (5,9399/1000000)*(T_exp3)^2 "Specific heat capacity of carbohydrates
at 75C"
Cp_water75C = 4,193 "From table at atmospheric pressure"
Cp_milk75C = x_fat*Cp_fat75C + x_protein*Cp_protein75C + x_carbs*Cp_carbs75C + x_water*Cp_water75C "Specific
heat capacity of milk at 75C"



"Specific heat capacity of milk at 143*C"
Cp_fat143C = 1,9842 + (1,4733/1000)*T_exp4 - (4,8008/1000000)*(T_exp4)^2 "Specific heat capacity of fat at 143*C"
Cp_protein143C =  2,0082 + (1,2089/1000)*T_exp4 - (1,3129/1000000)*(T_exp4)^2 "Specific heat capacity of protein at
143C"
Cp_carbs143C = 1,5488 + (1,9625/1000)*T_exp4 - (5,9399/1000000)*(T_exp4)^2 "Specific heat capacity of
carbohydrates at 143C"
Cp_water143C = 4,295 "Saturated liquid water at a temperetufe of 142,5C and pressure of approx 4 bar"
Cp_milk143C = x_fat*Cp_fat143C + x_protein*Cp_protein143C + x_carbs*Cp_carbs143C + x_water*Cp_water143C "
Specific heat capacity of milk at 143C"

"Specific heat capacity of milk at 76*C"
Cp_fat76C = 1,9842 + (1,4733/1000)*T_exp5 - (4,8008/1000000)*(T_exp5)^2 "Specific heat capacity of fat at 76*C"
Cp_protein76C =  2,0082 + (1,2089/1000)*T_exp5 - (1,3129/1000000)*(T_exp5)^2 "Specific heat capacity of protein at
76C"
Cp_carbs76C = 1,5488 + (1,9625/1000)*T_exp5 - (5,9399/1000000)*(T_exp5)^2 "Specific heat capacity of carbohydrates
at 76C"
Cp_water76C = 4,194 "From table at atmospheric pressure"
Cp_milk76C = x_fat*Cp_fat76C + x_protein*Cp_protein76C + x_carbs*Cp_carbs76C + x_water*Cp_water76C "Specific
heat capacity of milk at 76C"

"Specific heat capacity of milk at 8C"
Cp_fat8C = 1,9842 + (1,4733/1000)*T_exp7 - (4,8008/1000000)*(T_exp7)^2 "Specific heat capacity of fat at 8*C"
Cp_protein8C =  2,0082 + (1,2089/1000)*T_exp7 - (1,3129/1000000)*(T_exp7)^2 "Specific heat capacity of protein at 8C"
Cp_carbs8C = 1,5488 + (1,9625/1000)*T_exp7 - (5,9399/1000000)*(T_exp7)^2 "Specific heat capacity of carbohydrates
at 8C"
Cp_water8C = 4,196 "From table at atmospheric pressure"
Cp_milk8C = x_fat*Cp_fat8C + x_protein*Cp_protein8C + x_carbs*Cp_carbs8C + x_water*Cp_water8C "Specific heat
capacity of milk at 8C"
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Cp_preheat = (Cp_milk4C+Cp_milk75C)/2 " Average specific heat capacity of the prehеating process of products"
Cp_DH = (Cp_milk75C+Cp_milk143C)/2 "Average specific heat capacity of the direct heating process of products"
Cp_cooling = (Cp_milk76C+Cp_milk8C)/2 "Average specific heat capacity of the cooling process of products"



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"MODEL OF THE PRODUCT FLOW IN THE EXISTING UHT PLANT"

R$ = 'R718' "Water as refrigerant for the use of water calculations later"

"state of incomming products"

m_dotp = (6240/3600) [kg/s] "Massflow of dairy products in the uht-plant. The density is approx the same as for water"
T_exp1 = 4 [C] "Temperature of products entering the system"

"State after HX1, which is not utilized"
T_exp2= T_exp1
Q_exHX1 = 0

"State after HX2, thus after preheating"
T_exp3 = 75 [C]
Q_exHX2 = m_dotp*Cp_preheat*(T_exp3-T_exp2)

"State after injection, thus in holding tube"
T_exp4 = 143 [C]
Q_exDH = m_dotp*Cp_DH*(T_exp4-T_exp3)

"State after vacuum tower"
T_exp5 = 76 [C]
Q_exVT = m_dotp*Cp_DH*(T_exp5-T_exp4) "The vacuum tower cooling does not count in the energy balance"

"State after HX3, thus first part of cooling through heat exchangers"
T_exp6 = 24,8 [C]
Q_exHX3 = m_dotp*Cp_cooling*(T_exp6-T_exp5)

"State after HX4, thus the end product"
T_exp7 = 8 [C]
Q_exHX4 = m_dotp*Cp_cooling*(T_exp6-T_exp7)




"-------------------------------------------------------------------------------------------------------------------------------------------------------
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"MODELL OF THE ENERGY USE IN EXISTING PLANT"

"Calculation of efficiency for steam production"
E_690V = 1360119 [kwh] "Electricity used per year of 690V cable"
E_el = 1360119*0,7 [kwh] "Approx. yearly use of electricity for the boiler which run on electricity"
E_gas = 7666706 [kwh] "Yearly energy use of gas for the gas boilers."

share_el = E_el/(E_el+E_gas) "Share of steam production done by the boiler which run on electricity."
share_gas = E_gas/(E_el+E_gas) "Share of steam production done by the gas boilers"

eta_el = 0,92 "Efficiency of electricity boiler"
eta_gas = 0,86 "Efficiency of gas boilers"

eta_steam = share_el*eta_el + share_gas*eta_gas "Efficiency of the total steam production"

"Energy use for steam production to direct heating"
P_exDH = 600 [kPa] "The water steam for direct heating is at 6 bar"
x_exDHin = 1 [-] "Assuming saturated vapour at the start of direct heating"
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```
T_exDHout = 143 [C] "Assuming saturated liquid at the end of direct heating"
h_exDHin = enthalpy(R$;P=P_exDH;x=x_exDHin) "Enthalpy of water steam entering direct heating"
h_exDHout = enthalpy(R$;P=P_exDH;T=T_exDHout) "Enthalpy of water steam after direct heating"
m_dots = Q_exDH/(h_exDHin - h_exDHout) "Mass flow rate of steam needed for direct heating"
E_ex[1] = Q_exDH/eta_steam "Simple calculation of energy needed to steam production of steam going to direct heating"

"Calculation of the heat transfer in HX_PH"
T_exHX3in = 19,8 [C] "Temperature of the water entering HX3"
Cp_water20C = 4,182 "Approx specific heat capacity of 19,8C water liquid at atmospheric pressure, collected from table"

T_exHX3out = 70,7 [C] "Temperature of the water exiting HX3"
Cp_water71C = 4,190 "Approx specific heat capacity of 70,7C water liquid at atmospheric pressure, collected from table"

T_exHX2in = 81,1 [C] "Temperature of the water entering HX2"
Cp_water81C = 4,197 "Approx specific heat capacity of 81,1C water liquid at atmospheric pressure, collected from table"

T_exHX2out = 19, 8 [C] "Temperature of the water exiting HX2"
T_exHXPHin = T_exHX3out "Temperature of the water entering HX_PH"
T_exHXPHout = T_exHX2in "Temperature of the water exiting HX_PH"

Cp_HX3 = (Cp_water20C + Cp_water71C)/2 "Average specific heat capacity of the water in heat exchanger HX3"
m_dotHX3*Cp_HX3*(T_exHX3in-T_exHX3out) = Q_exHX3 "Mass flow rate calculation of the water stream flowing
through HX3"

Cp_HX2 = (Cp_water20C + Cp_water81C)/2 "Average specific heat capacity of the water in heat exchanger HX2"
m_dotHX2*Cp_HX2*(T_exHX2in-T_exHX2out) = Q_exHX2 "Mass flow rate calcultation of the water stream flowing
through HX2"

Cp_HXPH = (Cp_water71C + Cp_water81C)/2 "Average specific heat capacity of the water in heat exchanger HX_PH"
Q_exHXPH = m_dotHX2*Cp_HXPH*(T_exHXPHout-T_exHXPHin) "Assuming that the heat exchanger warms up the
mass flow rate of HX2 from 70,7 to 81,1C"

"Energy use for steam production to preheating"
E_ex[2] = Q_exHXPH/eta_steam "Assuming normal heat exchanger. Simple calculation of energy needed to steam
production of the steam going through HX_TW"




"Calculation of energy use in ice water production by ammonia heat pump"

T_IW1 = 2 [C] "Temperature of ice water entering HX4 - collected from product information of whole milk 13%"
Cp_water2C = 4,210 "Specific heat capacity of liquid water at 2C at atmospheric pressure"

T_IW2 = 17 [C] "Temperature of ice water exiting HX4 - collected from product information of whole milk 13%"
Cp_water17C = 4,184 "Specific heat capacity of liquid water at 17C at atmospheric pressure"

Cp_IW = (Cp_water2C + Cp_water17C)/2 "Average specific heat capacity of the water side in HX4"

m_dotIW*Cp_IW*(T_IW2-T_IW1) = Q_exHX4 "Calculation of mass flow rate of ice water going to HX4"

T_IWP1 = 6 [C] "Average yearly temperature of the water entering the ice water tank"
Cp_water6C = 4,200 "Specific heat capacity of liquid water at 6C at atmospheric pressure"

T_IWP2 = 1,3 [C] "Temperature of ice water after ammonia evaporator - Collected from a photo from the factory 26.02."
Cp_water1C = 4,213 "Specific heat capacity of liquid water at 1C at atmospheric pressure"

Cp_IWP = (Cp_water1C + Cp_water6C)/2 "Average specific heat capacity of the ice water tank"
Q_IWP = m_dotIW*Cp_IWP*(T_IWP1-T_IWP2) "Cooling required to produce ice water for the existing uht plant"

T_Hammonia = 21,9 [C] "Upper temperature in the ammonia refrigeration system - Collected from a photo from the
factory 26.02. STILL NOT SURE ABOUT THIS ONE, SHOULD ASK"
T_Lammonia = -12 [C] "Lowe temperatur in the ammonia refrigeration system - Collected from a photo from the factory
26.02"

