INVESTIGATION OF WASTE HEAT SOURCES AND THEIR UTILIZATION IN PROCESS INDUSTRY

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ABSTRACT

Waste heat is a commonly used term for such heat energy that is not utilized in the process but dissipates to the environment unused. Waste heat is a by-product which can be found in various streams exiting the process, and it can be categorized according to its temperature into three categories: temperature below 50 °C, temperature between 50…100 °C and temperature above 100 °C.

Utilization of waste heat is a means to improve the energy efficiency of a process or a plant. Actions towards improving energy efficiency are incorporated to strategies and legislation on the national and international level as actions to increase sustainability and reduce CO_2 and other emissions are crucial in ensuring a functional and viable society and environment in the future.

The theoretical, technical and economic potentials of waste heat sources must be evaluated when its utilization is contemplated. Theoretical potential describes the physical properties of the waste heat source and its possible limitations, technical potential determines the technologies available for heat recovery, and economic potential evaluates the investment's economic feasibility.

The heating and cooling demands as well as minimal utility demands for a process can systematically be analysed with Pinch Analysis. This analysis data can be utilized to determine the potentials for heat recovery through process integration as well as to evaluate the how energy efficiently the process is currently being operated.

According to the Laws of Thermodynamics, heat energy is interconvertible and energy will spontaneously flow between substances from a higher temperature to a lower one. These principled must be taken into account when the potentials for waste heat recovery are identified and suitable techniques for heat recovery are contemplated. Heat transfer between the waste heat source and a heat sink can be carried out in the same temperature with passive or by increasing the temperature or transforming heat energy to electric energy with active means. Passive means include heat transfer through heat exchangers and storage of heat. Active means include, for example, heat pumps and Organic Rankine Cycle.

A case study for the process of Finnfeeds Finland Co Naantali factory was carried out in this survey, in order to identify and evaluate the waste heat sources available in their operation and to contemplate on potentials for waste heat utilization. The process was analysed with Pinch Analysis and based on this analysis, there is a limited potential for waste heat recovery without increasing the temperature as the temperature of most of the identified waste heat sources remains relatively low. Several preliminary investment proposals were sketched to demonstrate the potentials for waste heat utilization both internally and externally.

Keywords: energy efficiency, heat exchanger, heat pump, heat storage, heat transfer, Organic Rankine Cycle, Pinch Analysis, waste heat, waste heat recovery

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Sari Valtonen
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ABBREVIATIONS

CC	Composite Curve
CIP	Cleaning in place
CO_2	Carbon dioxide
Co	Company
EED	Energy Efficiency Directive
EIA	Electrical, instrumentation, and automation
EU	the European Union
GCC	Grand Composite Curve
GDP	Gross domestic product
HEN	Heat exchanger network
IEA	International Energy Agency
IFF	International Flavors and Fragrances, Inc.
MVR	Mechanical vapour recompression
ORC	Organic Rankine Cycle
PI(D)	Piping and instrumentation (diagram)
RED2	Directive for Renewable Energy
VAT	Value added tax
UN	the United Nations

ΔT_{min}	Pinch point
c_p	Specific heat capacity
COP	Coefficient of Performance
dT / ΔT	Temperature change
EER	Energy Efficiency Ratio
H	Enthalpy
L	Latent heat capacity
m	Mass or mass flow ($\dot{m}$)
η	Carnot efficiency
Q	Heat load or heat flow ($\dot{Q}$)
S	Shifted temperature
T	Temperature
v	Volume or volume flow ($\dot{v}$)
W	Mechanical energy

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1 INTRODUCTION

In 2019, the Finnish industry sector used 520 PJ (appr. 144 TWh) worth of energy (Tilastokeskus, 2020). It has been estimated that inside the industry sector, the technical potential of energy found in waste heat sources is from 6 to 23 TWh annually, most of which can be found in the chemistry and metal industry sectors. This is a very significant amount of heat energy that could be utilized in many applications (Gynther *et al.*, 2019).

Every time mechanical work is executed with some equipment or in some process, it is inevitable that the primary energy that is fed into the machine cannot fully be utilized. This means that energy losses are formed, for example, via radiation of heat or as convection of heat to a process fluid. Figure 1 illustrates the way energy is utilized in a process and how the unused energy dissipates from it. Even though it is not the purpose of the process to create heat losses, these can act as sources for so called waste heat and contain a very considerable amount of usable heat energy (Brückner *et al.*, 2015).

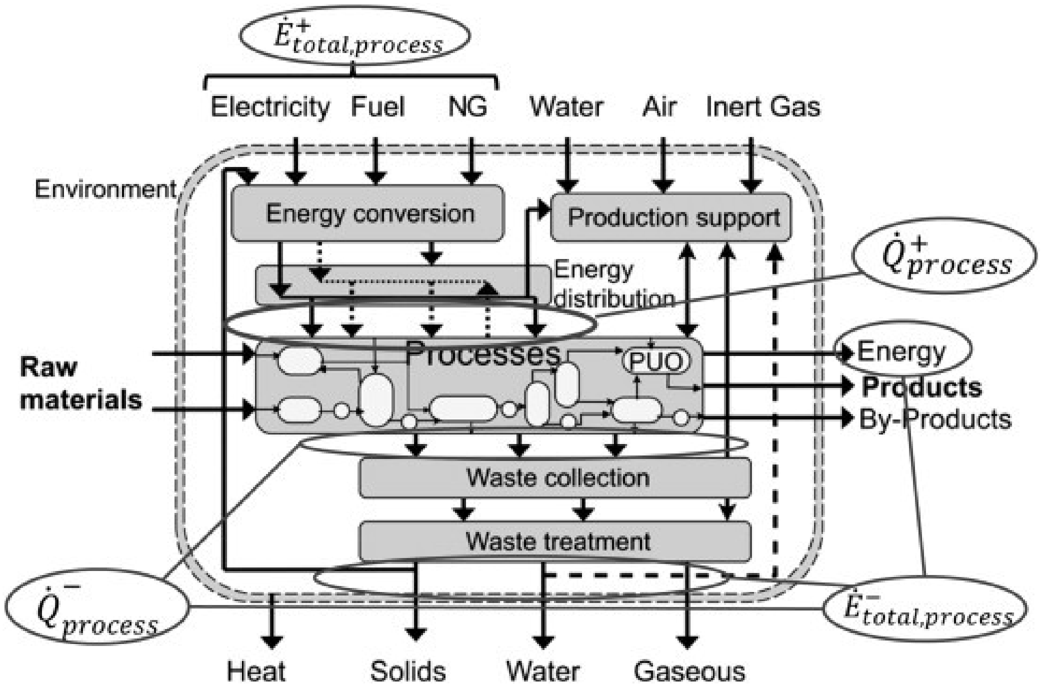

Figure 1 Illustration of energy flowing in a process (Bendig *et al.*, 2013).

Waste heat is a commonly used term with multiple definitions. Most literature references and legislative parties review waste heat simply as heat energy that is dissipated to the surroundings. This definition is solely based on the examination of energy balances and the first law of thermodynamics (the Law of Conservation of Energy; “the total energy of a system and its surroundings is preserved” (Perry *et al.*, 1997a)) and does not take into consideration neither the temperature nor the possibilities of utilization for this energy source (Bendig *et al.*, 2013).

A distinctive feature in the waste heat sources found in process industry is the fact that the heat that is bound in these waste streams is generated as a side product, and its production is dependent on how the process is controlled and in which stage the process currently is. For example, the process can require a certain temperature to which the process fluids are heated up into, or the heat energy can be introduced to the process by exothermal reactions or mechanical work. If this heat energy is not recovered after processing, the temperature will slowly decrease via dissipation of heat energy to the environment or it will leave the process bound into some waste stream (Rämä *et al.*, 2020, Brückner *et al.*, 2015).

Waste heat can be bound into different types of streams leaving the industrial facility, such as waste waters, cooling waters, exhaust steams, process and flue gases, exhaust gases from driers or in condensates from mechanical cooling. These waste heat sources can be categorized according to their temperature levels into the following categories (Gynther *et al.*, 2019):

- Temperature below 50 °C
    - for example, cooling waters, condensate from mechanical cooling, exhaust air flows from processes
- Temperature between 50 °C and 100 °C
    - for example, cooling waters, relief gases, cooling of oil lubricated compressors
- Temperature above 100 °C
    - for example, flue gases, hot exhaust gases from processes

Utilization of these waste heat sources inside the facility in its processes or for heating up the properties, or by selling the heat energy outside of the facility to external actors is highly beneficial in terms of improving the total energy efficiency of the plant. According to IEA Scenario 450, over 50% of reduction of CO_2 emissions in the atmosphere is accomplished with actions related to energy efficiency. This topic has been widely researched, and there is also a great deal of political and economic parties that are pressurising industries to improve their actions regarding energy efficiency, which speaks for the importance of energy efficiency in our society and industry (Bendig *et al.*, 2013).

A means of improving energy efficiency, that is to increase the value of used energy per unit, is to utilize waste heat energy to produce additional actions (Bendig *et al.*, 2013). Ensuring energy efficient operation provides good premises for a competitive, sustainable and environmentally friendly production process, which are all basic principles for a modern industry. Energy efficiency can be improved by implementing new energy efficient technologies or equipment, by modifying existing ones and implementing system integration to reduce waste heat generation, or by implementing techniques or equipment for recovery and reutilization of waste heat. All of these means to improve energy efficiency will require effective techniques to be developed through scientific and technological progress and will naturally require possibly large capital investments to be carried out before implementation (Semkov *et al.*, 2014).

The energy content in waste streams is, unfortunately, usually not fully available for utilization and, thus, when planning to utilize heat energy collected from a waste stream, its technical and economic potential needs to be evaluated in order to determine how much of the theoretical potential actually could be utilized. The theoretical potential of waste heat describes the physical properties and limitations of the source: for efficient utilization, the heat energy should be bound into a medium in a temperature above ambient temperature. Due to such limitations, the heat energy that is radiating from hot equipment surfaces to the surroundings is traditionally poorly available to be utilized, despite its large quantities. The technical potential determines how well the heat energy can be recovered, what technology should be used in the recovery and also where the recovered heat energy can be utilized. In addition to the theoretical and technical potential, an economic potential should also be determined for each waste heat source in order to evaluate the economic feasibility of the investment (Brückner *et al.*, 2015).

At least the following properties should be studied from each waste heat source when its utilization is contemplated (Gynther *et al.*, 2019, Motiva, 2014):

- Temperature level
- Power and energy (stability and fluctuation)
- Medium of waste heat stream and its state
- Chemical properties of the medium
- Purity of the medium
- Demand of heat energy
- How heat recovery will affect the main process

When evaluating the possible ways to utilize waste heat, possible applications can be evaluated in a recommended order, as demonstrated in Figure 2. Following this hierarchical order, the first step is to evaluate possible improvements in energy utilization and efficiency internally, before consideration of distributing the heat energy to external parties. By decreasing the consumption of primary energy, the energy costs of the facility naturally decrease, as do the emissions of the plant. In case waste heat energy is sold to external parties, in addition to additional revenue, new business opportunities are created (Gynther *et al.*, 2019).

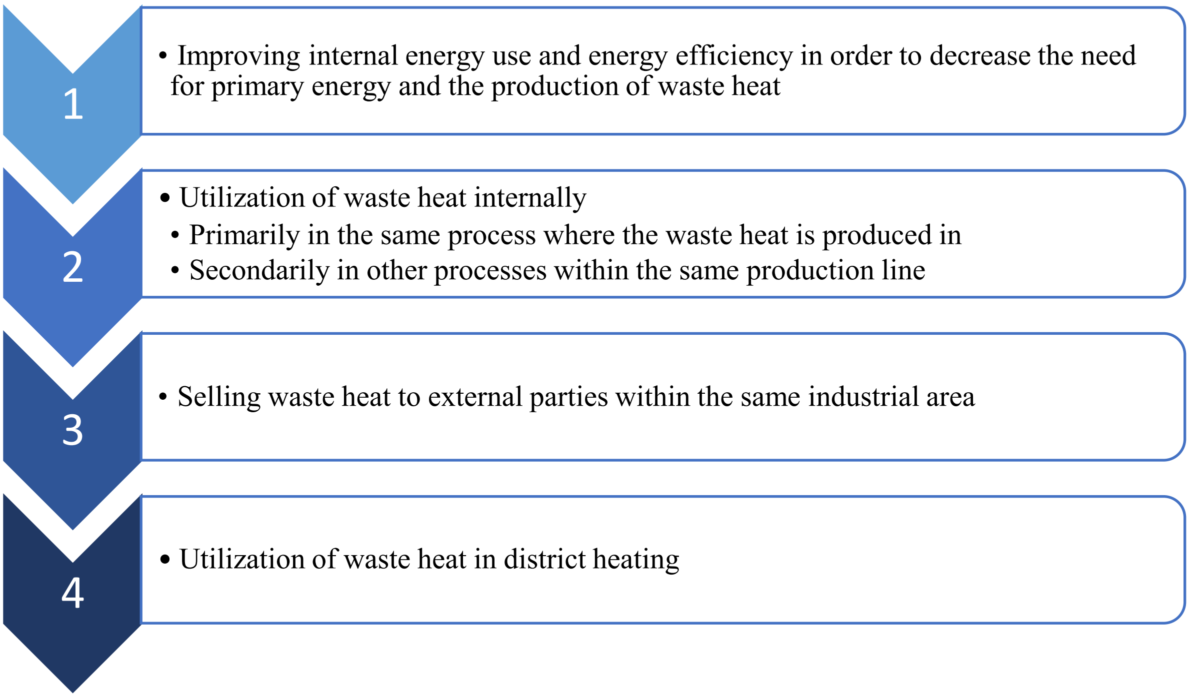

Figure 2 Recommended order to evaluation of waste heat utilization (Gynther *et al.*, 2019).

Waste heat energy can be utilized as such by recirculating it as secondary heat energy to a new utilization point, in heat exchangers to, for example, dry or pre-heat the raw materials or fuels of processes. It can also be used in heating systems of internal properties. If waste heat is utilized in district heating, it is often necessary to increase the temperature of the stream with a heat pump. Increase of waste heat stream temperature can naturally be a requirement in internal usage as heat source for processes as well (Motiva, 2014). Some common technologies that are applied to heat recovery and possible temperature increase of the waste heat are presented in Chapter 2.

1.1 ENERGY EFFICIENCY AND WASTE HEAT IN LEGISLATION AND POLITICS

All companies will have to comply on national and international legislation; for example, in Finland, the national legislation is enacted with reference to that of the European Union. In addition to legislation and directives, these international stakeholders guide their member countries with strategies and visions. For example, the United Nations has published an agenda with multiple goals to ensure sustainable development in 2015, and according to Goal 7.3 in this Agenda, the global rate of improvement in energy efficiency shall be doubled by 2030 (the United Nations, 2015).

On the legislative side, the EU's Energy Efficiency Directive (EED) and Directive for Renewable Energy (RED2) regulate the energy efficiency actions and utilization of renewable energy sources inside the European Union. Contrary to the definition for waste heat given in the introduction in this thesis, RED2 defines waste heat or waste cold only as such waste heat that is utilized in a district heating or cooling network. According to this definition, the industrial plant producing the waste heat has already executed all possible energy efficiency actions to minimize the production of waste heat. Additionally, this definition emphasizes the characteristics of waste heat being a by-product of the process, meaning that it is not a target for the process or plant to produce waste heat (Rämä *et al.*, 2020).

The Energy Efficiency Directive obliges all EU member states to set an indicative energy efficiency target level based on either the consumption of primary or final energy, the savings of primary or final energy, or energy intensity. This target level is reported to the union in a report including additionally the information on the absolute level of primary and final energy use with its calculation principles and the remaining cost-effective energy saving potential, GDP development with forecasts, changes in energy imports and exports, developments in renewable energy sources, nuclear power, carbon capture and storage, and early actions. EU member states shall increase their energy efficiency actions annually by 0.8% of final energy consumption (1.1.2021-31.12.2030) (Directive 2012/27/EU).

Another way for the European Union to guide its member states towards cleaner and more efficient energy solutions is the European Green Deal strategy. The purpose of this strategy is to overcome the challenges caused to the world by climate change and environmental degradation by transforming the EU into a modern economy with resource efficiency and competitiveness with no net emissions of greenhouse gases (by 2050), where economic growth would not be dependent on resource use, and where no persons or places would be left behind. In other words, the European Union is aiming towards a sustainable economy, and this vision will be achieved with a multifaceted plan focusing on the necessary investments and available financial tools for harnessing circular economy principles, restoring biodiversity, and decreasing pollution. The way to reach the targets set by the Green Deal differs for various sectors in the economy, but practical examples include decarbonization of the energy sector and implementing environmentally-friendly technologies to the industry sector (the European Commission, 2021).

Besides decarbonization of the energy sector, other modifications are additionally required in order to modify the current energy systems from their linear operation model towards a circular one, as is illustrated in Figure 3. Utilization of waste heat energy is one of the three main characteristics of the reform of energy systems into an Integrated EU Energy System, alongside making it cleaner in terms of fuel and power. Reduction of currently wasted resources is needed, and this applies also to the resource of heat energy in the form of waste heat. Primarily, the formation of waste heat energy must be reduced by promoting energy efficiency, and secondly, the reuse of waste heat utilization in the industry should be encouraged (the European Commission, 2020).

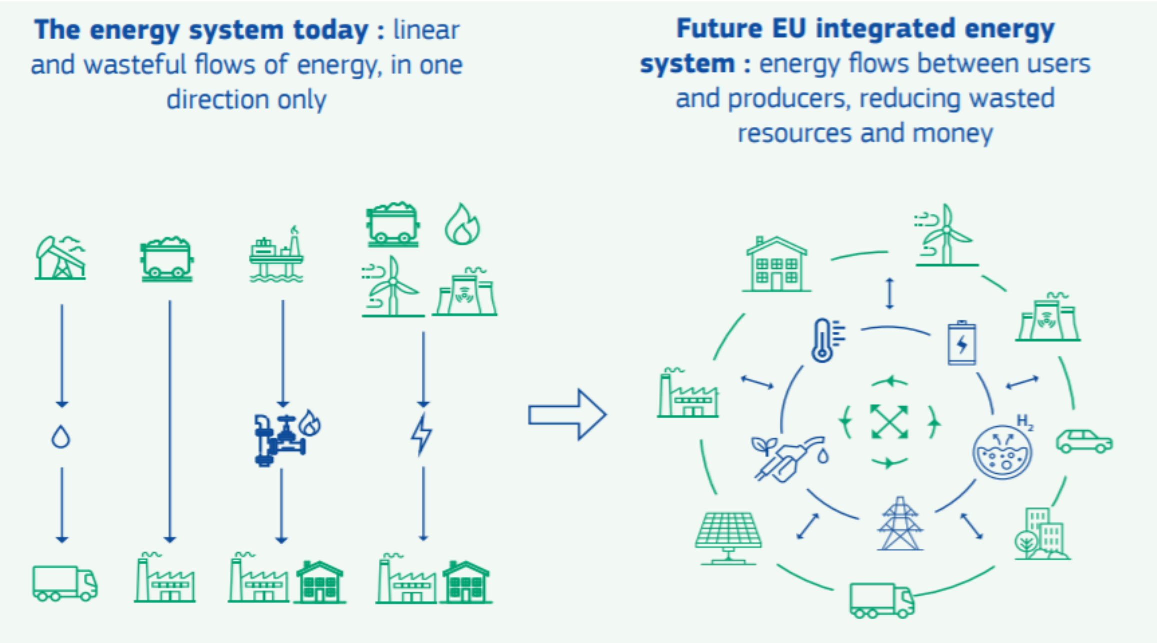

Figure 3 EU Green Deal Energy Integration principles for transforming energy systems into a more sustainable direction (the European Commission, 2020).

At the national level, these energy efficiency actions are steered to the industrial level with legislation and through various benefits or constraints. The Finnish Energy Efficiency Act (1429/2014) contains directions for actions concerning energy efficiency and for investigating the utilization of waste heat in district heating under certain circumstances (Energiatehokkuuslaki (1429/2014)), which means that waste heat is defined similarly in Finnish legislation as in the EU's RED2.

Taxation is an efficient tool for the government to guide the private sector towards the government's goal for a carbon neutral Finland by 2035 (Government Communications Department, 2020). For example, the taxes on electricity are divided into two categories, the second one of which is meant for companies in certain industries, including the manufacturing industry. Mainly companies obliged to pay electricity taxes are included in the first tax category, and these companies include the companies who are producing and distributing electricity (Vero, 2021).

The tax prices for each tax category are the following, including 24% VAT and strategic stockpile fee (Caruna, 2021):

- Tax category 1: 2.79372 c/kWh
- Tax category 2: 0.07812 c/kWh

The manufacturing industry will pay the electricity tax according to the second tax category for all electricity consumption in their production and its support functions, and also in the cases of internal heat production in an electricity-consuming heating plant, as long as the heat is consumed internally. If the company is producing electricity, no taxes or strategic stockpile fee will need to be paid for it, regardless of whether the electricity is consumed internally or sold to external parties (Vero, 2021).

To encourage the manufacturing industry to invest in technologies which utilize electricity (from renewable sources) instead of fossil fuels (Government Communications Department, 2020), the Finnish government has decided to lower the tax category 2 in steps to the EU's minimal level of 0.05 c/kWh (Parkkonen *et al.*, 2020). This decrease in the tax price is conducted in steps as of 2021, and it is also expected to help improve the competitiveness of Finnish industrial companies among their EU rivals (Government Communications Department, 2020). In addition to the decrease of tax category 2, the Government has also decided to move industrial heat pumps to this tax category as of 2021, but unfortunately, this currently applies only to such heat pumps that are producing heat for district heating (the European Commission, 2021).

As energy efficiency and the utilization of waste heat are highly promoted in both international and national strategies and legislation, there are also ways to apply for funding to aid companies fulfil their energy efficiency targets with new investments. For example, Business Finland, the Finnish government's organization for funding of innovations (Business Finland, 2021b), is one stakeholder that grants aids for investments in the form of "Energy Aid" to companies that are carrying out energy efficiency or energy savings investments, as long as these investments are not executed because of fulfilling compulsory environmental obligations or they are not compulsory energy audits stated in the Energy Efficiency Act (1429/2014) (Business Finland, 2021a).

1.2 METHODOLOGIES FOR ANALYSING WASTE HEAT UTILIZATION

In order to find the opportunities for improving cost-efficiently the energy efficiency of a process through the utilization of waste heat, systematic evaluation is required and the tools or methodologies that can be utilized for this purpose can be, for example, System Integration, Heat Exchanger Networks (HEN), Pinch Analysis, and Exergy Analysis (Semkov *et al.*, 2014). Out of these options, Pinch Analysis has been applied in this thesis for a case study survey.

1.2.1 PINCH ANALYSIS

Regarding engineering of chemical processes, the two types of engineering problems are unit operation design and total system design. Pinch Analysis is a systematic tool that can be used to address the total system design problem in order to create and optimize the process flowsheet and simultaneously minimize energy consumption. It was created in the 1970's in the ETH Zurich and Leeds University and has been developed ever since with rapid development in the early years with combined operation of ICI company-wide research and application studies. It has been widely used both in projects designing new plants and in projects modifying existing plant structures, in traditional chemistry industry such as oil refining or bulk chemical production, and also outside of these common examples. Pinch Analysis can be used as a tool for projects tackling problems with heat recovery, process condition changes, operability improvements, or effective use of utility systems. It is a key contributor to process development and design overall strategy, or process synthesis, which can be described with an onion diagram in Figure 4 (Kemp, 2007a).

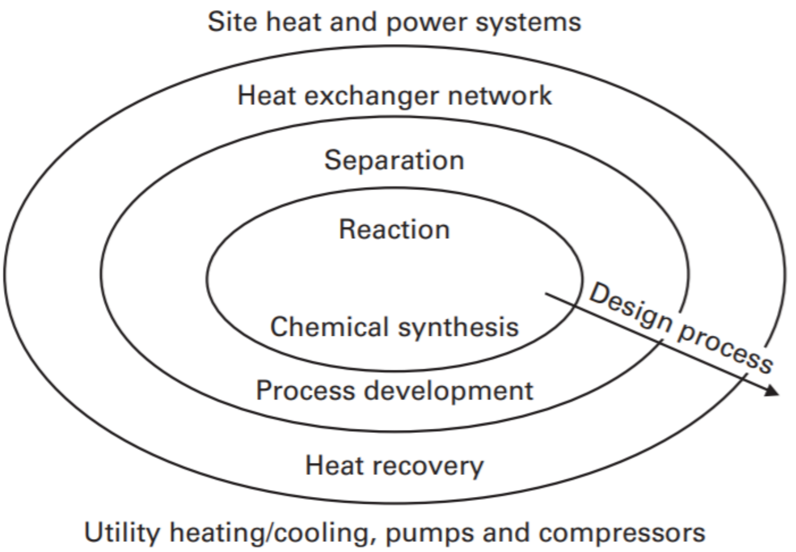

Figure 4 Onion diagram describing the hierarchical nature of process synthesis (Kemp, 2007a).

A design process has to proceed "inside out", starting from the chemical reaction occurring, determining how reaction products can be separated and only then is it possible to determine what kind of heat energy exchange is required in the process. The studied process entity or unit operation is simplified into a flow scheme focusing only on what kind of heating or cooling duties are required in the various streams. The analysis uses the basic principles of thermodynamics in a practical way to study eventually the whole process operation, while the potentials of energy saving or other process benefits are uncovered from raw data (Kemp, 2007a).

Pinch Analysis begins by recognizing and choosing all streams in need of temperature change without a simultaneous composition change, and dividing these into the ones that require cooling and the ones that require heating. Their initial and final temperatures are determined, as well as the heat flow or enthalpy (heat content Q [kW]) in each stream as a function of each substance's specific heat capacity and mass flowrate. This necessary information can be visualized as a temperature-heat content diagram, as shown in Figure 6 (Kemp, 2007b).

This diagram visualizes the change in the enthalpy of the stream when temperature changes from supply temperature T_s to target temperature T_t:

$$Q = \int_{T_S}^{T_T} c_p dT = c_p(T_T - T_S) = \Delta H \qquad (1)$$

$$\frac{dT}{dQ} = \frac{1}{c_p}, \tag{2}$$

where Q = heat flow [kW], c_p = specific heat capacity [kJ/kgK], dT = change in temperature [K], and ΔH = change in enthalpy [kW] (Kemp, 2007b).

As the change in enthalpy is the driving factor in this assessment instead of the absolute value of enthalpy, each stream can be drawn anywhere on the enthalpy axis as long as the slope as well as supply and target temperatures remain the same. The potentials for heat exchange can be interpreted directly from this figure, keeping in mind the fact that heat energy will be transferred between the streams as long as the difference in temperatures remains positive (hot stream temperature > cold stream temperature). This type of visualization can be utilized when simultaneously assessing all relevant process streams (Kemp, 2007b).

To visualize all hot or cold streams with two separate lines, the heat contents and target temperatures of all of them must be taken into account. Based on how many of the streams are viable within a given temperature range, the total heat content in that range is calculated by adding together all the heat contents of each individual stream. This procedure is illustrated in Figure 5, where (a) describes each individual hot stream and its temperature ranges, and (b) illustrates the temperature range of a Composite Curve and how the enthalpy changes throughout the largest and smallest temperature (Kemp, 2007b).