Carnot_ammonia = (T_Lammonia+273,15)/((T_Hammonia+273,15)-(T_Lammonia+273,15)) "Carnot efficiency of the
ammonia refrigeration systemt"

"Energy use for ice water production to HX4"

E_ex[3] = Q_IWP/(0,5*Carnot_ammonia) "Simple calculation of the energy used in the ammonia heat pump for ice water
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production"

"Energy use for hot tap water production in HX_TW"

T_HXTW1 = 76 [C] "Temperature of water entering the hot water heat exchanger HX_TW - collected from a photo from
the factory 26.02. This value is actually 75.4C, but is sat to 76C for simplicity"
"Cp_water76C is already calculated"

T_HXTW2 = 33,3 [C] "Temperature of water exiting the hot water heat exchanger HX_TW - collected from a photo from
the factory 26.02. This value is actually 32.7C, but is sat to 33,3C for simplicity"
Cp_water33C = 4,178 "Specific heat capacity of liquid water at 33C at atmospheric pressure"

Cp_HXTW = (Cp_water76C + Cp_water33C)/2 "Average specific heat capacity of the water in HX_TW"
Q_HXTW = m_dots*Cp_HXTW*(T_HXTW2-T_HXTW1) "Energy for hot tap water production in HX_TW"
E_ex[4] = Q_HXTW "Assuming no losses in heat exchanger"

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"MODELL OF ALL STREAMS USED IN THE SYSTEMS FOR PINCH ANALYSIS"

DeltaT_min = 5 [C] "Using deltaT_min equal 5 kelvin for all pinch method calculations"

"Cold streams"

"K1"
T_K1in = 4 [C] "Temperature of stream K1 entering the systems"
T_K1out = 75 [C] "Temperature of stream K1 exiting the systems"
T_K1inp = T_K1in + DeltaT_min/2 "Pinch corrected temperature for stream K1 entering the system"
T_K1outp = T_K1out + DeltaT_min/2 "Pinch corrected temperature for stream K1 exiting the system"
C_K1 = m_dotp*Cp_preheat "Heat capacity flow rate of K1"

"K2"
T_K2in = 75 [C] "Temperature of stream K2 entering the systems"
T_K2out = 143 [C] "Temperature of stream K2 exiting the systems"
T_K2inp = T_K2in + DeltaT_min/2 "Pinch corrected temperature for stream K2 entering the system"
T_K2outp = T_K2out + DeltaT_min/2 "Pinch corrected temperature for stream K2 exiting the system"
C_K2 = m_dotp*Cp_DH"Heat capacity flow rate of K2"

"K3"
T_K3in = 19,8 [C] "Temperature of stream K3 entering the systems"
T_K3out = 70,7 [C] "Temperature of stream K3 exiting the systems"
T_K3inp = T_K3in + DeltaT_min/2 "Pinch corrected temperature for stream K3 entering the system"
T_K3outp = T_K3out + DeltaT_min/2 "Pinch corrected temperature for stream K3 exiting the system"
C_K3 = m_dotHX3*Cp_HX3 "Heat capacity flow rate of K3"

"K4 - this stream is necessary to make stream H3"
T_K4in = 70,7 [C] "Temperature of stream K4 entering the systems"
T_K4out = 81,1 [C] "Temperature of stream K4 exiting the systems"
T_K4inp = T_K4in + DeltaT_min/2 "Pinch corrected temperature for stream K4 entering the system"
T_K4outp = T_K4out + DeltaT_min/2 "Pinch corrected temperature for stream K4 exiting the system"
C_K4 = m_dotHX2*Cp_HXPH "Heat capacity flow rate of K4"


"Hot streams"

"H1"
T_H1in = 76 [C] "Temperature of stream H1 entering the systems"
T_H1out = 8 [C] "Temperature of stream H1 exiting the systems"
T_H1inp = T_H1in - DeltaT_min/2 "Pinch corrected temperature for stream H1 entering the system"
T_H1outp = T_H1out - DeltaT_min/2 "Pinch corrected temperature for stream H1 exiting the system"
C_H1 = m_dotp*Cp_cooling  "Heat capacity flow rate of H1"

"H2 in existing system"
T_H2A1in = 76 [C] "Temperature of stream H2 alternative 1 entering the systems"
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T_H2A1out = 33,3 [C] "Temperature of stream H2 alternative 1 exiting the systems"
T_H2A1inp = T_H2A1in - DeltaT_min/2 "Pinch corrected temperature for stream H2 alternative 1 entering the system"
T_H2A1outp = T_H2A1out - DeltaT_min/2 "Pinch corrected temperature for stream H2 alternative 1 exiting the system"
C_H2 = m_dots*Cp_HXTW "Heat capacity flow rate of H2"

"H2 in alternative systems"
T_H2A2in = 90 [C] "Temperature of stream H2 alternative 2 entering the systems"
T_H2A2out = 33,3 [C] "Temperature of stream H2 alternative 2 exiting the systems"
T_H2A2inp = T_H2A2in - DeltaT_min/2 "Pinch corrected temperature for stream H2 alternative 2 entering the system"
T_H2A2outp = T_H2A2out - DeltaT_min/2 "Pinch corrected temperature for stream H2 alternative 2 exiting the system"
"Heat capacity flow is assumed the same for both alternatives for simplicity"

"H3"
T_H3in = 81,1 [C] "Temperature of stream H3 entering the systems"
T_H3out = 19,8 [C] "Temperature of stream H3 exiting the systems"
T_H3inp = T_H3in - DeltaT_min/2 "Pinch corrected temperature for stream H3 entering the system"
T_H3outp = T_H3out - DeltaT_min/2 "Pinch corrected temperature for stream H3 exiting the system"
C_H3 = m_dotHX2*Cp_HX2 "Heat capacity flow of H3"

"H4"
T_H4 = 90 [C] "Temperature of H4, which is constant"
T_H4p = T_H4 - DeltaT_min/2 "Pinch corrected temperature for stream H4"
x_H4in = 1 [-] "Quality of H4 entering the systems"
x_H4out = 0 [-] "Quality of H4 exiting the systems"
h_H4in = Enthalpy(R$;x=x_H4in;T=T_H4) "Enthalpy of H4 entering the systems"
h_H4out = Enthalpy(R$;x=x_H4out;T=T_H4) "Enthalpy of H4 exiting the systems"
Q_H4 = -m_dots*(h_H4in-h_H4out)

"HP - The stream which will be equal the steam produced from the heat pump in system 4 and 5"
T_HPin = T_HP[12] "Temperature of the stream after last compression step in heat pump. Not sure at all if the heat pump
can use its superheat in direct heating"
T_HPinp = T_HPin + DeltaT_min/2 "The minimum temperature difference in heat exchangers does not apply for direct
heating in injector. The superheated temperature out of the last compression step in the heat pump is therefore corrected
as a cold stream."
T_HPout = T_HP[13] "Temperature after condensing the stream going out of heat pump"
T_HPoutp = T_HPout + DeltaT_min/2 "The minimum temperature difference in heat exchangers does not apply for direct
heating in injector. The corrected outlet heat pump temperature is therefore the same as the pinch corrected temperature
in stream K2."
P_HP = P_HP[13] "Pressure of the condensing stream"
x_HPout = 0 [-] "The stream is saturated liquid after condensing"
h_HPin = Enthalpy(R$;T=T_HPin;P=P_HP) "Enthalpy of HP stream entering the system"
h_HPout = Enthalpy(R$;x=x_HPout;T=T_HPout) "Enthalpy of HP stream exiting the system"
m_dotHP = m_dotHP[13] "Mass flow rate of stream HP"
Q_HP = W_dotHP[1] "Energy which could be utilized from latent heat and condensation of HP"



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"MODELL OF PINCH METHOD FOR THE SYSTEM 1"

S1[1] = 0

"Temperature interval method of system 1"

T_intS1[1] = T_K2outp "Upper temperature"
T_intS1[2] = T_K4outp "First interval is between 145,5C og 83,6C"
T_intS1[3] = T_H3inp "Second interval is between 83,6C og 78,6C"
T_intS1[4] = T_K1outp "Third interval is between 78,6C og 77,5C"
T_intS1[5] = T_H1inp "Fourth interval is between 77,5C og 73,5C"
T_intS1[6] = T_K4inp "Fifth interval is between 73,5C and 73,2C"
T_intS1[7] = T_H2A1outp "Sixth interval is between 73,2C og 30,8C"
T_intS1[8] = T_K3inp "Seventh interval is between 30,8C og 22,3C"
T_intS1[9] = T_H3outp "Eighth interval is between 22,3C og 17,3C"
T_intS1[10] = T_K1inp  "Ninth interval is between 17,3C og 6,5C"
T_intS1[11] = T_H1outp "Tenth interval is between 6,5C og 5,5C"

C_intS1[2] = C_K2 "Sum of heat capacity flow rate in first interval"
C_intS1[3] = C_K2+C_K4 "Sum of heat capacity flow rate in second interval"
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C_intS1[4] = C_K2+C_K4-C_H3 "Sum of heat capacity flow rate in third interval"
C_intS1[5] = C_K1+C_K4-C_H3 "Sum of heat capacity flow rate in fourth interval"
C_intS1[6] = C_K1+C_K4-C_H1-C_H2-C_H3 "Sum of heat capacity flow rate in fifth interval"
C_intS1[7] = C_K1+C_K3-C_H1-C_H2-C_H3 "Sum of heat capacity flow rate in sixth interval"
C_intS1[8] = C_K1+C_K3-C_H1-C_H3 "Sum of heat capacity flow rate in seventh interval"
C_intS1[9] = C_K1-C_H1-C_H3 "Sum of heat capacity flow rate in eight interval"
C_intS1[10] = C_K1-C_H1 "Sum of heat capacity flow rate in ninth interval"
C_intS1[11] = -C_H1 "Sum of heat capacity flow rate in tenth interval"