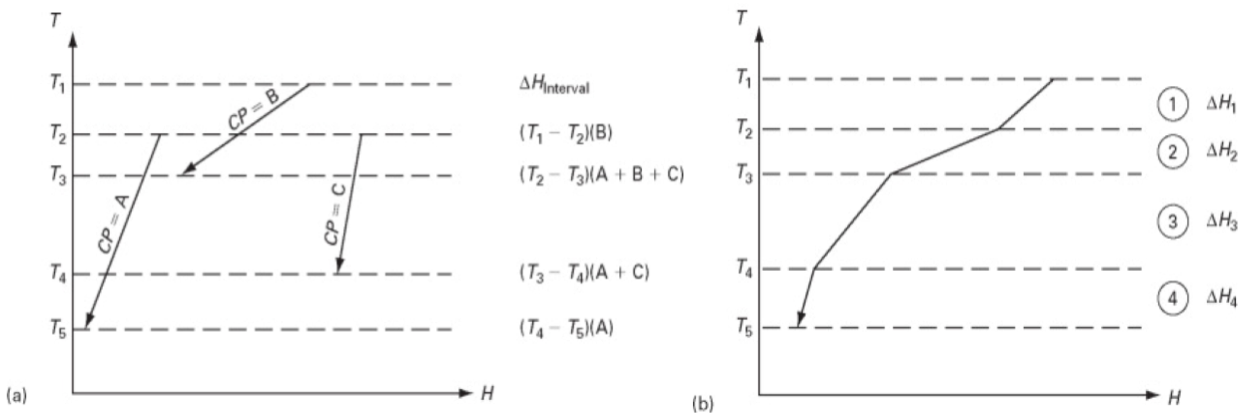

Figure 5 Illustration of how a Composite Curve is formed (Kemp, 2007b).

When the Composite Curves of both hot and cold streams are depicted in the same temperature-heat content curve, as shown in Figure 6, it is possible to determine a Pinch point and the potentials for heat exchange between these streams as well as minimum required amounts of utilities. The overlap of the hot and the cold Composite Curve illustrates how much heat energy could be recovered from the process, and the parts of the Composite Curves exceeding or falling under this overlap represent the minimum amount of required cooling or heating utilities, respectively. To be able to minimize the utilization of utilities for cooling or heating of process streams, the Pinch point or ΔT_{min} between the hot and the cold Composite Curve must be determined so that the two curves will not overlap and the value for ΔT_{min} remains positive and larger than zero at all times. The utility utilization of the process can be optimized to a minimal level based on the information provided by Pinch Analysis (Kemp, 2007b).

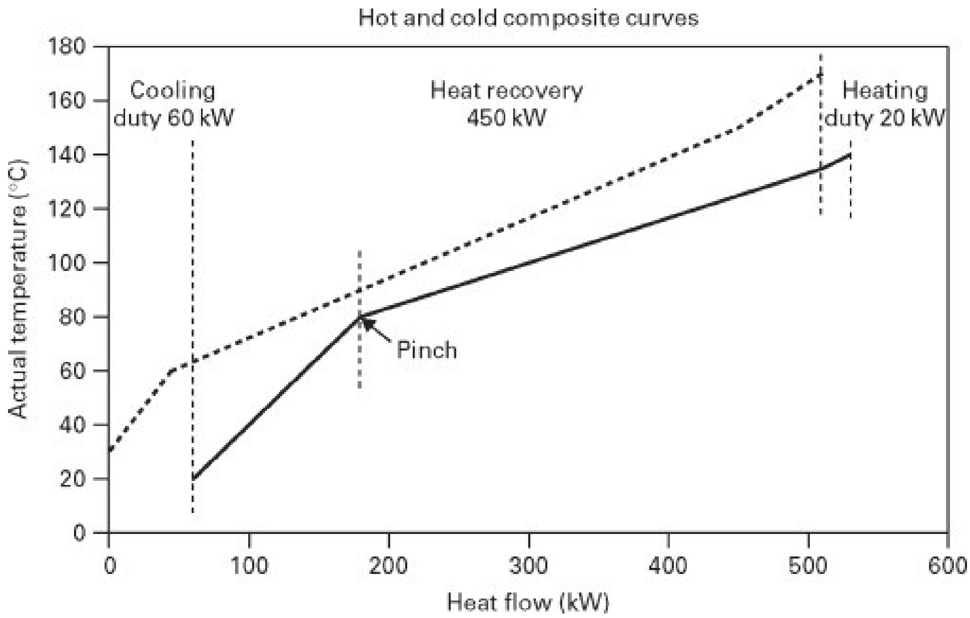

Figure 6 An example of a temperature-heat content diagram for multiple streams. The dotted line contains the hot streams and the continuous line contains the cold streams. The potentials for heat recovery as well as minimum utility needs can be read from the figure (Kemp, 2007b).

In order to set energy targets from the Composite Curves, the two curves must be illustrated in relation to one another with so called shifted temperatures. In all temperature intervals, the temperatures are modified or shifted $\frac{1}{2}\Delta T_{min}$ below or above the original temperatures, depending on whether the stream is a hot or a cold stream. This process is illustrated in Figure 7 (Kemp, 2007b).

For any temperature interval i, there will be a net surplus or deficit of heat energy based on the enthalpy balance (Kemp, 2007b):

$$\Delta H_i = (S_i - S_{i+1})(\sum CP_H - \sum CP_C)_i, \quad (3)$$

where S = shifted temperature in [°C], and $CP = \dot{m}$ [kg/s] · c_p [kJ/kgK].

After calculating the enthalpy balances for each temperature interval, according to Ian Kemp in his book "Pinch Analysis and Process Integration" (Kemp, 2007b) it is assumed that "*any heat available in interval* i *is hot enough to supply any duty in interval* i+1". If the heat is not utilized in a certain interval, or there is a surplus of heat from a certain interval, it can be cascaded into the next interval and utilized there. By studying the enthalpy balances and heat utilization, or how heat cascades through the system, the minimum heat added by cold and hot utilities can be determined. This study operation can also be utilized to determine the total amount of heat recovered by heat exchange by adding together the heat loads for the hot and cold streams and subtracting the hot and cold utility target heat values from them (Kemp, 2007b).

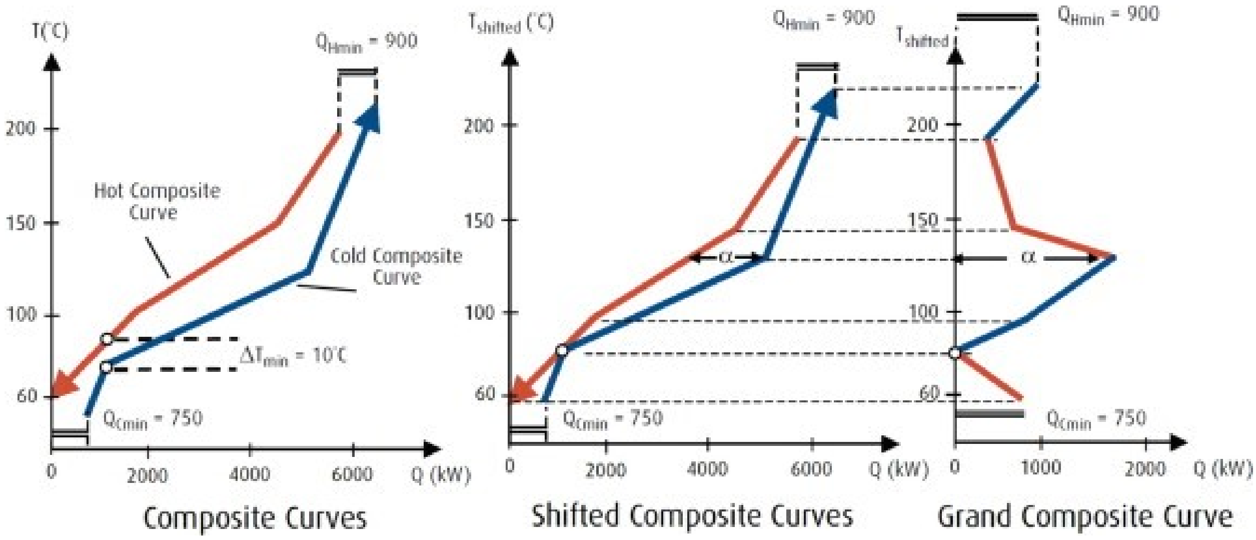

Figure 7 Modifying a Composite Curve into a Shifted Composite Curve and further into a Grand Composite Curve. The Grand Composite Curve illustrates the net heat flow in any given shifted temperature in relation to the Pinch point (Grip *et al.*, 2013).

The Composite Curves can be re-plotted with the shifted temperatures to find out a minimum amount for net heat inflow in any given temperature interval, and this can be plotted against the shifted temperatures to form the Grand Composite Curves as shown in Figure 7. The GCC can be utilized to visualize how much heating or cooling is required at each temperature: in the Pinch point, this net heat load is equal to zero (Kemp, 2007b).

The Pinch point can be utilized with three main rules, when the required heating and cooling energies are designed for a system:

- Heat should not be transferred across the Pinch point
- Cold utilities should not be utilized above the Pinch point
- Hot utilities should not be utilized below the Pinch point (Kemp, 2007b).

Above the Pinch point, all the streams requiring cooling should be cooled through process integration with the streams that require heating, and vice versa for the streams below the Pinch point. If these rules are not followed, it is not possible to accomplish a heat exchange system with minimal external energy from hot and cold utilities: for example, when a cold utility is utilized above the Pinch point, this results in utilizing additional hot utilities also (Kemp, 2007b).

2 THE TECHNICAL APPLICATIONS FOR HEAT RECOVERY

When identifying suitable sinks for waste heat utilization, the first and second laws of thermodynamics must be kept in mind. The first law of thermodynamics states that the internal energy of a system is conserved, and that energy in the form of work or heat are interconvertible. The second law of thermodynamics states that heat will always flow from hot to cold temperature (Luscombe, 2018). These are the same principles that are applied with other types of applications or unit operations involved in heat transfer, and finding pairs of flows with suitable heat flow as well as initial and target temperatures is the target when examining the potentials of waste heat recovery.

The heat flow provided by the waste heat source or required by the heat sink can be calculated with the equation (Abedin *et al.*, 2011):

$$\dot{Q} = \dot{m} c_p \Delta T, \quad (4)$$

where $\dot{Q}$ = heat flow [kW], $\dot{m}$ = mass flow [kg/s], c_p = specific heat capacity [kJ/kgK], and ΔT = change in temperature [K].

There are multiple technologies available for heat recovery of waste heat, and they can be categorized into passive and active means of heat recovery. Passive technologies utilize the heat energy in the same or decreased temperature as in the heat source, whereas active technologies increase the temperature level or convert heat energy into a different form of energy, such as electrical energy. Figure 8 presents different commonly used heat recovery technologies, from which by far the most popular technologies in addition to traditional heat exchangers are different heat pumps and Organic Rankine Cycle (ORC), which is used to convert heat energy into electric energy (Brückner *et al.*, 2015).

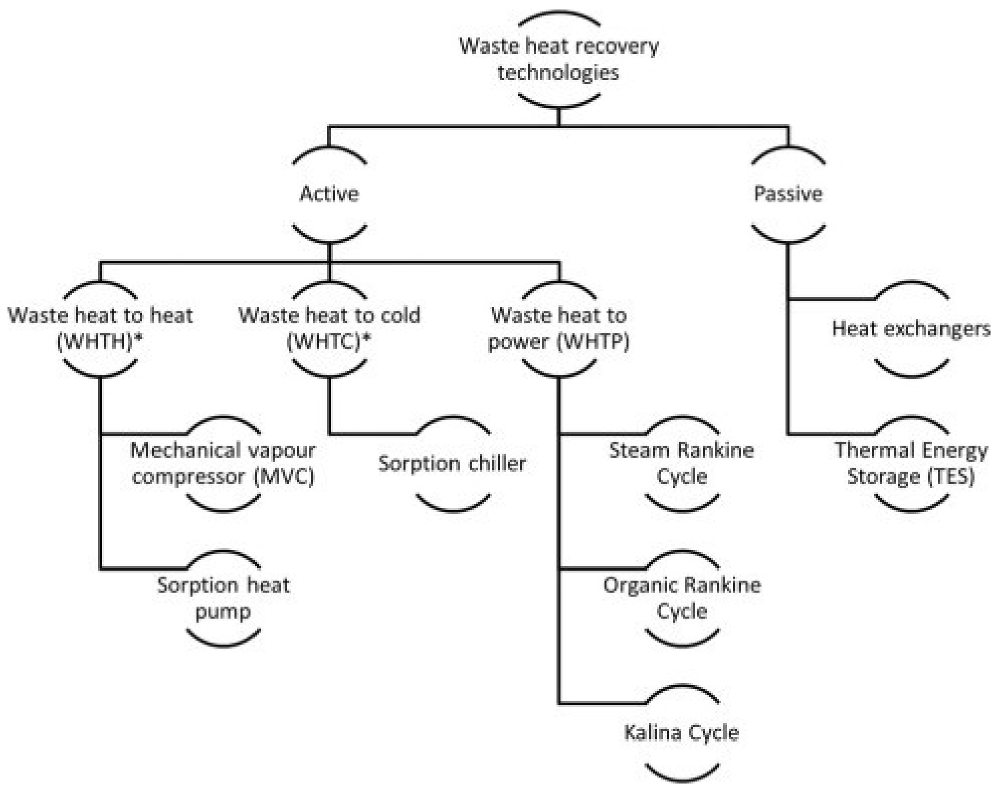

Figure 8 Heat recovery techniques categorized into active and passive categories (Brückner *et al.*, 2015).

The temperature level and quantity of the waste stream containing the waste heat energy play an immensely large role when a suitable technique for heat recovery is chosen. If the waste heat source is in a low temperature level, the quantity of the flow it is bound into needs to be sufficiently large and the utilization application in which the heat energy will be utilized, will need to be simultaneously operational, as heat energy in low temperature is not reasonable to store. The largest benefits and economic savings are provided by high temperature level waste heat sources, and a broader range of technologies for heat recovery is applicable with them (Woolnough, 2011).

Table 1 presents and compares some ordinarily used heat recovery techniques with guideline waste heat source temperatures T_S as well as their advantages and disadvantages.

Table 1 Comparison of different ordinary heat recovery techniques (adapted from Gynther *et al.*, 2019)

Technology	T_s	Advantages	Disadvantages
Utilization directly in heating (e.g, property heating)	appr. 40…100 °C	+ Typically low investment costs + Simple to implement and operate	- The sink and source for heat energy do not necessarily meet
Drying / Evaporation / Pre-heating	appr. 40…450 °C	+ Typically low investment costs + Possible to utilize waste heat energy locally for example in drying or pre-heating of fuel or raw material in order to improve its quality and the efficiency of the process + Pre-heating of water or air is usually a continuous sink for heat energy	- Drying of raw material or product may develop emissions - When designing a heat sink in drying, the whole entity of the process facility needs to be considered: for example, if a heat plant has a flue gas heat recovery system, it is not rational to dry the fuel
Mechanical heat pumps	<10…100 °C	+ Reliable technology with a sound commercial market + A means to utilize low temperature waste heat energy efficiently + Relatively quick to start up + Good qualities even with partial load	- Requires a sufficiently large and stable waste heat source and sink - Economic profitability may be weakened by electricity prices together with the price for waste heat
Absorption (heat pump and cooling)	appr. 60…200 °C	+ A wide range for adjustability + A means to utilize waste heat energy for cooling purposes + Simple and durable devices which do not require much maintenance + Low electricity consumption	- A low COP in low temperatures - Relatively large investment cost compared to benefits
Organic Rankine Cycle	appr. 80…350 °C	+ Commercial technology to utilize waste heat energy directly in electricity production + Quick adjustability of electricity production + Long operational lifetime + Combined heat and power production provides a possibility to improve total operating efficiency	- Relatively large investment cost compared to benefits - Operating efficiency for electricity production typically 7-25% - In low temperature processes, the operating efficiency for electricity production typically < 10% - Requires a heat energy sink as cold as possible as over 90% of the heat energy in transferred into the heat sink so cold that it is not reasonable to utilize further

2.1 Passive means for waste heat energy utilization

Passive means for waste heat recovery usually require the heat energy to be transferred directly from the heat source to the heat sink. All of the three means of heat transfer, conduction, convection, and radiation (Perry *et al.*, 1997b) need to be considered when assessing the possibilities and applications for heat recovery.

2.1.1 HEAT EXCHANGERS

Heat exchangers are vastly utilized in all fields of technology, and their operation principle complies with the same principles regardless of the heat energy source being originated from waste heat energy. For example, there is a long tradition in different power or boiler plants for the utilization of an abundant source of waste heat energy in the flue gases to heat up water or fuel or pre-heat the combustion air (Jouhara *et al.*, 2018).

Utilization of waste heat energy as a direct heat energy source with a heat exchanger requires excellent predictability of the availability of the amount of energy, as this information is required to size the heat exchanger. The amount and temperature of the waste heat stream must be known in order to realistically and optimally size the equipment and enable efficient transition of heat energy from the source to the sink. The properties of the fluids in both sides of the heat exchanger are also crucial initial data for heat exchanger design, especially if evaporation or condensation is expected to occur in the heat transfer process (Nitsche *et al.*, 2016).

Heat recovery from waste heat with heat exchangers can be utilized, for example, for heating of properties through a central heating system or through process integration, where the waste heat source is coupled with a heat exchanger directly to another process flow.

2.1.2 HEAT ENERGY STORAGES

Heat storages are used to temporarily store heat energy either by cooling down or warming up the storage material. Different kinds of techniques and applications exist, and the choice of the applicable type of technology to each subject must be chosen depending on the costs for storage, the temperatures of both the heat energy source and sink, storage capacity, heat losses and available space for storage (Abedin *et al.*, 2011).

Heat storage technologies can be divided into sensible heat storage or latent heat storage and, additionally, to thermochemical heat storage. In sensible heat storage, the heat energy is stored by altering the temperature of the storage material, whereas a latent heat storage utilizes state change of substance to store the energy, for example, by freezing down water into ice. Because the state change of a substance requires significantly larger amounts of energy compared to just increasing the temperature in the same state of the substance, latent heat storages can be smaller in size than sensible heat storages. In other words, a latent heat storage is capable of storing a larger amount of energy into a smaller volume of storage material compared to a sensible heat storage. The size of a sensible heat storage can be decreased by turning it into a thermochemical sensible heat storage, which utilizes chemical reactions under the influence of heat energy. However, this storage type is not yet commercially very well available (Abedin *et al.*, 2011).

Guideline values can be defined to different types of heat storage technologies to describe how much energy per volume it is possible to store with a particular application. Examples of such values for a sensible heat storage with a water tank is 0.2 GJ/m^3, for latent heat storage 0.3…0.5 GJ/m^3, and for thermochemical storage 0.5…3 GJ/m^3 (Abedin *et al.*, 2011).

Sensible heat storage uses the heat energy source to cool down or heat up a solid or liquid storage material, such as water, oil, stone, concrete, or sand. The stored heat energy Q [kW] is calculated as a product of storage material mass m [kg], specific heat capacity c_p [kJ/kgK] and temperature change ΔT [K], as shown with the following equation (Abedin *et al.*, 2011):

$$Q = mc_p\Delta T \quad (5)$$

In a latent heat storage, the state of the storage material is changed in a fixed temperature, and the heat energy Q [kW] stored by this state change can be determined by the mass m [kg] and latent heat capacity L [kJ/kg] of the storage material (Abedin *et al.*, 2011):

$$Q = mL \tag{6}$$

Different materials utilized in a latent heat storage include, for example, water, paraffins and eutectic salts, and these materials are typically available for operation in a relatively narrow temperature range. The state changes (evaporation, melting, and crystallization) are all dependent on the specific heat capacity of each substance, and the change in enthalpy is extremely large when state change occurs, which enables a larger amount of energy to be stored in a certain volume by using latent heat storage rather than a sensible heat storage (Abedin *et al.*, 2011).

In a thermochemical heat storage, the heat energy is stored via chemical decomposition of storage material and it is released in an opposite reaction, when the decomposition products are reformed into the original storage material. Thermochemical heat storage has the largest energy density out of these three types of heat storage technologies, which means that it is capable of storing the most heat energy into the same storage volume. Thermochemical heat storage is particularly suitable for long-term heat storage, as it has almost non-existent heat losses, and the storage is usually operated in ambient temperature (Abedin *et al.*, 2011).

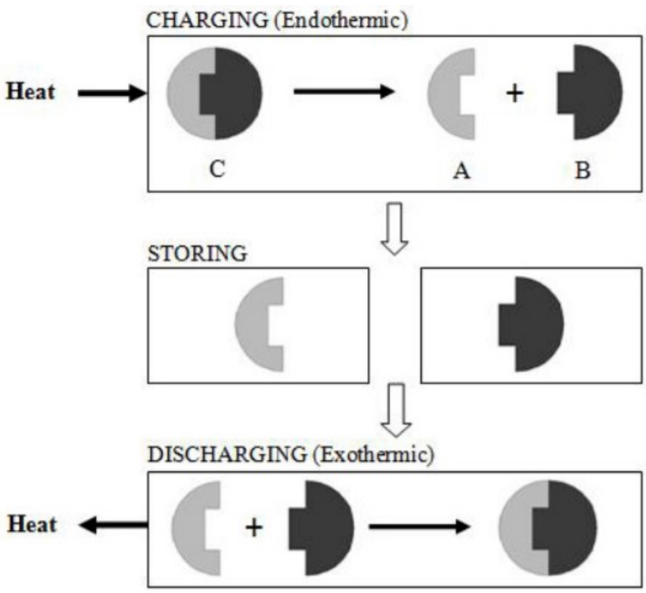

Figure 9 Storage cycle in a thermochemical heat storage (Abedin *et al.*, 2011).

Thermochemical heat storage is based on a three-phased cycle, as illustrated in Figure 9. The first phase is an endothermic reaction which charges up the storage via absorption of heat energy into storage material causing it to decompose into decomposition products. These decomposition products are stored separately in the storage phase with minimal heat losses. The third phase is an exothermic discharging phase, where the decomposition products are reunited in order to recompose storage material C for future use in this same cycle. The energy released in this reaction is the net energy that is obtained from the storage process (Abedin *et al.*, 2011).

$$\mathrm{C} + \text{heat energy} \leftrightarrow \mathrm{A} + \mathrm{B}$$

The states of the substances are not relevant, especially for the decomposition products, but usually the original storage material C is a solid or a liquid material. Decomposition product A can, for example, be a hydrate or a carbonate, and the other decomposition product B water, carbon monoxide or ammonia. When choosing a suitable storage material, the following properties must be considered: price, reactivity and durability over multiple cycles, toxicity and other safety issues, corrosiveness, availability and applicability to each process (Abedin *et al.*, 2011).

2.2 ACTIVE MEANS FOR WASTE HEAT ENERGY UTILIZATION

2.2.1 HEAT PUMPS

Heat pumps are devices that are used to transfer heat energy in the opposite direction than the Second Law of Thermodynamics commands; the temperature of the heat source is increased and released into a heat sink. There are multiple types of heat pumps, as illustrated in Figure 10, and they all require some form of external energy to operate their heat transfer cycle. Heat pumps are quite common and they have many advantages, as they typically have very high efficiency, they can be used to produce heat from an environmentally friendly and renewable source such as air or ground, and they can be used to reduce CO_2 and other greenhouse gas emissions with their energy saving potentials. The disadvantages associated with heat pump operation include the need of external energy such as electricity, potentially high investment costs, and toxicity or flammability of refrigerants (Kiss *et al.*, 2016a). A closer look at a mechanically driven and a thermally driven heat pump is addressed in the next sections.

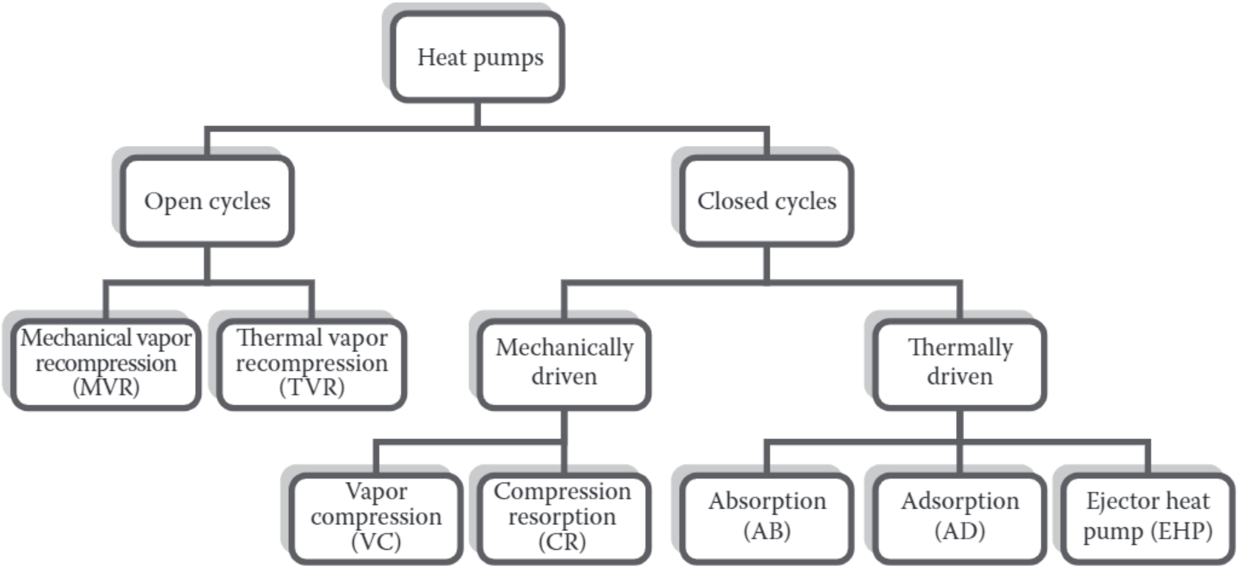

Figure 10 Different types of heat pumps (Kiss *et al.*, 2016b)

2.2.1.1 MECHANICAL HEAT PUMPS

Mechanical heat pumps are very common equipment both in industrial and domestic use, as they can be used both for heating and cooling. For example, refrigerators and air-conditioning equipment deploy heat pump technology with even a chance of switching the operation principle from heating to cooling and vice versa with the same device. Heat pumps are used to change the temperature level of heat energy Q_{EV}, and a mechanical heat pump spends mechanical energy W to release heat energy Q_C in target temperature T_C (Dimian *et al.*, 2014):

$$Q_C = Q_{EV} + W \quad (7)$$

A mechanical heat pump consists of an evaporator, a condenser, a mechanical compressor, and an expansion valve. A refrigerant fluid is circulating between these components and the whole operation of this device is dependent on the physical properties of the refrigerant: its boiling point's dependency from pressure (Brückner *et al.*, 2015).

The operation cycle of a mechanical heat pump is shown in Figure 11. The energy flow from a waste heat source will be used to evaporate the refrigerant in the evaporator even in low temperatures and pressures. The evaporated refrigerant will travel to the compressor, where it is polytropic compressed into a higher pressure and temperature. This compression step requires outside energy, usually in the form of electrical energy, to operate the compressor. This means that when an investment for a heat pump is under examination, the potentially increasing electricity consumption will need to be taken into account in operation costs (Brückner *et al.*, 2015, Dimian *et al.*, 2014).

After compression, the evaporated refrigerant will travel to the condenser and the refrigerant is condensed into liquid state. The decrease in refrigerant temperature and its state change will release an energy flow Q_C into the heat sink, and this is the desired net energy flow that is obtained from the process. The liquefied refrigerant will be directed back into the evaporator through an expansion valve, and the decrease in pressure will create a mixed vapour-liquid phase for the refrigerant before a new cycle begins with evaporation (Brückner *et al.*, 2015, Dimian *et al.*, 2014).