Q_intS1[2] = C_intS1[2]*(T_intS1[1]-T_intS1[2]) "Surplus of heat/cooling in firsth interval"
Q_intS1[3] = C_intS1[3]*(T_intS1[2]-T_intS1[3]) "Surplus of heat/cooling in second interval"
Q_intS1[4] = C_intS1[4]*(T_intS1[3]-T_intS1[4]) "Surplus of heat/cooling in third interval"
Q_intS1[5] = C_intS1[5]*(T_intS1[4]-T_intS1[5]) "Surplus of heat/cooling in fourth interval"
Q_intS1[6] = C_intS1[6]*(T_intS1[5]-T_intS1[6]) "Surplus of heat/cooling in fifth interval"
Q_intS1[7] = C_intS1[7]*(T_intS1[6]-T_intS1[7]) "Surplus of heat/cooling in sixth interval"
Q_intS1[8] = C_intS1[8]*(T_intS1[7]-T_intS1[8]) "Surplus of heat/cooling in seventh interval"
Q_intS1[9] = C_intS1[9]*(T_intS1[8]-T_intS1[9]) "Surplus of heat/cooling in eighth interval"
Q_intS1[10] = C_intS1[10]*(T_intS1[9]-T_intS1[10]) "Surplus of heat/cooling in ninth interval"
Q_intS1[11] = C_intS1[11]*(T_intS1[10]-T_intS1[11]) "Surplus of heat/cooling in tenth interval"

deltaQ_intS1[2] = Q_intS1[2] "Sum of Surplus of heat/cooling in intervall 1"
deltaQ_intS1[3] = deltaQ_intS1[2]+Q_intS1[3] "Sum of Surplus of heat/cooling in intervall 1 and 2"
deltaQ_intS1[4] = deltaQ_intS1[3]+Q_intS1[4] "Sum of Surplus of heat/cooling in intervall 1, 2 and 3"
deltaQ_intS1[5] = deltaQ_intS1[4]+Q_intS1[5] "Sum of Surplus of heat/cooling in intervall 1, 2, 3 and 4"
deltaQ_intS1[6] = deltaQ_intS1[5]+Q_intS1[6] "Sum of Surplus of heat/cooling in intervall 1, 2, 3, 4 and 5"
deltaQ_intS1[7] = deltaQ_intS1[6]+Q_intS1[7] "Sum of Surplus of heat/cooling in intervall 1, 2, 3, 4, 5 and 6"
deltaQ_intS1[8] = deltaQ_intS1[7]+Q_intS1[8] "Sum of Surplus of heat/cooling in intervall 1, 2, 3, 4, 5, 6 and 7"
deltaQ_intS1[9] = deltaQ_intS1[8]+Q_intS1[9] "Sum of Surplus of heat/cooling in intervall 1, 2, 3, 4, 5, 6, 7 and 8"
deltaQ_intS1[10] = deltaQ_intS1[9]+Q_intS1[10] "Sum of Surplus of heat/cooling in intervall 1, 2, 3, 4, 5, 6, 7, 8 and 9"
deltaQ_intS1[11] = deltaQ_intS1[10]+Q_intS1[11] "Sum of surplus of heat/cooling in interval 1, 2, 3, 4, 5, 6, 7, 8, 9 and 10"

"deltaQ_star is used to make the grand composite curve plot"
deltaQ_starS1[1] = deltaQ_intS1[5]
deltaQ_starS1[2] = deltaQ_starS1[1]-Q_intS1[2]
deltaQ_starS1[3] = deltaQ_starS1[2]-Q_intS1[3]
deltaQ_starS1[4] = deltaQ_starS1[3]-Q_intS1[4]
deltaQ_starS1[5] = deltaQ_starS1[4]-Q_intS1[5]
deltaQ_starS1[6] = deltaQ_starS1[5]-Q_intS1[6]
deltaQ_starS1[7] = deltaQ_starS1[6]-Q_intS1[7]
deltaQ_starS1[8] = deltaQ_starS1[7]-Q_intS1[8]
deltaQ_starS1[9] = deltaQ_starS1[8]-Q_intS1[9]
deltaQ_starS1[10] = deltaQ_starS1[9]-Q_intS1[10]
deltaQ_starS1[11] = deltaQ_starS1[10]-Q_intS1[11]



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"Composite curves for system 1, made from the knowledge of the temperature interval method results"

"Cold stream temperatures"
T_CCS1[1] = T_K1in
T_CCS1[2] = T_K3in
T_CCS1[3] = T_K3out
T_CCS1[4] = T_K2in
T_CCS1[5] = T_K4out
T_CCS1[6] = T_K2out

"Hot stream temperatures"
T_CCS1[7] = T_H1out
T_CCS1[8] = T_H3out
T_CCS1[9] = T_H2A1out
T_CCS1[10] = T_H1in
T_CCS1[11] = T_H3in

"Energy points for the cold composite curve"
Q_CCS1[1] = deltaQ_starS1[11]
Q_CCS1[2] = C_K1*(T_CCS1[2]-T_CCS1[1]) + Q_CCS1[1]
Q_CCS1[3] = (C_K1+C_K3)*(T_CCS1[3]-T_CCS1[2]) + Q_CCS1[2]
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Q_CCS1[4] = (C_K1+C_K4)*(T_CCS1[4]-T_CCS1[3]) + Q_CCS1[3]
Q_CCS1[5] = (C_K2+C_K4)*(T_CCS1[5]-T_CCS1[4]) + Q_CCS1[4]
Q_CCS1[6] = C_K2*(T_CCS1[6]-T_CCS1[5]) + Q_CCS1[5]

"Energy points for the hot composite curve"
Q_CCS1[7] = 0
Q_CCS1[8] = C_H1*(T_CCS2[8]-T_CCS2[7]) + Q_CCS2[7]
Q_CCS1[9] = (C_H1+C_H3)*(T_CCS2[9]-T_CCS2[8]) + Q_CCS2[8]
Q_CCS1[10] = (C_H1+C_H3+C_H2)*(T_CCS2[10]-T_CCS2[9]) + Q_CCS2[9]
Q_CCS1[11] = C_H3*(T_CCS2[11]-T_CCS2[10]) + Q_CCS2[10]

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"MODELL OF THE PRODUCT FLOW IN MER SYSTEM FROM THE PINCH METHOD ON SYSTEM 1"

"state of incomming products"

T_1MERS1 = 4 [C] "Temperature of products entering the system"

"State after HX_new3"
T_2MERS1 = 71 [C] "Value collected from figure 6.1 in latex"
Q_MERS1HXnew3 = m_dotp*Cp_preheat*(T_2MERS1-T_1MERS1)

"State after HX2, thus after preheating"
T_3MERS1 = 75 [C]
Q_MERS1HX2 = m_dotp*Cp_preheat*(T_3MERS1-T_2MERS1)

"State after injection, thus in holding tube"
T_4MERS1 = 143 [C]
Q_MERS1DH = m_dotp*Cp_DH*(T_4MERS1-T_3MERS1)

"State after vacuum tower"
T_5MERS1 = 76 [C]
Q_MERS1VT = m_dotp*Cp_DH*(T_5MERS1-T_4MERS1) "The vacuum tower cooling does not count in the energy
balance"

"State after HX_new3"
T_6MERS1 = 9 [C] "Value collected from figure 6.1 in latex"
Q_MERS1HX_new32 = m_dotp*Cp_cooling*(T_6MERS1-T_5MERS1)

"State after HX4, thus the end product"
T_7MERS1 = 8 [C]
Q_MERS1HX4 = m_dotp*Cp_cooling*(T_6MERS1-T_7MERS1)


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"MODELL OF THE ENERGY USE IN THE MER SYSTEM FROM THE PINCH METHOD ON SYSTEM 1"

Q_MERS1HXnew1 = C_H3*(31,53-T_H3out) "Ice water needed to cool down the rest of stream H3, where 31,5 is the
temperature that is going in to the new heat exchanger"

m_dotMERS1IW*Cp_IW*(T_IW1-T_IW2) = Q_MERS1HX4 + Q_MERS1HXnew1 "Calculation of mass flow rate of ice
water. Assuming the inlet and outlet temperature of ice water in the system are the same as existing system"
Q_MERS1IWP = m_dotMERS1IW*Cp_IWP*(T_IWP2-T_IWP1) "Energy required to produce the ice water"

Q_MERS1TW = -C_H2*(T_H2A1in-T_H2A1out) "Hot tap water production in HX_TW in this system"

Q_MERS1PH = C_K4*(T_K4out-72,65) "The energy needed to get the water circuit back to 81,1*C by steam in the PH
heat exchanger, where 72,65C is the temperature the heating starts."

E_MERS1[1] = Q_MERS1DH/eta_steam "Net energy required for steam production to the direct heating"
E_MERS1[2] = Q_MERS1PH/eta_steam "Net energy required for steam production to the preheating og products"
E_MERS1[3] = Q_MERS1IWP/(0,5*Carnot_ammonia) "Net energy required for ice water production to cooling of products"
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E_MERS1[4] = Q_MERS1TW "Net energy released to hot tap water production"

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"MODELL OF THE PRODUCT FLOW IN HEN SYSTEM FROM THE PINCH METHOD ON SYSTEM 1"
"DON'T THINK I NEED TO MAKE THIS, BUT RATHER JUST WRITE THAT THE EXISTING SYSTEM COULD EARN
SOME BY CHANGING THE OUTLET TEMPERATURE OF PH TO 80 RATHER THAN 81,1, BUT IT IS ALSO OK TO DO
THIS TO HAVE SOM SECURITY"

"state of incomming products"

T_1HENS1 = 4 [C] "Temperature of products entering the system"

"State after HX1, which is not utilized"
T_2HENS1 = 5,27 [C] "Value collected from figure 6.2 in latex"
Q_HENS1HX1 = m_dotp*Cp_preheat*(T_2HENS1-T_1HENS1)

"State after HX2, thus after preheating"
T_3HENS1 = 75 [C]
Q_HENS1HX2 = m_dotp*Cp_preheat*(T_3HENS1-T_2HENS1)

"State after injection, thus in holding tube"
T_4HENS1 = 143 [C]
Q_HENS1DH = m_dotp*Cp_DH*(T_4HENS1-T_3HENS1)

"State after vacuum tower"
T_5HENS1 = 76 [C]
Q_HENS1VT = m_dotp*Cp_DH*(T_5HENS1-T_4HENS1) "The vacuum tower cooling does not count in the energy
balance"

"State after HX3, thus first part of cooling through heat exchangers"
T_6HENS1 = 24,8 [C] "Value collected from figure 6.2 in latex"
Q_HENS1HX3 = m_dotp*Cp_cooling*(T_6HENS1-T_5HENS1)

"State after HX4, thus the end product"
T_7HENS1 = 8 [C]
Q_HENS1HX4 = m_dotp*Cp_cooling*(T_6HENS1-T_7HENS1)

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"MODELL OF THE ENERGY USE IN A HEN SYSTEM FROM THE PINCH METHOD ON SYSTEM 1"

"MODELL OF THE ENERGY USE IN A MER SYSTEM FROM THE PINCH METHOD ON SYSTEM 1"

m_dotHENS1IW*Cp_IW*(T_IW1-T_IW2) = Q_HENS1HX4 "Calculation of mass flow rate of ice water. Assuming the inlet
and outlet temperature of ice water in the system are the same as existing system"
Q_HENS1IWP = m_dotHENS1IW*Cp_IWP*(T_IWP2-T_IWP1) "Energy required to produce the ice water"

Q_HENS1PH = C_K4*(80-T_K4in) "Reducing the upper temperature in the water circuit from 81,1C to 80C"

Q_HENS1HXTW = Q_HXTW+Q_HENS1HX1 "The energy transfer in HX_TW in this HEN system is the same as for the
existing system"

E_HENS1[1] = Q_HENS1DH/eta_steam "Net energy required for steam production to the direct heating"
E_HENS1[2] = Q_HENS1PH/eta_steam "Net energy required for steam production to the preheating og products"
E_HENS1[3] = Q_HENS1IWP/(0,5*Carnot_ammonia) "Net energy required for ice water production to cooling of products"
E_HENS1[4] = Q_HENS1HXTW "Net energy used in hot tap water production for this system"
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"MODELL OF PINCH METHOD FOR SYSTEM 2"

S2[1] = 0 "Made just to separate the different systems in the array"

"Temperature interval method of system 2"

T_intS2[1] = T_K2outp "Upper system temperature"
T_intS2[2] = T_H4p "First interval is between 145,5 og 87,5C"
T_intS2[3] = T_H4p "H4 condenses"
T_intS2[4] = T_K4outp "Second interval is between 87,5 og 83,6C"
T_intS2[5] = T_H3inp "Third interval is between 83,6 og 78,6C"
T_intS2[6] = T_K2inp "Fourth interval is between 78,6 og 77,5C"
T_intS2[7] = T_H1inp  "Fifth interval is between 77,5 og 73,5C"
T_intS2[8] = T_K4inp "Sixth interval is between 73,5 and 73,2C"
T_intS2[9] = T_H2A2outp "Sixth interval is between 73,2 og 30,8C"
T_intS2[10] = T_K3inp "Seventh interval is between 30,8 og 22,3C"
T_intS2[11] = T_H3outp "Eighth interval is between 22,3 og 17,3C"
T_intS2[12] = T_K1inp "Ninth interval is between 17,3 og 6,5C"
T_intS2[13] = T_H1outp "Tenth interval is between 6,5 og 5,5C"

C_intS2[2] = C_K2 "Sum of heat capacity flow rate in interval 1."
C_intS2[3] = 0 "Sum of heat capacity flow rate while condensing"
C_intS2[4] = C_K2 - C_H2 "Sum of heat capacity flow rate in interval 2"
C_intS2[5] = C_K2 + C_K4 - C_H2 "Sum of heat capacity flow rate in interval 3"
C_intS2[6] = C_K2 + C_K4 - C_H2 - C_H3 "Sum of heat capacity flow rate in interval 4"
C_intS2[7] = C_K1 + C_K4 - C_H2 - C_H3 "Sum of heat capacity flow rate in interval 5"
C_intS2[8] = C_K1 + C_K4 - C_H1 - C_H2 - C_H3 "Sum of heat capacity flow rate in interval 6"
C_intS2[9] = C_K1 + C_K3 - C_H1 - C_H2 - C_H3 "Sum of heat capacity flow rate in interval 7"
C_intS2[10] = C_K1 + C_K3 - C_H1 - C_H3 "Sum of heat capacity flow rate in interval 8"
C_intS2[11] = C_K1 - C_H1 - C_H3 "Sum of heat capacity flow rate in interval 9"
C_intS2[12] = C_K1 - C_H1 "Sum of heat capacity flow rate in interval 10"
C_intS2[13] = -C_H1

Q_intS2[2] = C_intS2[2]*(T_intS2[1]-T_intS2[2]) "Surplus of heat/cooling in interval 1"
Q_intS2[3] = Q_H4 "Surplus of heat while H4 condenses"
Q_intS2[4] = C_intS2[4]*(T_intS2[3]-T_intS2[4]) "Surplus of heat/cooling in interval 2"
Q_intS2[5] = C_intS2[5]*(T_intS2[4]-T_intS2[5]) "Surplus of heat/cooling in interval 3"
Q_intS2[6] = C_intS2[6]*(T_intS2[5]-T_intS2[6]) "Surplus of heat/cooling in interval 4"
Q_intS2[7] = C_intS2[7]*(T_intS2[6]-T_intS2[7]) "Surplus of heat/cooling in interval 5"
Q_intS2[8] = C_intS2[8]*(T_intS2[7]-T_intS2[8]) "Surplus of heat/cooling in interval 6"
Q_intS2[9] = C_intS2[9]*(T_intS2[8]-T_intS2[9]) "Surplus of heat/cooling in interval 7"
Q_intS2[10] = C_intS2[10]*(T_intS2[9]-T_intS2[10]) "Surplus of heat/cooling in interval 8"
Q_intS2[11] = C_intS2[11]*(T_intS2[10]-T_intS2[11]) "Surplus of heat/cooling in interval 9"
Q_intS2[12] = C_intS2[12]*(T_intS2[11]-T_intS2[12]) "Surplus of heat/cooling in interval 10"
Q_intS2[13] = C_intS2[13]*(T_intS2[12]-T_intS2[13]) "Surplus of heat/cooling in interval 11"

DeltaQ_intS2[2] = Q_intS2[2] "Surplus of heat/cooling in interval 1"
DeltaQ_intS2[3] = DeltaQ_intS2[2]+Q_intS2[3] "Surplus of heat/cooling in interval 1 + condense"
DeltaQ_intS2[4] = DeltaQ_intS2[3]+Q_intS2[4] "Surplus of heat/cooling in interval 1 and 2 + condense"
DeltaQ_intS2[5] = DeltaQ_intS2[4]+Q_intS2[5] "Surplus of heat/cooling in interval 1, 2 and 3 + condense"
DeltaQ_intS2[6] = DeltaQ_intS2[5]+Q_intS2[6] "Surplus of heat/cooling in interval 1, 2, 3 and 4 + condense"
DeltaQ_intS2[7] = DeltaQ_intS2[6]+Q_intS2[7] "Surplus of heat/cooling in interval 1, 2, 3, 4 and 5 + condense"
DeltaQ_intS2[8] = DeltaQ_intS2[7]+Q_intS2[8] "Surplus of heat/cooling in interval 1, 2, 3, 4, 5 and 6 + condense"
DeltaQ_intS2[9] = DeltaQ_intS2[8]+Q_intS2[9] "Surplus of heat/cooling in interval 1, 2, 3, 4, 5, 6 and 7 + condense"
DeltaQ_intS2[10] = DeltaQ_intS2[9]+Q_intS2[10] "Surplus of heat/cooling in interval 1,2, 3, 4, 5, 6, 7 and 8 + condense"
DeltaQ_intS2[11] = DeltaQ_intS2[10]+Q_intS2[11] "Surplus of heat/cooling in interval 1, 2, 3, 4, 5, 6, 7, 8 and 9 +
condense"
DeltaQ_intS2[12] = DeltaQ_intS2[11]+Q_intS2[12] "Surplus of heat/cooling in interval 1, 2, 3, 4, 5, 6, 7, 8, 9 and 10 +
condense"
DeltaQ_intS2[13] = DeltaQ_intS2[12]+Q_intS2[13] "Surplus of heat/cooling in interval 1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 11 +
condense"
```

```
"deltaQ_star is used to make the grand composite curve plot"
DeltaQ_starS2[1] = DeltaQ_intS2[2]
DeltaQ_starS2[2] = DeltaQ_starS2[1] - Q_intS2[2]
DeltaQ_starS2[3] = DeltaQ_starS2[2] - Q_intS2[3]
DeltaQ_starS2[4] = DeltaQ_starS2[3] - Q_intS2[4]
DeltaQ_starS2[5] = DeltaQ_starS2[4] - Q_intS2[5]
DeltaQ_starS2[6] = DeltaQ_starS2[5] - Q_intS2[6]
DeltaQ_starS2[7] = DeltaQ_starS2[6] - Q_intS2[7]
DeltaQ_starS2[8] = DeltaQ_starS2[7] - Q_intS2[8]
DeltaQ_starS2[9] = DeltaQ_starS2[8] - Q_intS2[9]
DeltaQ_starS2[10] = DeltaQ_starS2[9] - Q_intS2[10]
DeltaQ_starS2[11] = DeltaQ_starS2[10] - Q_intS2[11]
DeltaQ_starS2[12] = DeltaQ_starS2[11] - Q_intS2[12]
DeltaQ_starS2[13] = DeltaQ_starS2[12] - Q_intS2[13]

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"Composite curves for system 2, made from the knowledge of the temperature interval method results"

"Cold stream temperatures"
T_CCS2[1] = T_K1in
T_CCS2[2] = T_K3in
T_CCS2[3] = T_K3out
T_CCS2[4] = T_K2in
T_CCS2[5] = T_K4out
T_CCS2[6] = T_K2out

"Hot stream temperatures"
T_CCS2[7] = T_H1out
T_CCS2[8] = T_H3out
T_CCS2[9] = T_H2A2out
T_CCS2[10] = T_H1in
T_CCS2[11] = T_H3in
T_CCS2[12] = T_H2A2in
T_CCS2[13] = T_H4