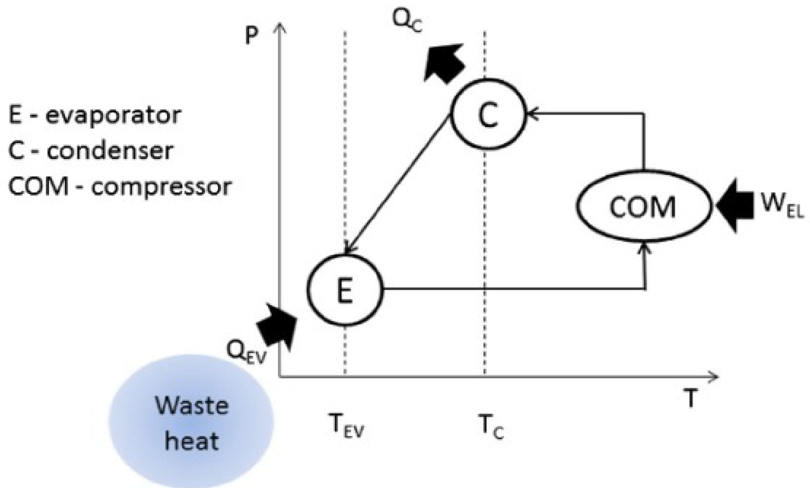

Figure 11 The operation principle of a mechanical heat pump. Waste heat provides the energy flow Q_{EV} to evaporate the refrigerant and the condenser releases the net energy flow Q_C in an elevated temperature (Brückner *et al.*, 2015).

Temperatures of approximately 120…140 °C in the heat sink are reachable by using a mechanical heat pump, although a more typical maximal temperature level that can be reached in commercial applications is approximately 80 °C. The temperature increase that is achieved with commercial applications is typically approximately 60 °C, and the range of how much temperature can be increased in the heat pump is restricted by the operation of the compressor, in other words, the pressure difference that can be obtained over the compressor as well as the dependency ratio between the chosen refrigerant's temperature and pressure (Brückner *et al.*, 2015).

2.2.1.2 Absorption heat pumps

Similarly to mechanical heat pumps, the operation principle of absorption heat pumps is also based on evaporation and condensation of a refrigerant, but instead of a compressor requiring external (electrical) energy for operation, the circulation is thermally driven. There are two circulating components, an absorbent and a refrigerant, and currently there are two commercial alternatives for these closed system mediums: water / aqueous lithium bromide solution, and ammonia / water. The circulation loop for the refrigerant flows through all components in the absorption heat pump, while the absorbent circulates between the generator and absorber, as is illustrated in Figure 12 (Brückner *et al.*, 2015).

The refrigerant is evaporated in low pressure and temperature with the energy provided by waste heat, and the gasified refrigerant travels to the absorber, where it is absorbed to the absorbent. Heat energy is released to a heat sink as the absorption occurs. The mixture of the absorbent and refrigerant is pumped to the generator, thus increasing its pressure, where thermal energy is introduced in order to evaporate the refrigerant out of the absorbent. The thermal energy can be obtained from high-temperature waste heat or from a utility. The absorbent continues in its own loop back to the absorber through an expansion valve, while the refrigerant flows to the condenser. The refrigerant will condense to liquid state in the condenser, releasing heat energy into a heat sink, and the liquid will flow through an expansion valve to the evaporator in a lowered pressure (Brückner *et al.*, 2015, Kiss *et al.*, 2016a).

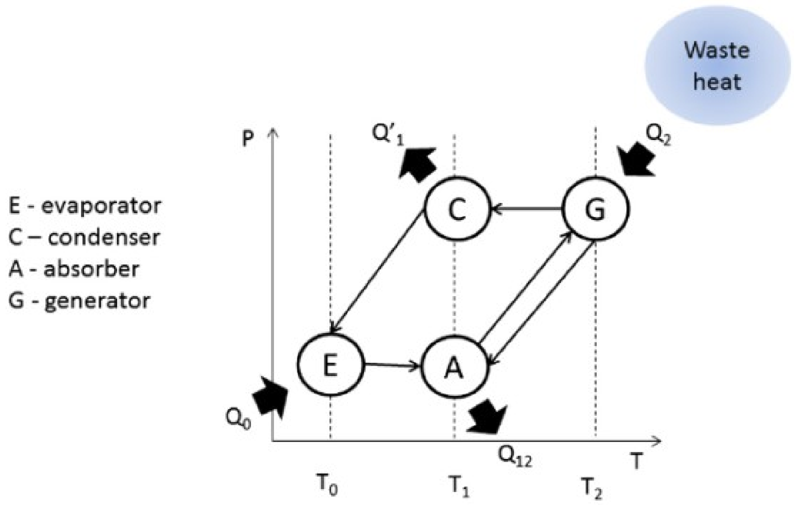

Figure 12 Operation principle of an absorption heat pump (Brückner *et al.*, 2015).

The amount of energy released by an absorption heat pump is the sum of energies released from the absorber and the condenser. The absorption of the refrigerant can be executed either in one or multiple steps with different types of equipment and technologies, some of which are presented in Figure 13. The properties of the refrigerant as well as the capacity the equipment is operated on, affect how the circulation of the refrigerant is flowing through the equipment (Brückner *et al.*, 2015).

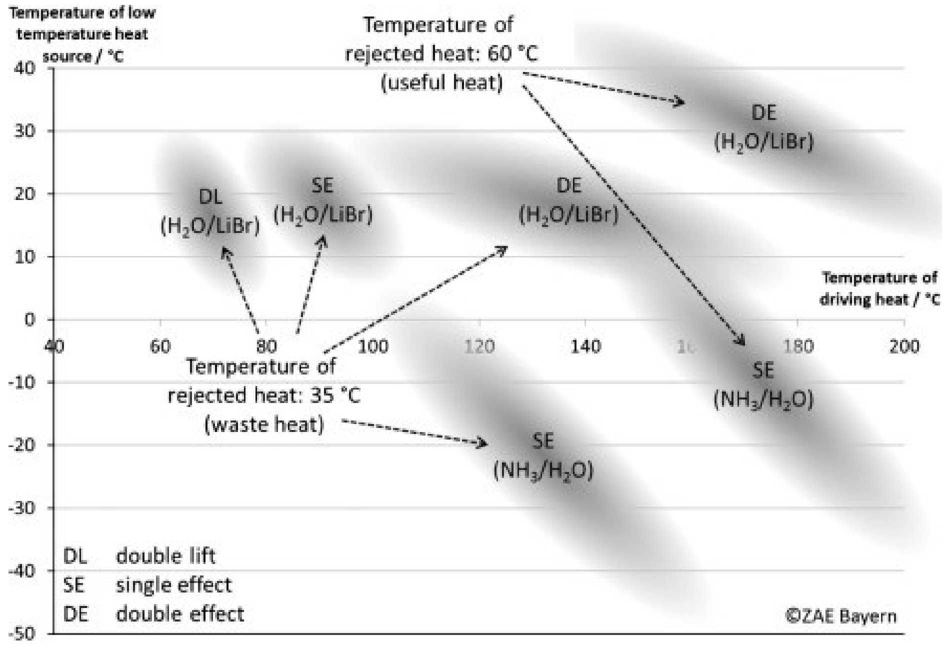

Figure 13 The choice of a suitable absorption heat pump application is dependent on the temperature levels of low and high temperature heat sources as well as the target temperature of the rejected heat energy (Brückner *et al.*, 2015).

The operation of an absorption heat pump requires relatively large volume flows and it is more complex compared to a mechanical heat pump. However, the efficiency of an absorption heat pump is extremely high in well-designed systems and it is not that dependent on temperature increases. Despite these benefits for an absorption heat pump, the mechanical heat pumps are significantly more common in commercial applications with their extremely well-known and optimized technique. This can also be caused by the larger investment costs of absorption heat pumps (Brückner *et al.*, 2015).

2.2.1.3 Efficiency ratios for heat pumps

Characteristic ratios describing different forms of efficiency can be calculated for both a mechanical and an absorption heat pump, and these ratios can be utilized when the heat pump equipment is designed and when an investment decision is being made (Brückner *et al.*, 2015).

For a mechanical heat pump, a theoretical efficiency or Energy Efficiency Ratio EER_{th} can be calculated with the temperature difference between the lower temperature T_0 [K] and the increased temperature T_1 [K] (Brückner *et al.*, 2015):

$$EER_{th} = \frac{T_0}{T_1 - T_0} \tag{8}$$

The Coefficient of Performance (COP_{th}) can be calculated also by using these two temperatures. The temperatures are in Kelvins (Brückner *et al.*, 2015):

$$COP_{th} = \frac{T_1}{T_1 - T_0} = EER_{th} + 1 \tag{9}$$

The actual efficiency is calculated based on the electrical energy required to operate the system. In these equations, $\dot{Q}_0$ is the waste heat energy provided to the evaporator and $\dot{Q}_1$ is the heat energy released from the condenser into the heat sink (Brückner *et al.*, 2015):

$$COP_{el} = \frac{\dot{Q}_1}{W_{el}} \quad (10)$$

$$EER_{el} = \frac{\dot{Q}_0}{W_{el}} \quad (11)$$

In normal operation, typical values for a mechanical heat pump are $COP_{el} = 4$ at most, and $EER_{el} = 3$. Typical efficiencies for nominal operation (EER_{th}) are approximately 60% for mechanical heat pumps and approximately 70% for absorption heat pumps.

When these characteristic ratios are calculated for an absorption heat pump, the temperatures in the generator/absorber T_2 [K] are also required (Brückner *et al.*, 2015):

$$EER_{th} = \frac{T_0}{(T_1 - T_0)} \cdot \frac{(T_2 - T_1)}{T_2} \quad (12)$$

$$COP_{th} = \frac{T_1}{(T_1 - T_0)} \cdot \frac{(T_2 - T_0)}{T_2} = EER_{th} + 1 \quad (13)$$

For absorption heat pumps used as chillers, the actual efficiencies of the application are calculated analogous to those of a mechanical heat pump, but the values are typically significantly higher. EER_{el} can be even 20 or higher with well-designed refrigeration systems, the value typically being approximately 12. These chiller applications do not require a cooler to release heat energy and the heat released is approximately 2.5 times the amount of waste heat energy provided to the system, and the values of COP_{el} can reach 40, even 60 (Brückner *et al.*, 2015).

Heat pumps can be used in various applications and obtain moderate *COP* values even with under 10 °C waste heat sources. The stability and sufficiency of the flow of waste heat energy are emphasized with colder heat energy sources in order to have technical and economic feasibility for a heat pump investment (Gynther *et al.*, 2019).

2.2.2 ORGANIC RANKINE CYCLE

The Organic Rankine Cycle or ORC is a combined heat and power generation process that can be used for internal small-scale electricity production (Quoilin *et al.*, 2013). It is a circulation process where instead of water being used as a circulating liquid, an organic substance is used, and this substance will evaporate and condensate in the different cycle steps based on its substance properties. With suitable circulation liquids, even 100 °C heat sources can be utilized in small-scale electricity production, but typically the generation of electricity will be achieved with a better efficiency by using hotter energy sources up to 350 °C (Maaskola *et al.*, 2014).

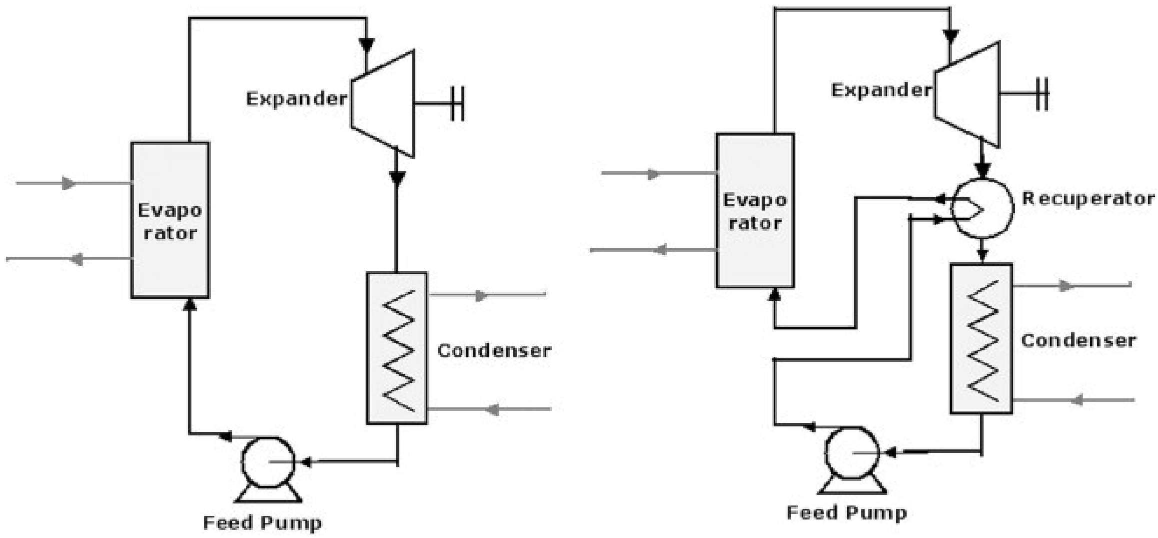

Figure 14 Two simplified illustrations of ORC, without a recuperator and with it (Quoilin *et al.*, 2013).

The main equipment utilized for ORC are a feed pump, a boiler or a heat exchanger to operate as an evaporator, a turbine, a condenser, and a recuperator, as shown in Figure 14. The feed pump increases the pressure of the circulation liquid up to its boiling pressure, and then the liquid's temperature is elevated until it boils in a boiler or in a heat exchanger evaporator. The function of the turbine is to salvage mechanical energy which is released when the evaporated circulation liquid expands into a decreased pressure. This low-pressure steam will then travel into the condenser, where it is condensed into liquid form (Maaskola *et al.*, 2014). The efficiency of the equipment can be improved by inserting a recuperator to pre-heat the liquid that is traveling from the feed pump into the evaporator. Pre-heating can be executed by using the heat energy provided by the low-pressure superheated steam that is traveling to the condenser. (Quoilin *et al.*, 2013).