"Energy points for the cold composite curve"
Q_CCS2[1] = DeltaQ_starS2[13]
Q_CCS2[2] = C_K1*(T_CCS2[2]-T_CCS2[1]) + Q_CCS2[1]
Q_CCS2[3] = (C_K1+C_K3)*(T_CCS2[3]-T_CCS2[2]) + Q_CCS2[2]
Q_CCS2[4] = (C_K1+C_K4)*(T_CCS2[4]-T_CCS2[3]) + Q_CCS2[3]
Q_CCS2[5] = (C_K2+C_K4)*(T_CCS2[5]-T_CCS2[4]) + Q_CCS2[4]
Q_CCS2[6] = C_K2*(T_CCS2[6]-T_CCS2[5]) + Q_CCS2[5]

"Energy points for the hot composite curve"
Q_CCS2[7] = 0
Q_CCS2[8] = C_H1*(T_CCS2[8]-T_CCS2[7]) + Q_CCS2[7]
Q_CCS2[9] = (C_H1+C_H3)*(T_CCS2[9]-T_CCS2[8]) + Q_CCS2[8]
Q_CCS2[10] = (C_H1+C_H3+C_H2)*(T_CCS2[10]-T_CCS2[9]) + Q_CCS2[9]
Q_CCS2[11] = (C_H3+C_H2)*(T_CCS2[11]-T_CCS2[10]) + Q_CCS2[10]
Q_CCS2[12] = C_H2*(T_CCS2[12]-T_CCS2[11]) + Q_CCS2[11]
Q_CCS2[13] = -Q_H4 + Q_CCS2[12]

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"MODELL OF THE PRODUCT FLOW IN HEN SYSTEM FROM THE PINCH METHOD ON SYSTEM 2"

"state of incomming products"

T_1HENS2 = 4 [C] "Temperature of products entering the system"

"State after HX1, which is not utilized"
T_2HENS2 = 4 [C] "Value collected from figure 6.1 in latex"
Q_HENS2HX1 = m_dotp*Cp_preheat*(T_2HENS2-T_1HENS2)
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"State after HX2, thus after preheating"
T_3HENS2 = 75 [C]
Q_HENS2HX2 = m_dotp*Cp_preheat*(T_3HENS2-T_2HENS2)

"State after injection, thus in holding tube"
T_4HENS2 = 143 [C]
Q_HENS2DH = m_dotp*Cp_DH*(T_4HENS2-T_3HENS2)

"State after vacuum tower"
T_5HENS2 = 76 [C]
Q_HENS2VT = m_dotp*Cp_DH*(T_5HENS2-T_4HENS2) "The vacuum tower cooling does not count in the energy balance"

"State after HX3, thus first part of cooling through heat exchangers"
T_6HENS2 = 24,8 [C] "Value collected from figure 6.1 in latex"
Q_HENS2HX3 = m_dotp*Cp_cooling*(T_6HENS2-T_5HENS2)

"State after HX4, thus the end product"
T_7HENS2 = 8 [C]
Q_HENS2HX4 = m_dotp*Cp_cooling*(T_6HENS2-T_7HENS2)

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"MODELL OF THE ENERGY USE IN A HEN SYSTEM FROM THE PINCH METHOD ON SYSTEM 2"

"The energy utilized for refill of mass flow rate between K3 and H3 is dependent on where this refill comes from. If it comes from H4, this would be less than from steam production, since this is done with an efficiency lower than 1"
"MODELLING THE REFILL AS STEAM PRODUCTION. THIS IS SO MUCH EASIER, SINCE THE OTHER SOLUTION AFFECTS THE MASS FLOW RATE OF H2 AND THEREBY CHANGING THE WHOLE SYSTEM"

m_dotHENS2IW*Cp_IW*(T_IW1-T_IW2) = Q_HENS2HX4 "Calculation of mass flow rate of ice water. Assuming the inlet and outlet temperature of ice water in the system are the same as existing system"
Q_HENS2IWP = m_dotHENS2IW*Cp_IWP*(T_IWP2-T_IWP1) "Energy required to produce the ice water"

E_HENS2[1] = Q_HENS2DH/eta_steam "Net energy input in steam production for direct heating"
E_HENS2[2] = 0 "No steam needed to heat the water circuit"
E_HENS2[3] = Q_HENS2IWP/(0,5*Carnot_ammonia)
E_HENS2[4] = -(m_dots*(h_H4in-h_H4out) - C_K4*(T_K4out-T_K4in)) - C_H2*(T_H2A2in-T_H2A2out) "The available heat to for instance hot tap water production is all energy from H4 except from the heat released to K4 and all the energy in H2"

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"MODEL OF THE PINCH METHOD FOR SYSTEM 3"
S3[1] = 0
"Temperaturintevallmetoden. Fortsatt litt usikker på om jeg skal programmere denne slik at den endrer seg om jeg endrer ulike verdier"

T_intS3[1] = T_K2outp "Øvre temperatur"
T_intS3[2] = T_H4p "Første intervall ligger mellom 145,5 og 87,5C"
T_intS3[3] = T_H4p "Kondensering"
T_intS3[4] = T_K2inp "Andre intervall ligger mellom 87,5 og 77,5C"
T_intS3[5] = T_H1inp "Tredje intervall ligger mellom 77,5 og 73,5C"
T_intS3[6] = T_H2A2outp "Fjerde intervall ligger mellom 73,5 og 30,8C"
T_intS3[7] = T_K1inp "Femte intervall ligger mellom 30,8 og 6,5C"
T_intS3[8] = T_H1outp "Sjette intervall ligger mellom 6,5 og 5,5C"

C_intS3[2] = C_K2
C_intS3[3] = 0
C_intS3[4] = C_K2 - C_H2
C_intS3[5] = C_K1 - C_H2
C_intS3[6] = C_K1 - C_H1 - C_H2
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C_intS3[7] = C_K1 - C_H1
C_intS3[8] = - C_H1

Q_intS3[2] = C_intS3[2]*(T_intS3[1]-T_intS3[2])
Q_intS3[3] = Q_H4
Q_intS3[4] = C_intS3[4]*(T_intS3[3]-T_intS3[4])
Q_intS3[5] = C_intS3[5]*(T_intS3[4]-T_intS3[5])
Q_intS3[6] = C_intS3[6]*(T_intS3[5]-T_intS3[6])
Q_intS3[7] = C_intS3[7]*(T_intS3[6]-T_intS3[7])
Q_intS3[8] = C_intS3[8]*(T_intS3[7]-T_intS3[8])

DeltaQ_intS3[2] = Q_intS3[2]
DeltaQ_intS3[3] = DeltaQ_intS3[2]+Q_intS3[3]
DeltaQ_intS3[4] = DeltaQ_intS3[3]+Q_intS3[4]
DeltaQ_intS3[5] = DeltaQ_intS3[4]+Q_intS3[5]
DeltaQ_intS3[6] = DeltaQ_intS3[5]+Q_intS3[6]
DeltaQ_intS3[7] = DeltaQ_intS3[6]+Q_intS3[7]
DeltaQ_intS3[8] = DeltaQ_intS3[7]+Q_intS3[8]

DeltaQ_starS3[1] = DeltaQ_intS3[2]
DeltaQ_starS3[2] = DeltaQ_starS3[1]-Q_intS3[2]
DeltaQ_starS3[3] = DeltaQ_starS3[2]-Q_intS3[3]
DeltaQ_starS3[4] = DeltaQ_starS3[3]-Q_intS3[4]
DeltaQ_starS3[5] = DeltaQ_starS3[4]-Q_intS3[5]
DeltaQ_starS3[6] = DeltaQ_starS3[5]-Q_intS3[6]
DeltaQ_starS3[7] = DeltaQ_starS3[6]-Q_intS3[7]
DeltaQ_starS3[8] = DeltaQ_starS3[7]-Q_intS3[8]




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"Composite curves of system 3 from the temperature interval method results"

"Cold stream temperature interval"

T_CCS3[1] = T_K1in
T_CCS3[2] = T_K1out
T_CCS3[3] = T_K2out

"Hot streams temperature interval"
T_CCS3[4] = T_H1out
T_CCS3[5] = T_H2A2out
T_CCS3[6] = T_H1in
T_CCS3[7] = T_H2A2in
T_CCS3[8] = T_H4

"Energy points for the cold composite curve"
Q_CCS3[1] = DeltaQ_starS3[8]
Q_CCS3[2] = C_K1*(T_CCS3[2]-T_CCS3[1]) + Q_CCS3[1]
Q_CCS3[3] = C_K2*(T_CCS3[3]-T_CCS3[2]) + Q_CCS3[2]

"Energy points for the hot composite curve"
Q_CCS3[4] = 0
Q_CCS3[5] = C_H1*(T_CCS3[5]-T_CCS3[4])
Q_CCS3[6] = (C_H1+C_H2)*(T_CCS3[6]-T_CCS3[5]) + Q_CCS3[5]
Q_CCS3[7] = C_H2*(T_CCS3[7]-T_CCS3[6]) + Q_CCS3[6]
Q_CCS3[8] = -Q_H4 + Q_CCS3[7]



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"MODELL OF THE PRODUCT FLOW IN HEN SYSTEM FROM THE PINCH METHOD ON SYSTEM 3"

"state of incomming products"
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T_1HENS3 = 4 [C] "Temperature of products entering the system"

"State after HX_new"
T_2HENS3 = 71 [C] "Value collected from figure 6.1 in latex"
Q_HENS3HXNEW = m_dotp*Cp_preheat*(T_2HENS3-T_1HENS3)

"State after HX1, thus after preheating"
T_3HENS3 = 75 [C]
Q_HENS3HX1 = m_dotp*Cp_preheat*(T_3HENS3-T_2HENS3)

"State after injection, thus in holding tube"
T_4HENS3 = 143 [C]
Q_HENS3DH = m_dotp*Cp_DH*(T_4HENS3-T_3HENS3)

"State after vacuum tower"
T_5HENS3 = 76 [C]
Q_HENS3VT = m_dotp*Cp_DH*(T_5HENS3-T_4HENS3) "The vacuum tower cooling does not count in the energy balance"

"State after HXnew, thus first part of cooling through heat exchangers"
T_6HENS3 = 9 [C] "Value collected from figure 6.1 in latex"
Q_HENS3HXNEW2 = m_dotp*Cp_cooling*(T_6HENS3-T_5HENS3)

"State after HX4, thus the end product"
T_7HENS3 = 8 [C]
Q_HENS3HX4 = m_dotp*Cp_cooling*(T_6HENS3-T_7HENS3)



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"MODELL OF THE ENERGY USE IN HEN SYSTEM FROM THE PINCH METHOD ON SYSTEM 3"

T_IW2new = 4 [C] "Since the ice water needs to cool the product stream from 9->8C, the ice water can't be heated more than to 4C in the cooling process to obtain the minimum temperature difference in the heat exchanger."
"Cp_water4C is already calculated"
Cp_IWnew = (Cp_water4C + Cp_water2C)/2 "New average specific heat capacity in heat exchanger HX4"

m_dotHENS3IW*Cp_IWnew*(T_IW1-T_IW2new) = Q_HENS3HX4 "Calculation of mass flow rate of ice water. Since the ice water needs to cool the product stream from 9->8C, the ice water can't be heated more than to 4C in the cooling process to obtain the minimum temperature difference in the heat exchanger."
Q_HENS3IWP = m_dotHENS3IW*Cp_IWP*(T_IWP2-T_IWP1) "Energy required to produce the ice water."