A theoretical efficiency for this cycle can be calculated as a Carnot efficiency with temperatures in Kelvins:

$$\eta = 1 - \frac{T_{min}}{T_{max}} \qquad (14)$$

where T_{max} is the temperature of the waste heat source and T_{min} is the minimum temperature that is gained in the condenser. In practice, 45% of this theoretical efficiency can maximally be achieved, but in low temperature processes, the efficiency for electricity generation remains nevertheless relatively low, under 10%. These temperatures that are used in determining the efficiency of the cycle also affect the election of a suitable circulation liquid for each application (Maaskola *et al.*, 2014).

The circulation liquids can be categorized based on their saturation vapour curves to wet, isentropic and dry liquids. Figure 15 presents the differences between these three types of fluids. For example, water is a wet fluid, which will demand to be superheated in order to prevent premature condensation which might damage the turbine blades. The isentropic and dry fluids do not require to be superheated, making these types of fluids more suitable for ORC. Isentropic fluids remain as saturated vapour throughout the turbine (Bao & Zhao, 2013) and a characteristic quality for dry circulation liquids is that they can be superheated when they are expanding in the turbine (Maaskola *et al.*, 2014).

A suitable circulation liquid should be chosen for each application in order to ensure that the evaporation and condensation properties of this substance are optimal regarding the pressure and temperature utilized in the Rankine process. As most dry organic circulation liquids utilized in these applications superheat as they expand in the turbine, traditional erosion problems in the turbine caused by moisture can be avoided. Compared to water, these organic liquids typically have a significantly larger molar mass, which contributes to and enhances the transformation of gas expansion into mechanical work in the turbine with decreased pressures and more simplified turbine structures. Additionally, the decrease in specific enthalpy is typically lower than with water, which allows single step turbines to be utilized without compromising efficiency (Maaskola *et al.*, 2014).

Additionally, compared to a traditional water cycle, the organic circulation liquids tend to wear out mechanical parts less, they lubricate the moving parts or their electrical properties are such that there are no adverse effects if they are released, for example, to the generator. Despite these positive properties, organic circulation liquids naturally also have disadvantages, such as the fact that they are often harmful to the environment, toxic, or inflammable. As organic substances, they tend to decompose in high temperatures and they can react chemically if contact with other substances, such as air or lubricants, is permitted. They are typically also more expensive than water and less available. Typical organic liquids that are used often in the Organic Rankine Cycle are for example, n-pentane, silicon oil (octamethyltrisiloxane, polyphenylemethyledimethylesiloxane), and toluene (Maaskola *et al.*, 2014).

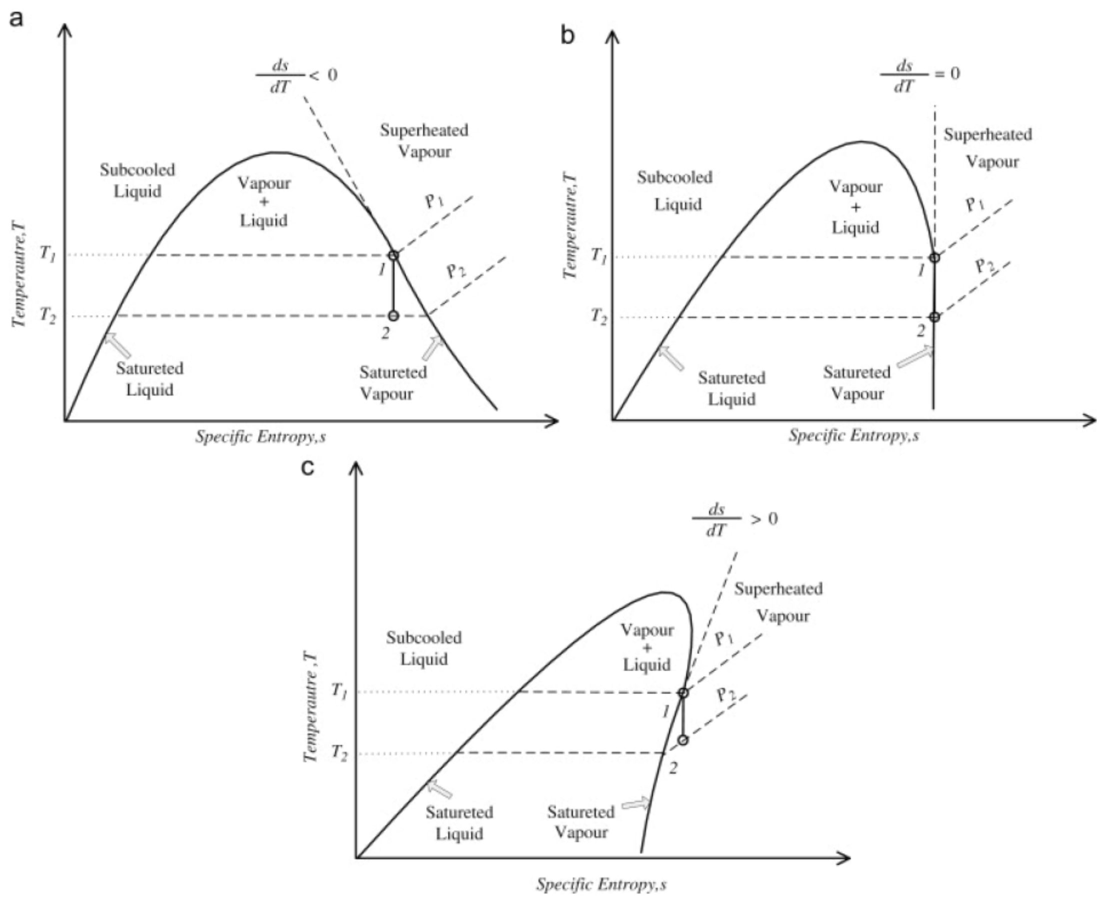

Figure 15 The saturation vapour curves for different types of circulation fluids, a) wet, b) isentropic, and c) dry (Bao & Zhao, 2013).

Instead of using an organic substance as the circulation liquid, a mixture of water and ammonia can be used. This also changes the name of the cycle from Rankine to Kalina cycle, and enables higher efficiencies to be achieved, but more complex and expensive equipment is required for the process (Maaskola *et al.*, 2014).

The heat can be introduced to the Rankine process with a thermal oil circulation or by directly heating up the organic cycling substance with the heat source. ORC can utilize simpler equipment in its circulation than traditional steam cycles, as with once-through boilers there is no need for water drums and recirculation. (Quoilin *et al.*, 2013). A traditional turbine technique is typically used with an air-cooled generator and a lubrication oil system. In ORC, besides acting as the liquid to be circulated in the system, the organic circulation liquid can simultaneously act as a lubricant and a refrigerant for the generator, which enables hermetic sealing of the whole equipment entity, thus, preventing any leakages to the surroundings (Maaskola *et al.*, 2014).

The efficient operation of the turbine and a suitable circulation liquid are crucial for the total performance of the ORC system (Quoilin *et al.*, 2013). Operating the turbine with optimal efficiency requires that it is operated with a sufficient gas flow velocity. If a small equipment is utilized with small impellers, the necessary decrease in circumferential speed due to the size must be compensated by increasing the rotation speed. In scenarios such as this, the impellers of the turbine can rotate approximately 30,000 rounds per second, which means that the turbine is operated with a so-called high velocity technique (Maaskola *et al.*, 2014).

The thermal power obtained with ORC is typically 1 to 50 MW_{th}, and the electrical efficiency of the turbine is 0.2 to 10 MW with a power to heat ratio of approximately 20% / 80%, leading to the total efficiency being as high as 98%. The condensation step that releases energy can be executed for example to water or to pre-heat combustion air, thus, enabling the recovery and utilization of heat energy. When ORC is used to recover heat energy from low temperature waste heat sources, the electricity efficiencies typically remain relatively low, under 10%. Additionally, as a sufficiently large heat energy sink is required where the heat energy can be released from the process, having a possibility to recover condensation to heat up a stream of water would be ideal, instead of removing the heat energy unused in a waste stream (Maaskola *et al.*, 2014).

The problem with low temperature heat energy flows is that most of the heat energy released to the heat sink has such a low temperature that its further utilization is no longer reasonable. Additionally, especially in low temperature processes, the volume flows should be large, which also increases the costs caused by the electricity utilized to enable pumping of the liquid flows, thus, potentially decreasing the overall cost-efficient profitability for the ORC investment. In order to make use of ORC profitable for low temperature heat sources, the heat energy provided to the Rankine cycle should be practically free of charge and it should be readily available in a sufficient amount so that the peak operation time in the Organic Rankine Cycle operation can be maximized. The electricity price should also be relatively high in order for the ORC process to maximize the savings it creates, and also, the distances that liquids are pumped should not be unreasonably long (Maaskola *et al.*, 2014).

2.3 Utilization of waste heat in regional or district heating networks

If there are no possibilities for utilization of recovered heat energy within the plant itself, either in its processes or to keep the properties heated, its utilization by external parties can be studied. Waste heat can be sold to external parties in the same industrial area, possibly to a regional heat network, or to a district heating company to be used in the district heating network.

All possibilities to join new heat sources to a district heating network requires case-by-case examination. The temperature and pressure levels of the waste heat source must be harmonized according to the point of the district heating network the waste heat would be attached to, and additionally, the whole district heating network stability and equilibrium must be evaluated to consider the changes this new heat source might cause for it. These evaluations are necessary in order to ensure reliable and effective operation of the network. The following questions can be used to aid in the design of a correct attachment point to each source (Sirola *et al.*, 2018):

- What is the value of the heat provided to the network?
- What kind of heat energy and how much of its production is displaced by the new heat energy source?
- What is the capacity of the current district heating network and where are its potential constraints?
- What is the point and means of attachment?
- What is the temperature provided to the network? Is it controllable?
- What is the pressure in the attachment point and is it possible to pump the water to be fed into the network into this pressure?
- What is the reliability of delivery for the waste heat source?
- How is the operability and control of the network ensured?
- How to ensure the optimization of the operability of the entire district heating network?
- How will the responsibilities be divided between the two parties (purchaser and supplier) regarding investment, maintenance, and operational costs?

The temperature level of the waste heat source to be attached to a district heating network will have an effect on the amount of heat transfer available and the technique to be chosen for heat transfer. The design temperature of a district heating network is 120 °C. The temperature of water in the flow going to the utilization points changes depending on ambient temperature, which is typically between 75 and 115 °C (or for low temperature network 65 and 90 °C), and temperature in the flow returning from the network is typically between 40 and 60 °C. The design pressure is typically 1.6 MPa (1.0 MPa for some district heating networks), and the pressure in the outgoing pipeline close to the main heat energy production plant is practically close to this design pressure when the plant is operating at maximal capacity. The pressure difference between the outgoing and incoming pipelines of the network can be relatively high near the main production plant, even 1.0 MPa, even though the design pressure difference between these two flows is typically 0.55...0.6 MPa (Sirola *et al.*, 2018).

Heat energy is primarily fed into the pipeline outgoing from the main production plant, because feeding it into the returning pipeline will increase heat losses and decrease the overall efficiency of the network. Design criteria and values for the pump that is feeding water into the network from a waste heat source are selected upon the prevailing pressure in the network at the attachment point (Sirola *et al.*, 2018).

The attachment of waste heat to a district heating network is always conducted indirectly with a heat exchanger separating the network and the waste heat energy. The possibility and opportunity to utilize waste heat energy to heat up water that is fed into a district heating network can be improved by increasing its temperature with different techniques, such as heat pumps or boilers (Sirola *et al.*, 2018).

The amount of heat energy that can be fed into a district heating network is always restricted by the current consumption in the network, which can be relatively low especially during summer months. The basic principle for choosing the heat sources to be utilized among the ones that are currently available in the network is by utilizing them in the order of affordability. This means, that the intake of heat energy may not be perpetual, and fluctuation is caused by seasonal change as well as potential failure or disturbance situations. In these kinds of occurrences, the heat reception capacity might be weakened or even entirely cut off. The waste heat energy supplies must be prepared and equipped for these kinds of situations and, when necessary and with sufficient speed, be able to interrupt waste heat production or route it to some other destination temporarily (Sirola *et al.*, 2018).

In a situation where a heat pump is connected to a district heating system, it will act as a boundary between the district heating network and an individual heat source and, thus, the party that owns, maintains and controls the heat pump should be carefully considered. Would it be more efficient and profitable, if the district heating company owns and operates the heat pump, thus, being able to better control the efficiency and operability of the entire district heating network? When examined from the opposing viewpoint, the investment, operation, and maintenance costs of a single heat pump might not outweigh the benefits. When considering this issue, also taxation should be borne in mind (Rämä *et al.*, 2020). In Finland, since the beginning of 2021, heat pumps providing heat to a district heating system can be taxed with a lower taxation group (the European Commission, 2021). This obviously requires the consumption of electricity used for that given heat pump to be measured and taxed separately if the company itself belongs to the higher tax category. Lowering the tax group of such heat pumps can increase the interest of industrial companies to seek possibilities for such co-operation with district heating companies (Rämä *et al.*, 2020).

The waste heat sources provided by industry are typically large and point-like, and even though the temperature level they provide can have great variations, they are usually on a higher temperature level when compared to waste heat sources provided by urban areas or nature. With industrial waste heat sources, one of the most typical features is the uncertainty about their stable delivery certainty or the level of heat energy provided at any given time point. These variations and uncertainties may be caused by planned or unplanned plant shutdowns, and the level of heat energy provided to the district heating network can alternate in relation to predefined process parameters and production levels. As waste heat is a by-product, its production levels are not the driving parameters to how the production plant is operated, and this also means that a district heating company does not have any controlling possibilities concerning the amount of heat energy they receive from one source. Typically, the heat energy provided to the district heating network remains relatively stable, despite these characteristics, enabling them to be counted on as one of the basic loads provided to the district heating system. Other alternative heat production capacities are operated accordingly based on the basic load in the system (Rämä *et al.*, 2020).