Q_HENS3TW = -(m_dots*(h_H4in-h_H4out) - C_K1*(T_K1out-71)) - C_H2*(T_H2A2in-T_H2A2out) "The energy available to for instance hot tap water production from all energy in H4 except from the energy used to heat K1 and all energy from H2"

E_HENS3[1] = Q_HENS3DH/eta_steam "Energy required to produce the steam for direct heating"
E_HENS3[2] = 0 "The water circuit is removed, hence there is no need for steam to this purpose"
E_HENS3[3] = Q_HENS3IWP/(0,5*Carnot_ammonia) "Energy required to produce the ice water"
E_HENS3[4] = Q_HENS3TW

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"HEAT PUMP MODEL FOR SYSTEM 4 AND 5"

HP[1] = 0

"HEAT PUMP (HX AS INTERCOOLING WITH 5K SUPERHEAT) - SINCE THERE IS NO DOWNSTREAM IN THE HEAT PUMP IT MUST BE PROGRAMMED DIFFERENT. I'M LETTING THE FIRST ONE STAND JUST IN CASE"

"The water vapour entering the first compressor step should be a little superheated. Maybe it is possible to use a circuit for intercooling to heat this vapour?"
"Basic states"
T_HPlow = 90 [C]
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T_HPhigh = 159 [C] "Don't know if i should reduce this because of losses or if i should hold it like this and compress
higher to compensate for losses?"
W_dotHP[1] = Q_exDH "Heat output of the heat pump system"
P_HP[4] = 142,37 [kPa] "139,98 [kPa] Calculated from the equation P_2 = sqrt(P_1*P_4)"
P_HP[8] = 289,05 [kPa] "290,32 [KPa] Calculated from the equation P_4 = sqrt(P_2*P_7). This and the latter equation is
an equation system with 2 eq and 2 unknown variables, which EES will not calculate"

"State 1 - temperature is allready given"
x_HPcalc1 = 1 [-]
P_HP[1] = Pressure(R$;T=T_HPlow;x=x_HPcalc1) "This should be 70 kpa"
T_HP[1] = Temperature(R$;P=P_HP[1];x=x_HPcalc1) + 5 "5C superheated when entering first compressor"
h_HP[1] = Enthalpy(R$;T=T_HP[1];P=P_HP[1])
m_dotHP[1] = m_dotHP[13]
rho_HP[1] = Density(R$;T=T_HP[1];P=P_HP[1]) "Not sure if I need this"
V_dotHP[1] = m_dotHP[1]/rho_HP[1]

"State 2 - state 1 with losses in suction pipe"
T_HPcalc2 = T_HPlow-1 "For calculation of the pressure in state 2"
x_HPcalc2 = 1 [-] "For calculation of the pressure in state 2"
P_HP[2] = Pressure(R$;T=T_HPcalc2;x=x_HPcalc2) "Corrected for pressure loss in suction pipe of first compressor"
h_HP[2] = h_HP[1]
T_HP[2] = Temperature(R$;P=P_HP[2];h=h_HP[2])
rho_HP[2] = Density(R$;P=P_HP[2];h=h_HP[2]) "Not sure if i need this"
m_dotHP[2] = m_dotHP[13]
V_dotHP[2] = m_dotHP[2]/rho_HP[2]

"State 3 - after first compression step"
eta_HP = 0,73 "Realistic isentropic efficiency collected from rotrex compressor, which would be used for all compressors
in this system. Probably the picture with volumetric flow should be used in the thesis to argue for this isentropic efficiency"
x_HPcalc3 = 1 [-] "Not real, just used in calculation of discharge losses in first compression step"
T_HPcalc3 = Temperature(R$;P=P_HP[4];x=x_HPcalc3) + 1 "One kelvin pressure loss in discharge losses for
compression step 1"
P_HP[3] = Pressure(R$;x=x_HPcalc3;T=T_HPcalc3) "One kelvin higher compression because of discharge losses"
Call Compressor2_CL(h_HP[2]; P_HP[2]; P_HP[3]; m_dotHP[2]; R$; eta_HP: h_HP[3]; W_dotHP[2]; eta_real1HP)

T_HP[3] = Temperature(R$;h=h_HP[3];P=P_HP[3])
m_dotHP[3] = m_dotHP[13]
rho_HP[3] = Density(R$;h=h_HP[3];P=P_HP[3]) "Not sure if i need this"
V_dotHP[3] = m_dotHP[3]/rho_HP[3]



"State 4"
h_HP[4] = h_HP[3] "The enthalpy is constant under pressure losses"
T_HP[4] = Temperature(R$;P=P_HP[4];h=h_HP[4])
m_dotHP[4] = m_dotHP[13]
rho_HP[4] = Density(R$;P=P_HP[4];T=T_HP[4]) "not sure if i need this"
V_dotHP[4] = m_dotHP[4]/rho_HP[4]

"State 5 - after the first intercooling. Intercooling is here done with a heat exchanger and not with direct heat exchange.
The vapour is 5K superheated out of intercooler HX"
x_HPcalc5 = 1 [-]
P_HP[5] = P_HP[4]
T_HP[5] = Temperature(R$;P=P_HP[5];x=x_HPcalc5) + 5 "5 kelvin superheated out of intercooling HX"
h_HP[5] = Enthalpy(R$;P=P_HP[5];T=T_HP[5])
m_dotHP[5] = m_dotHP[13]
rho_HP[5] = Density(R$;P=P_HP[5];T=T_HP[5]) "not sure if i need this"
V_dotHP[5] = m_dotHP[5]/rho_HP[5]

"State 6 - correcting for losses in suction pipe"
x_HPcalc6 = 1 [-]
T_HPcalc6 = Temperature(R$;x=x_HPcalc6;P=P_HP[5]) - 1 "Pressure loss in suction pipe is equal 1 kelvin"
P_HP[6] = Pressure(R$;x=x_HPcalc6;T=T_HPcalc6) "Corrected for pressure loss in suction pipe of compressor step 2"
h_HP[6] = h_HP[5]
T_HP[6] = Temperature(R$;P=P_HP[6];h=h_HP[6])
m_dotHP[6] = m_dotHP[13]
rho_HP[6] = Density(R$;h=h_HP[6];P=P_HP[6]) "not sure if i need this"
V_dotHP[6] = m_dotHP[6]/rho_HP[6]

"State 7 - after second compression step"
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x_HPcalc7 = 1 [-] "Not real, just used in calculation of discharge losses in second compression step"
T_HPcalc7 = Temperature(R$;P=P_HP[6];x=x_HPcalc7) + 1 "One kelvin pressure loss in discharge losses for
compression step 2"
P_HP[7] = Pressure(R$;x=x_HPcalc7;T=T_HPcalc7) "One kelvin higher compression because of discharge losses"
Call Compressor2_CL(h_HP[6]; P_HP[6]; P_HP[7]; m_dotHP[6]; R$; eta_HP: h_HP[7]; W_dotHP[3]; eta_real2HP)

T_HP[7] = Temperature(R$;P=P_HP[7];h=h_HP[7])
m_dotHP[7] = m_dotHP[13]
rho_HP[7] = Density(R$;P=P_HP[7];h=h_HP[7]) "not sure if i need this"
V_dotHP[7] = m_dotHP[7]/rho_HP[7]

"State 8 - correcting for pressure drop in discharge pipe of compressor step 2"
h_HP[8] = h_HP[7]
T_HP[8] = Temperature(R$;P=P_HP[8];h=h_HP[8])
m_dotHP[8] = m_dotHP[13]
rho_HP[8] = Density(R$;P=P_HP[8];h=h_HP[8]) "not sure if need this"
V_dotHP[8] = m_dotHP[8]/rho_HP[8]

"State 9 - after intercooling in HX between second and third compression step"
P_HP[9] = P_HP[8]
x_HPcalc9 = 1 [-]
T_HP[9] = Temperature(R$;P=P_HP[9];x=x_HPcalc9) + 5 "5kelvin superheated out of intercooler HX"
h_HP[9] = Enthalpy(R$;P=P_HP[9];T=T_HP[9])
m_dotHP[9] = m_dotHP[13]
rho_HP[9] = Density(R$;P=P_HP[9];T=T_HP[9]) "not sure if i need this"
V_dotHP[9] = m_dotHP[9]/rho_HP[9]

"State 10 - correction for pressure drop in suction pipe of compressor step 3"
x_HPcalc10 = 1 [-]
T_HPcalc10 = Temperature(R$;P=P_HP[9];x=x_HPcalc10) - 1 "1 kvelvin pressure drop in suction pipe"
P_HP[10] = Pressure(R$;T=T_HPcalc10;x=x_HPcalc10)
h_HP[10] = h_HP[9] "Enthalpy is constant when pressure drop occurs"
T_HP[10] = Temperature(R$;P=P_HP[10];h=h_HP[10])
m_dotHP[10] = m_dotHP[13]
rho_HP[10] = Density(R$;P=P_HP[10];h=h_HP[10]) "not sure if i need this"
V_dotHP[10] = m_dotHP[10]/rho_HP[10]

"State 11 - after third compression step"
x_HPcalc11 = 1 [-] "Not real, just used in calculation of discharge losses in third compression step"
P_HP[11] = Pressure(R$;x=x_HPcalc11;T=T_HPhigh) "One kelvin higher compression because of discharge losses"
Call Compressor2_CL(h_HP[10]; P_HP[10]; P_HP[11]; m_dotHP[10]; R$; eta_HP: h_HP[11]; W_dotHP[4]; eta_real3HP)

T_HP[11] = Temperature(R$;P=P_HP[11];h=h_HP[11])
m_dotHP[11] = m_dotHP[13]
rho_HP[11] = Density(R$;P=P_HP[11];h=h_HP[11]) "not sure if i need this"
V_dotHP[11] = m_dotHP[11]/rho_HP[11]

"State 12 - correction of discharge losses"
x_HPcalc12 = 1 [-]
T_HPcalc12 = T_HPhigh - 1
P_HP[12] = Pressure(R$;T=T_HPcalc12;x=x_HPcalc12) "Pressure loss in in discharge is 1kelvin"
h_HP[12] = h_HP[11]
T_HP[12] = Temperature(R$;P=P_HP[12];h=h_HP[12])
m_dotHP[12] = m_dotHP[13]
rho_HP[12] = Density(R$;P=P_HP[12];h=h_HP[12]) "not sure if i need this"
V_dotHP[12] = m_dotHP[12]/rho_HP[12]

"State 13 after condensing and subcooling in direct heating"
"x_HP[13] = 0 [-]"
T_HP[13] = 143 [C] "Temperature of products and water after direct heating"
P_HP[13] = P_HP[12]
h_HP[13] = Enthalpy(R$;P=P_HP[13];T=T_HP[13])
"T_HP[13] = Temperature(R$;P=P_HP[13];x=x_HPcalc13)"
"h_HP[13] = Enthalpy(R$;P=P_HP[13];x=x_HPcalc13)"
rho_HP[13] = Density(R$;P=P_HP[13];T=T_HP[13]) "not sure if i need this"
V_dotHP[13] = m_dotHP[13]/rho_HP[13]

m_dotHP[13]*(h_HP[12] - h_HP[13]) = W_dotHP[1] "Calculation of the mass flow rate in the whole system"
```

```
m_dotHPcheck = m_dotHP[13]*3600 "Check if the mass flow rate is the same as the product information say, most likely
less since superheat is included in the condensing process"
W_dotHP = W_dotHP[2] + W_dotHP[3] + W_dotHP[4] "Compressor power of all compressors combined"
COP_HP = W_dotHP[1]/W_dotHP



"---------------------------------------------------------------------------------------------------------------------------------------------
-----------------------------------------------------------------"

"PINCH METHOD FOR SYSTEM 4"
S4[1] = 0
"Temperature interval method"
"At this time I don't see the reason of calculating the specific heat capacity of the latent heat of the superheat from the
heat pump. I'm just putting everything in the kondensation"
T_intS4[1] = T_HPinp "Highest temperature is 245.69C, which is the superheated outlet temperature of the heat pump
when it is corrected as a cold stream."
T_intS4[2] = T_K2outp "First interval is between 245.69 and 145,5C"
T_intS4[3] = T_K4outp "Second interval is between 145,5 and 83,6C"
T_intS4[4] = T_H3inp "Third interval is between 83,6 and 78,6C"
T_intS4[5] = T_K1outp "Fourth interval is between 78,6 and 77,5C"
T_intS4[6] = T_H1inp "Fifth interval is between 77,5 and 73,5C"
T_intS4[7] = T_K4inp "Sixth interval is between 73,5 and 73,2C"
T_intS4[8] = T_K3inp "Sixth interval is between 73,2 and 22,3C"
T_intS4[9] = T_H3outp "Seventh interval is between 22,3 and 17,3C"
T_intS4[10] = T_K1inp "Eighth interval is between 17,3 and 6,5C"
T_intS4[11] = T_H1outp "Ninth interval is between 6,5 and 5,5C"


C_intS4[2] = 0 "Condensation"
C_intS4[3] = C_K2 "Sum of heat capacity flow rate in interval 1"
C_intS4[4] = C_K2 + C_K4 "Sum of heat capacity flow rate in interval 2"
C_intS4[5] = C_K2 + C_K4 - C_H3 "Sum of heat capacity flow rate in interval 3"
C_intS4[6] = C_K1 + C_K4 - C_H3 "Sum of heat capacity flow rate in interval 4"
C_intS4[7] = C_K1 + C_K4 - C_H1 - C_H3 "Sum of heat capacity flow rate in interval 5"
C_intS4[8] = C_K1 + C_K3 - C_H1 - C_H3 "Sum of heat capacity flow rate in interval 6"
C_intS4[9] = C_K1 - C_H1 - C_H3 "Sum of heat capacity flow rate in interval 7"
C_intS4[10] = C_K1 - C_H1 "Sum of heat capacity flow rate in interval 8"
C_intS4[11] = -C_H1 "Sum of heat capacity flow rate in interval 9"




Q_intS4[2] = -Q_HP "Surplus heat from condensation"
Q_intS4[3] = C_intS4[3]*(T_intS4[2]-T_intS4[3]) "Surplus heat/cooling in interval 1"
Q_intS4[4] = C_intS4[4]*(T_intS4[3]-T_intS4[4]) "Surplus heat/cooling in interval 2"
Q_intS4[5] = C_intS4[5]*(T_intS4[4]-T_intS4[5]) "Surplus heat/cooling in interval 3"
Q_intS4[6] = C_intS4[6]*(T_intS4[5]-T_intS4[6]) "Surplus heat/cooling in interval 4"
Q_intS4[7] = C_intS4[7]*(T_intS4[6]-T_intS4[7]) "Surplus heat/cooling in interval 5"
Q_intS4[8] = C_intS4[8]*(T_intS4[7]-T_intS4[8]) "Surplus heat/cooling in interval 6"
Q_intS4[9] = C_intS4[9]*(T_intS4[8]-T_intS4[9]) "Surplus heat/cooling in interval 7"
Q_intS4[10] = C_intS4[10]*(T_intS4[9]-T_intS4[10]) "Surplus heat/cooling in interval 8"
Q_intS4[11] = C_intS4[11]*(T_intS4[10]-T_intS4[11]) "Surplus heat/cooling in interval 9"

DeltaQ_intS4[2] = Q_intS4[2] "Surplus heat from condensation"
DeltaQ_intS4[3] = DeltaQ_intS4[2]+Q_intS4[3] "Surplus heat/cooling from condensation + interval 1"
DeltaQ_intS4[4] = DeltaQ_intS4[3]+Q_intS4[4] "Surplus heat/cooling from condensation + interval 1 and 2"
DeltaQ_intS4[5] = DeltaQ_intS4[4]+Q_intS4[5] "Surplus heat/cooling from condensation + interval 1, 2 and 3"
DeltaQ_intS4[6] = DeltaQ_intS4[5]+Q_intS4[6] "Surplus heat/cooling from condensation + interval 1, 2, 3 and 4"
DeltaQ_intS4[7] = DeltaQ_intS4[6]+Q_intS4[7] "Surplus heat/cooling from condensation + interval 1, 2, 3, 4 and 5"
DeltaQ_intS4[8] = DeltaQ_intS4[7]+Q_intS4[8] "Surplus heat/cooling from condensation + interval 1, 2, 3, 4, 5 and 6"
DeltaQ_intS4[9] = DeltaQ_intS4[8]+Q_intS4[9] "Surplus heat/cooling from condensation + interval 1, 2, 3, 4, 5, 6 and 7"
DeltaQ_intS4[10] = DeltaQ_intS4[9]+Q_intS4[10] "Surplus heat/cooling from condensation + interval 1, 2, 3, 4, 5, 6, 7 and
8"
DeltaQ_intS4[11] = DeltaQ_intS4[10]+Q_intS4[11] "Surplus heat/cooling from condensation + interval 1, 2, 3, 4, 5, 6, 7, 8
and 9"


"Heat/cool energy values for grand composite curve of system 4"
DeltaQ_starS4[1] = DeltaQ_intS4[7]
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```
DeltaQ_starS4[2] = DeltaQ_starS4[1]-Q_intS4[2]
DeltaQ_starS4[3] = DeltaQ_starS4[2]-Q_intS4[3]
DeltaQ_starS4[4] = DeltaQ_starS4[3]-Q_intS4[4]
DeltaQ_starS4[5] = DeltaQ_starS4[4]-Q_intS4[5]
DeltaQ_starS4[6] = DeltaQ_starS4[5]-Q_intS4[6]
DeltaQ_starS4[7] = DeltaQ_starS4[6]-Q_intS4[7]
DeltaQ_starS4[8] = DeltaQ_starS4[7]-Q_intS4[8]
DeltaQ_starS4[9] = DeltaQ_starS4[8]-Q_intS4[9]
DeltaQ_starS4[10] = DeltaQ_starS4[9]-Q_intS4[10]
DeltaQ_starS4[11] = DeltaQ_starS4[10]-Q_intS4[11]

"------------------------------------------------------------------------------------------------------------------------------------------------------------
--------------------------------------------------------------------------------"
"Composite curves for system 4, made from the knowledge of the temperature interval method results"

"Cold stream temperatures"
T_CCS4[1] = T_K1in
T_CCS4[2] = T_K3in
T_CCS4[3] = T_K3out
T_CCS4[4] = T_K2in
T_CCS4[5] = T_K4out
T_CCS4[6] = T_K2out

"Hot stream temperatures"
T_CCS4[7] = T_H1out
T_CCS4[8] = T_H3out
T_CCS4[9] = T_H1in
T_CCS4[10] = T_H3in
T_CCS4[11] = T_HPout
T_CCS4[12] = T_HPin