According to a study conducted among different Finnish district heating companies during summer 2020, approximately 88% of these companies are already utilizing waste heat sources in their networks. The remaining 12% are currently not utilizing waste heat but are interested in it and are negotiating possibilities for utilization. Generally speaking, district heating companies are interested in acquiring heat energy from waste heat when it is technically and economically possible, as this enables them to decrease their emissions and source cost-effective heat to their customers, thus, also ensuring that district heating remains a competitive and interesting product (Tiitinen, 2020).

District heating companies are well familiarized with large and high temperature waste heat sources, but they are also working with locating potential sources to provide a lower temperature heat, which could be utilized by increasing the temperature with, for example, a heat pump. It is evaluated that the share of waste heat, geoenergy, and heat pumps will increase from the current 10% to 30% in the next 10 years (Tiitinen, 2020).

3 CASE STUDY FOR IDENTIFICATION AND UTILIZATION OF EXCESS HEAT

The potentials of utilizing excess heat energy were studied with a case study for Finnfeeds Finland Co factory in Naantali. This survey was carried out as a part of an obligatory energy audit, where the energy consumption in the process and facilities were examined in its entirety. The opportunities to utilize waste energy was chosen for closer examination in this audit, because the energy efficiency of the plant has been recognized as an important potential for improvement.

Finnfeeds Finland Co is a part of the American IFF corporation, and it refines sugar beet molasses into fine chemicals. The main products can be utilized in food, feed, cosmetic or chemical industry applications. The Naantali plant has a long history, as it began operation already in the 1950s as a sugar factory. The factory has been in its current use since the 1990s (Port of Naantali, 2021, DeVeau, 2021).

The refining process of sugar beet molasses is a highly energy-intensive process, as it includes processing in sufficient temperatures as well as removing a large amount of water, as the products are primarily in crystal form. The raw materials are heated up to temperatures of 70…95 °C, and after the pre-treatment steps, such as filtration, the raw material is separated into different fractions with chromatographic separation. All fractions are evaporated to remove excess water, and most of the fractions are evaporated by an external party with an MVR evaporator. By utilizing this external party, Adven, to evaporate the main fractions, significant reduction in steam usage has been achieved by the Naantali plant since beginning this cooperation in 2018 (Adven, 2021).

After the separation and evaporation of the fractions, they are further processed into their final form by first crystallizing them in a crystallization boiler. The solids content is significantly increased in the crystallization boiler and the crystals are separated from the massequite with centrifugation. Before the product is ready to be stored and packed, it has to be dried even further with a drier utilizing cold and hot air to remove any remains of water, producing a dry and solid product crystal.

The major utility systems in Naantali are steam, air, and different water systems such as hot water, cold water, and cooling water which comes from the sea. Heat transfer is mainly carried out with either steam or cooling water; only a few heat exchangers utilize hot water or other process streams as a source for heating or cooling. Besides these utilities, for example, some chemicals are also utilized in the process or in cleaning of the equipment. Many of the utility systems also require heating, and also these heat exchange operations are mainly carried out with steam.

As there are many unit operations where water is evaporated, there is a significant amount of secondary steam available. Currently this secondary steam is condensed with surface condensers and the condensate is utilized again in the processes. This state change is achieved with cooling water, meaning that the large amount of energy released from the state change is transferred to the cooling water and exits the plant as the cooling water is returned to sea after use.

The process is a mixture of continuously operating unit operations and batch-wise operating unit operations, which makes the integration of processes and their energy requirements more difficult. Currently, there is very little utilization of waste heat energy anywhere in the Naantali plant, even though some theoretically possible waste heat sources have been identified.

The premise for this survey was to systematically go through the process and define an energy balance for it, as well as:

- identify theoretically, technically and economically available waste heat sources;
- identify theoretically, technically and economically available heat sinks;
- carry out a preliminary design for aforementioned potentials for utilization of waste heat;
- carry out preliminary cost estimation calculations as well as evaluate potential investment payback time and savings in operation costs (utility costs).

The principles of waste heat utilization illustrated in Figure 2 in the Introduction chapter of this thesis were followed in this case study survey, meaning that the primary focus was to find the available heat sinks for waste heat internally. Despite this, also the alternative for waste heat utilization in district heating was briefly examined, as this option was thought to be an interesting one worth of investigating.

3.1 ENERGY BALANCE

Currently, most of the heat exchangers are operated with steam or cooling water, which is not necessarily particularly energy efficient; only a few heat exchangers are using hot water or another process stream as a heat transfer medium. Most heat exchangers have a temperature measurement in the hot side of the stream that is being heated, so additional measurements were required with a portable surface temperature measurement device in order to be able to define a change in the temperature over the heat exchangers. The substantial use of steam as a heat transfer medium provides a fruitful starting point for decreasing primary energy consumption, if useful alternative medias for heat exchange are identified.

The survey began with gaining an understanding of the production process and the energy sinks in it. This phase included multiple interviews with various staff members, first introducing the operation principles of the process and then deepening this knowledge with more information about the ways energy is utilized throughout the plant. The different unit operations of the process were discussed, as were also the different utility systems that are in operation. After gaining an understanding of how the process is operating, studying the current state of the process and evaluating the parts in need of heat transfer were carried out. This was carried out, in addition to the previously mentioned interviews, by examining the process from PI diagrams, collecting measurement data for useful process measurements, and executing additional measurements *in situ*. In addition, a large number of documents provided by Finnfeeds Finland were utilized to examine the production and energy consumption from 2020, as well as production plans and mass balances illustrating how the product compounds are moving through the operation of the process.

The properties of different substances and streams were determined based on mass balances and universal values. For example, the densities of water, sea water and air were determined to be 999 kg/m^3, 1,026 kg/m^3, and 1.006 kg/m^3, respectively. The densities of the process streams range between 970…1,100 kg/m^3. For water (and steam), many of the values needed in the calculations, such as specific enthalpy or enthalpy of evaporation, are dependable on pressure, so these values in the calculations were chosen accordingly. The specific enthalpy and specific heat capacity for all gases (e.g., exhaust air and odour gases) were not known, so reference values for air were used in calculations. As the specific heat capacities for the process streams (raw materials, intermediary flows, or products) are not known, the reference values of either water (4.19 kJ/kgK) or Thermera R (3 kJ/kgK) were utilized.

The energy balance for the process was divided into the different departments functioning inside the plant, which was also a logical way of division, because only one of the departments operates continuously, while the other ones are operated according to production demands in a batch-wise style. The purpose of this energy balance was to give an overall illustration of the process properties such as flow, temperature, and solids content, to show the unit operations in the process where heating or cooling of the process stream is required, and to illustrate estimates on how much energy is wasted either to cooling water (sea water), effluent water, or in gas phase into the odour gas treatment system.

Based on average consumptions of heating and cooling utilities, the total value of consumption can be divided into percentages for different unit operations according to Figures 16 and 17. Besides presenting the cold utility utilization, Figure 17 presents other means for heat energy to exit the process, in other words some of the identified waste heat sources are also considered in this pie chart.

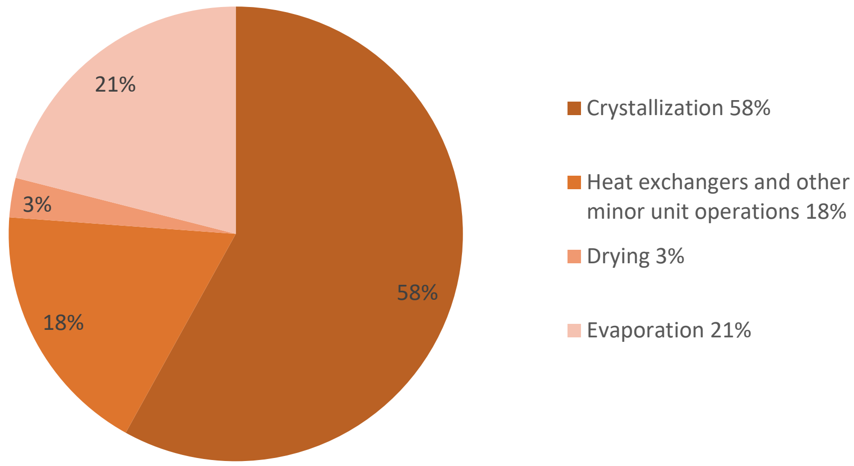

Figure 16 Division of average heating energy consumption to different utilizations. Evaporation does not include MVR evaporation by external party.

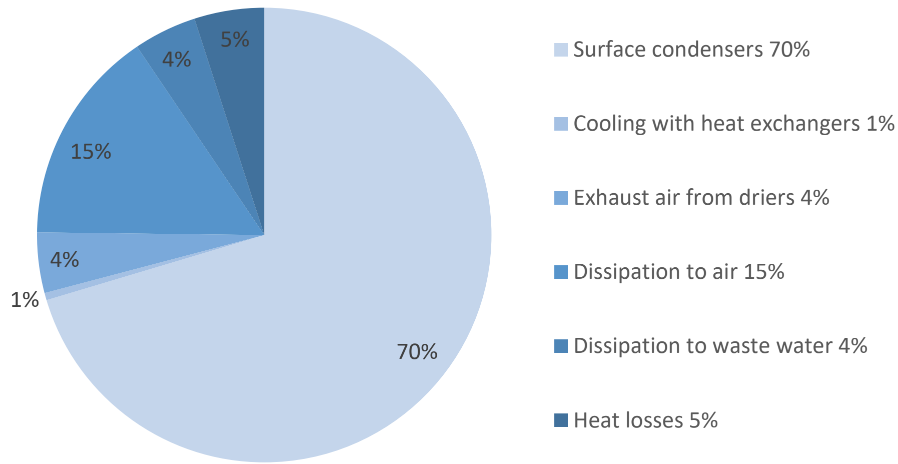

Figure 17 Division of heat dissipation either to cold utilities or to waste heat sources.

The average total energy consumptions were calculated based on measurement data from 2020. The average total steam consumption was calculated to be 5.5 MW or 4,000 MWh per month, and 0.1 MW and 40 MWh of condensate was returned from the factory to the company providing the steam on average per hour or month, respectively. The heat load transferred from the process to the cooling water system and returned to the sea was calculated based on average monthly flow rates and a temperature difference of 10 °C between the incoming and outgoing cooling water. The average heat load in the cooling water was estimated to be 3.8 MW and 2,400

MWh per month. The amount of odour gases was estimated based on the design value and operational efficiency of the odour gas blower and an average temperature 25 °C. An illustration of the total energy balance with different waste heat sources is presented in Chapter 3.3 in Figure 24.

3.2 PINCH ANALYSIS

The energy balance was used as an information source for the Pinch Analysis, which was carried out with three individual scenarios to best describe the different production models in the factory. These scenarios considered the continuous and batch-wise unit operations in the different process departments. In all of these scenarios, the continuously operating unit processes are addressed similarly, and the scenarios differ according to alternative products batch processes:

1. Continuous unit operations and batch process for Product 1
2. Continuous unit operations and batch process for Product 2
3. Continuous unit operations and batch process for Product 3

The Pinch Analysis was carried out with Aspen Energy Analyzer V10. A value of 2 °C was used as ΔT_{min}, as most of the heat exchangers in the Naantali plant are counter-current plate heat exchangers with possibilities for very efficient heat transfer. The streams chosen to be included in the Pinch Analysis were such that require regular heating or cooling. Due to this, all of the different CIP (Cleaning In Place) cleanings that are carried out regularly or irregularly in the factory were simplified as one CIP stream in this survey. The initial and target temperatures for each stream were chosen to represent a stable operation situation for the process, so the results obtained from this analysis do not consider such situations as start-up or shut-down, where additional energy could be required.

A Composite Curve and a Grand Composite Curve were plotted for each scenario and they are presented in the following Figures 18…23.

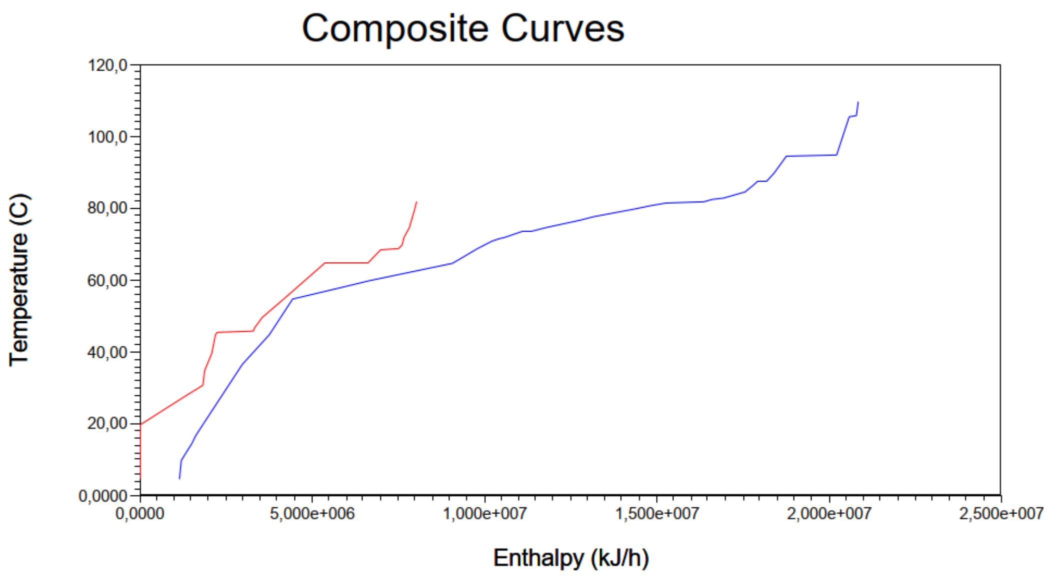

Figure 18 Composite Curve for scenario 1.