"Energy points for the cold composite curve"
Q_CCS4[1] = DeltaQ_starS4[11]
Q_CCS4[2] = C_K1*(T_CCS4[2]-T_CCS4[1]) + Q_CCS4[1]
Q_CCS4[3] = (C_K1+C_K3)*(T_CCS4[3]-T_CCS4[2]) + Q_CCS4[2]
Q_CCS4[4] = (C_K1+C_K4)*(T_CCS4[4]-T_CCS4[3]) + Q_CCS4[3]
Q_CCS4[5] = (C_K2+C_K4)*(T_CCS4[5]-T_CCS4[4]) + Q_CCS4[4]
Q_CCS4[6] = C_K2*(T_CCS4[6]-T_CCS4[5]) + Q_CCS4[5]

"Energy points for the hot composite curve"
Q_CCS4[7] = 0
Q_CCS4[8] = C_H1*(T_CCS4[8]-T_CCS4[7]) + Q_CCS4[7]
Q_CCS4[9] = (C_H1+C_H3)*(T_CCS4[9]-T_CCS4[8]) + Q_CCS4[8]
Q_CCS4[10] = C_H3*(T_CCS4[10]-T_CCS4[9]) + Q_CCS4[9]
Q_CCS4[11] =  Q_CCS4[10]
Q_CCS4[12] = Q_HP + Q_CCS4[11]




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"MODELL OF THE PRODUCT FLOW IN HEN SYSTEM FROM THE PINCH METHOD ON SYSTEM 4"

"state of incomming products"

T_1HENS4 = 4 [C] "Temperature of products entering the system"

"State after HX2, thus after preheating"
T_2HENS4 = 75 [C]
Q_HENS4HX2 = m_dotp*Cp_preheat*(T_2HENS4-T_1HENS4)

"State after injection, thus in holding tube"
T_3HENS4 = 143 [C]
Q_HENS4DH = m_dotp*Cp_DH*(T_3HENS4-T_2HENS4)

"State after vacuum tower"
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T_4HENS4 = 76 [C]
Q_HENS4VT = m_dotp*Cp_preheat*(T_4HENS4-T_3HENS4) "The vacuum tower cooling does not count in the energy
balance"

"State after HX3, thus first part of cooling through heat exchangers"
T_5HENS4 = 24,8 [C] "Value collected from figure 6.1 in latex"
Q_HENS4HX3 = m_dotp*Cp_preheat*(T_4HENS4-T_5HENS4)

"State after HX4, thus the end product"
T_6HENS4 = 8 [C]
Q_HENS4HX4 = m_dotp*Cp_preheat*(T_5HENS4-T_6HENS4)


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"MODELL OF THE ENERGY USE IN HEN SYSTEM FROM THE PINCH METHOD ON SYSTEM 4"

m_dotHENS4IW*Cp_IW*(T_IW1-T_IW2) = Q_HENS4HX4 "Calculation of mass flow rate of ice water. Assuming the inlet
and outlet temperature of ice water in the system are the same as existing system"
Q_HENS4IWP = m_dotHENS4IW*Cp_IWP*(T_IWP2-T_IWP1) "Energy required to produce the ice water"

Q_HENS4HXPH = C_K4*(T_K4out-T_K4in)

E_HENS4[1] = Q_HENS4DH/COP_HP "Energy required to produce the steam for direct heating by the heat pump"
E_HENS4[2] = Q_HENS4HXPH/eta_steam "Energy required to produce the steam for the water circuit"
E_HENS4[3] = Q_HENS4IWP/(0,5*Carnot_ammonia) "Energy required to produce the ice water"
E_HENS4[4] = 0 "There is no hot tap water production in this system"




"-----------------------------------------------------------------------------------------------------------------------------------------------------
--------------------------------------------------------------------------------"
"PINCH METHOD FOR SYSTEM 5"
S5[1] = 0

"Temperature interval method of system 5"

T_intS5[1] = T_HPinp "Highest temperature is 245.69C, which is the superheated outlet temperature of the heat pump
when it is corrected as a cold stream."
T_intS5[2] = T_K2outp "First interval is between 245.69 and 145,5C"
T_intS5[3] = T_K1outp "Second interval is between 145,5C and 77,5C"
T_intS5[4] = T_H1inp "Third interval is between 77,5 and 73,5C"
T_intS5[5] = T_K1inp "Fourth interval is between 73,5 and 6,5C"
T_intS5[6] = T_H1outp "Fifth interval is between 6,5 and 5,5C"

C_intS5[2] = 0 "Heat pump interval 1"
C_intS5[3] = C_K2 "Sum of heat capacity flow rate in interval 2"
C_intS5[4] = C_K1 "Sum of heat capacity flow rate in interval 3"
C_intS5[5] = C_K1 - C_H1 "Sum of heat capacity flow rate in interval 4"
C_intS5[6] = -C_H1 "Sum of heat capacity flow rate in interval 5"

Q_intS5[2] = -Q_HP "Surplus energy from heat pump in interval 1"
Q_intS5[3] = C_intS5[3]*(T_intS5[2]-T_intS5[3]) "Surplus heat/cooling from interval 2"
Q_intS5[4] = C_intS5[4]*(T_intS5[3]-T_intS5[4]) "Surplus heat/cooling from interval 3"
Q_intS5[5] = C_intS5[5]*(T_intS5[4]-T_intS5[5]) "Surplus heat/cooling from interval 4"
Q_intS5[6] = C_intS5[6]*(T_intS5[5]-T_intS5[6]) "Surplus heat/cooling from interval 5"


DeltaQ_intS5[2] = Q_intS5[2] "Surplus heat/cooling from interval 1"
DeltaQ_intS5[3] = DeltaQ_intS5[2]+Q_intS5[3] "Surplus heat/cooling from interval 1 and 2"
DeltaQ_intS5[4] = DeltaQ_intS5[3]+Q_intS5[4] "Surplus heat/cooling from interval 1, 2 and 3"
DeltaQ_intS5[5] = DeltaQ_intS5[4]+Q_intS5[5] "Surplus heat/cooling from interval 1, 2, 3 and 4"
DeltaQ_intS5[6] = DeltaQ_intS5[5]+Q_intS5[6] "Surplus heat/cooling from interval 1, 2, 3, 4 and 5"

"Plots for grand composite curve of system 5"
DeltaQ_starS5[1] = DeltaQ_intS5[5]
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DeltaQ_starS5[2] = DeltaQ_starS5[1]-Q_intS5[2]
DeltaQ_starS5[3] = DeltaQ_starS5[2]-Q_intS5[3]
DeltaQ_starS5[4] = DeltaQ_starS5[3]-Q_intS5[4]
DeltaQ_starS5[5] = DeltaQ_starS5[4]-Q_intS5[5]
DeltaQ_starS5[6] = DeltaQ_starS5[5]-Q_intS5[6]

"-----------------------------------------------------------------------------------------------------------------------------------------------------
-------------------------------------------------------------------------------"
"Composite curves for system 4, made from the knowledge of the temperature interval method results"

"Cold stream temperatures"
T_CCS5[1] = T_K1in
T_CCS5[2] = T_K2in
T_CCS5[3] = T_K2out

"Hot stream temperatures"
T_CCS5[4] = T_H1out
T_CCS5[5] = T_H1in
T_CCS5[6] = T_HPout
T_CCS5[7] = T_HPin

"Energy points for the cold composite curve"
Q_CCS5[1] = DeltaQ_starS5[6]
Q_CCS5[2] = C_K1*(T_CCS5[2]-T_CCS5[1]) + Q_CCS5[1]
Q_CCS5[3] = C_K2*(T_CCS5[3]-T_CCS5[2]) + Q_CCS5[2]


"Energy points for the hot composite curve"
Q_CCS5[4] = 0
Q_CCS5[5] = C_H1*(T_CCS5[5]-T_CCS5[4]) + Q_CCS5[4]
Q_CCS5[6] =  Q_CCS5[5]
Q_CCS5[7] = Q_HP + Q_CCS5[6]




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"MODELL OF THE PRODUCT FLOW IN MER SYSTEM FROM THE PINCH METHOD ON SYSTEM 5"

"state of incomming products"

T_1MERS5 = 4 [C] "Temperature of products entering the system"

"State after HXnew1, thus after first part of preheating"
T_2MERS5 = 71 [C]
Q_MERS5HXNEW1 = m_dotp*Cp_preheat*(T_2MERS5-T_1MERS5)

"State after HXnew2, thus after preheating"
T_3MERS5 = 75 [C]
Q_MERS5HXNEW2 = m_dotp*Cp_preheat*(T_3MERS5-T_2MERS5)

"State after injection, thus in holding tube"
T_4MERS5 = 143 [C]
Q_MERS5DH = m_dotp*Cp_DH*(T_4MERS5-T_3MERS5)

"State after vacuum tower"
T_5MERS5 = 76 [C]
Q_MERS5VT = m_dotp*Cp_DH*(T_5MERS5-T_4MERS5) "The vacuum tower cooling does not count in the energy
balance"

"State after HXnew1, thus first part of cooling through heat exchangers"
T_6MERS5 = 9 [C] "Value collected from figure 6.1 in latex"
Q_MERS5HX3 = m_dotp*Cp_cooling*(T_5MERS5-T_6MERS5)

"State after HX4, thus the end product"
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T_7MERS5 = 8 [C]
Q_MERS5HX4 = m_dotp*Cp_cooling*(T_6MERS5-T_7MERS5)

"-----------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------"

"MODELL OF THE ENERGY USE IN HEN SYSTEM FROM THE PINCH METHOD ON SYSTEM 5"

m_dotMERS5IW*Cp_IWnew*(T_IW1-T_IW2new) = Q_MERS5HX4 "Calculation of mass flow rate of ice water. Assuming the outlet of ice water cannot exceed the deltaTmin requirement of 5 kelvin temperature difference. This means that the ice water temperature out of the system is only 4C instead of 17C."
Q_MERS5IWP = m_dotMERS5IW*Cp_IWP*(T_IWP2-T_IWP1) "Energy required to produce the ice water"


E_MERS5[1] = Q_MERS5DH/COP_HP "Energy required to produce the steam for direct heating by the heat pump"
E_MERS5[2] = Q_MERS5HXNEW2/eta_steam "Energy required to produce the steam for the preheating in HXnew2"
E_MERS5[3] = Q_MERS5IWP/(0,5*Carnot_ammonia) "Energy required to produce the ice water"
E_MERS5[4] = 0 "There is no hot tap water production in this system"
```