As can be seen from the CC, there is a larger demand for heating energy than for cooling. This can be explained partly with the data that was analysed with Pinch Analysis, as it also included utility streams that require heating. Additionally, the difference between these two Composite Curves is explained with the energy that is now dissipated to the environment, and both of these aspects can be seen also more substantial in the results of the other two scenarios. All in all, there are only a few process streams that require cooling but do not include a state change, and also explains the comparably small hot Composite Curve.

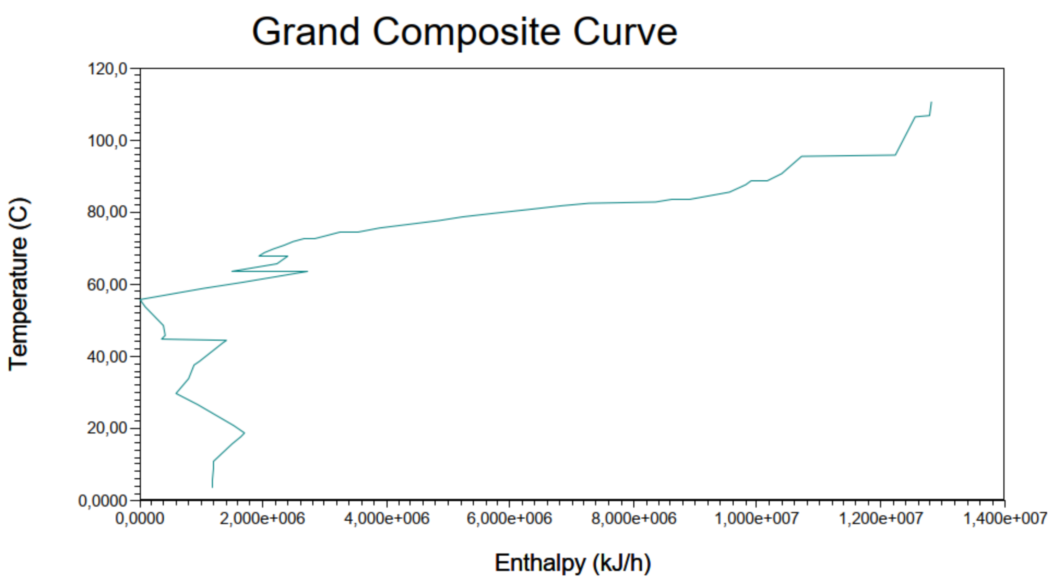

Figure 19 Grand Composite Curve for scenario 1.

The Pinch point for scenario 1 was 56 °C, as shown in the GCC. The plant's heat exchangers were examined based on the Pinch points obtained from each scenario to determine whether or not they are operating over the Pinch point. Appendix 1 contains the list of all heat exchangers and the evaluation for each scenario, and it was discovered that for scenario 1, there are 16 heat exchangers out of 50 that are operated over the scenario's Pinch point. This represents an estimated 37% of the heating and cooling demands in this scenario.

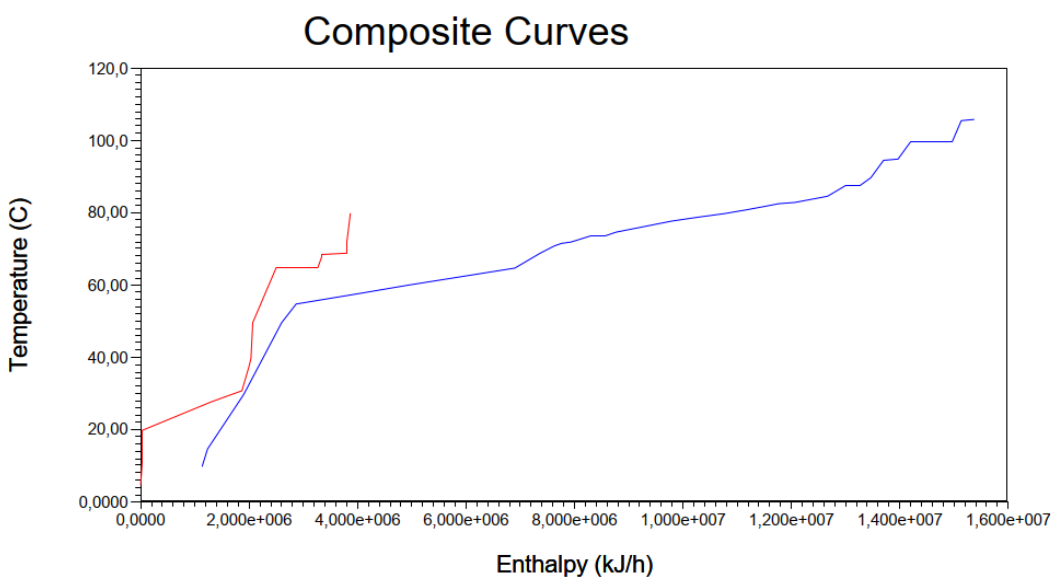

Figure 20 Composite Curve for scenario 2.

The significant difference in heating and cooling demands was much more emphasized in the second scenario, as there are less streams that are cooled when Product 2 is in production. Most of the streams requiring cooling are cooled in order to carry out a state change, which can be seen in the CC as steep slopes.

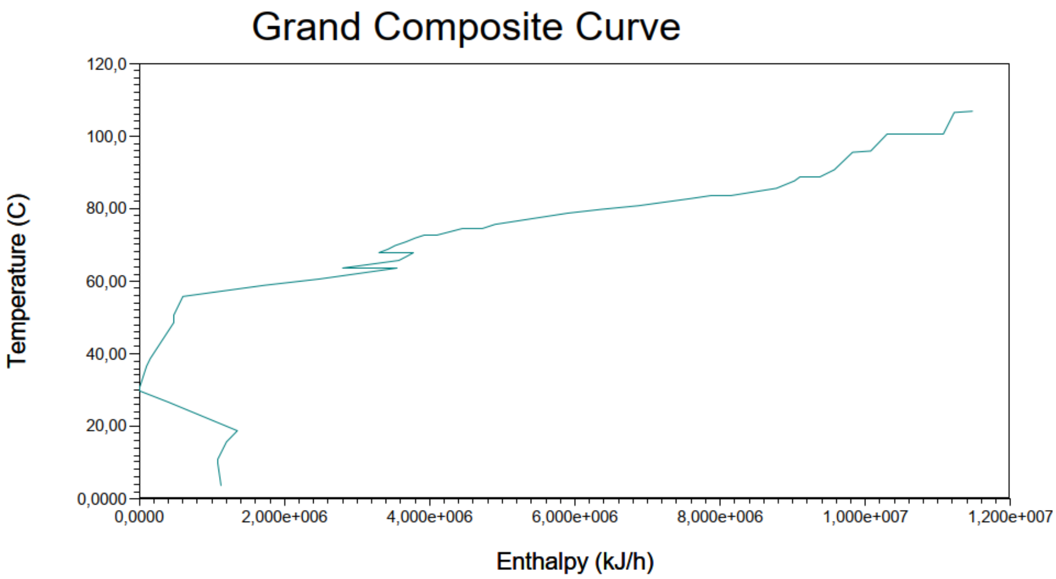

Figure 21 Grand Composite Curve for scenario 2.

The Pinch point for scenario 2 was 30 °C and according to Appendix 1, there are currently 9 out of 46 heat exchangers that are operated over the Pinch point, which stands for 33% of the scenario's heating and cooling energy demands.

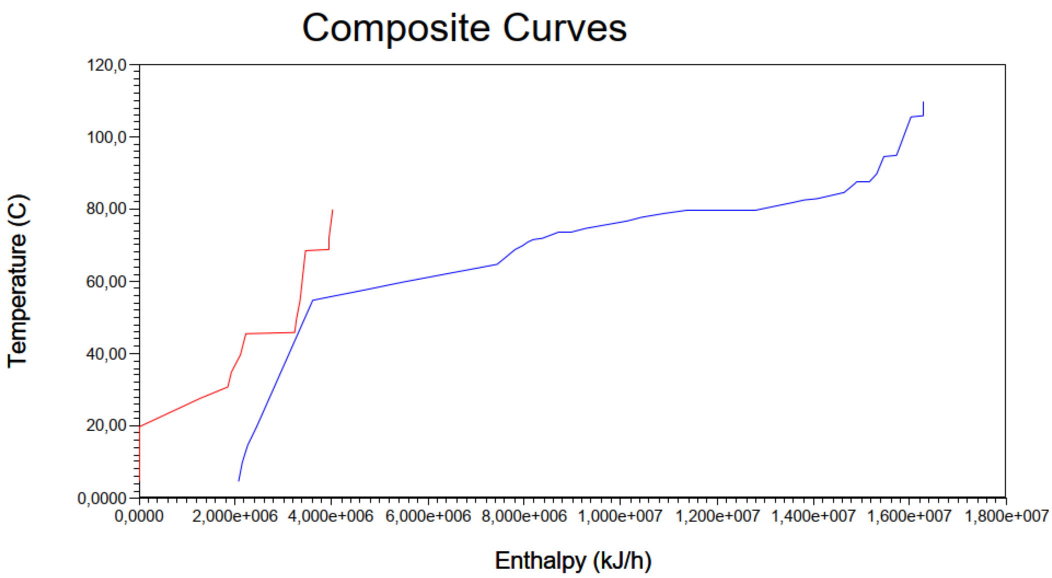

Figure 22 Composite Curve for scenario 3.

Similarly, also scenario 3 contains less streams requiring cooling than scenario 1, and the form of the cooling CC is heavily impacted of the state changes caused by cooling of secondary steams. The GCC for scenario 3 presents that the Pinch point for this scenario is at 45 °C. According to Appendix 1, there are 9 out of 46 heat exchangers that are operated over the Pinch point, which represent approximately 29% of the heating and cooling energy demands in this scenario.

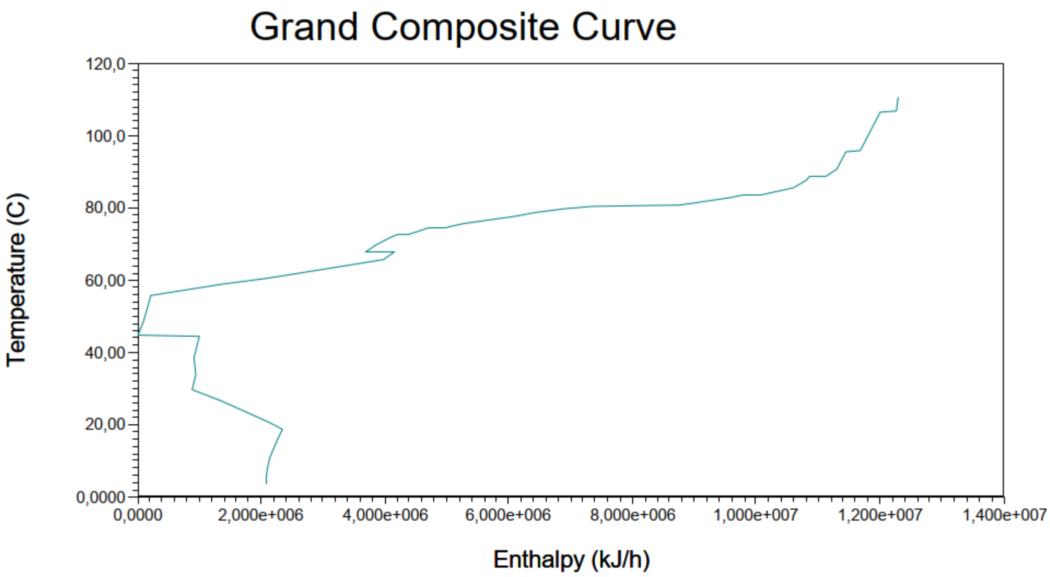

Figure 23 Grand Composite Curve for scenario 3.

When examining the heat exchangers that are operated over the Pinch points in different scenarios, it can be noticed that there are seven heat exchangers that are operated over all of the three Pinch points. Some of these are associated with utilities, one with heating of raw material, and some are surface condensers.

Despite the differences between these scenarios and their Composite Curves, it can be concluded that in the current operational models, there is a limited number of possibilities for process integration due to the scarcity of streams that need to be cooled down compared to the number of streams that require heating in suitable temperature levels. This means that most probably there are multiple separate sources for waste heat, and these are examined further in the following section. As most of the process streams or utilities are heated up to a relatively high temperature of 70…95 °C, it is especially difficult to find possibilities for process integration in these higher temperature levels. Additionally, as most of the streams in need of cooling actually require to be cooled down in order to carry out a state change which needs to be executed in a controlled fashion, this also complicates process integration.

The minimum energy requirements or energy targets for hot and cold utilities are obtained from the Pinch Analysis, and they are gathered together with the actual average hot and cold utility consumptions in Table 2. The actual average consumptions are based on measurement data from 2020.

Table 2 Energy targets according to Pinch Analysis and average consumption of hot and cold utilities

Scenario	Heating utility demands		Cooling utility demands	
	kJ/h	MW	kJ/h	MW
1	1.28E+07	3.6	1.18E+06	0.3
2	1.15E+07	3.2	1.14E+06	0.3
3	1.21E+07	3.4	2.23E+06	0.6
Average consumption		5.5		3.8

As can be seen from the table data, the average consumptions are larger than the energy targets, especially with cooling utilities. This can at least partly be explained by the simultaneous operation of batch processes producing different products. In addition, the scarcity of process integration explains why the average consumption for utilities is higher than the Energy Targets. As the heating demands are much more emphasized than the cooling demands, the possibilities of finding suitable process integration pairs with compatible temperature levels may be challenging.

3.3 WASTE HEAT SOURCES

The biggest sources for waste heat sources such as secondary steam and cooling water were already identified as waste heat prior to this survey. By utilizing the data obtained from the energy balances and Pinch Analysis, the different sources for waste heat were identified. All of these sources are gathered in Tables 13…15 in Appendix 2, and only the largest individual sources that were identified are illustrated in Figure 24 together with the average steam consumption in order to illustrate them in as a type of energy balance. The heat content in each waste heat source was estimated with average mass flows and specific enthalpies. The amount of condensates' expansion steam was estimated based on total steam consumption.

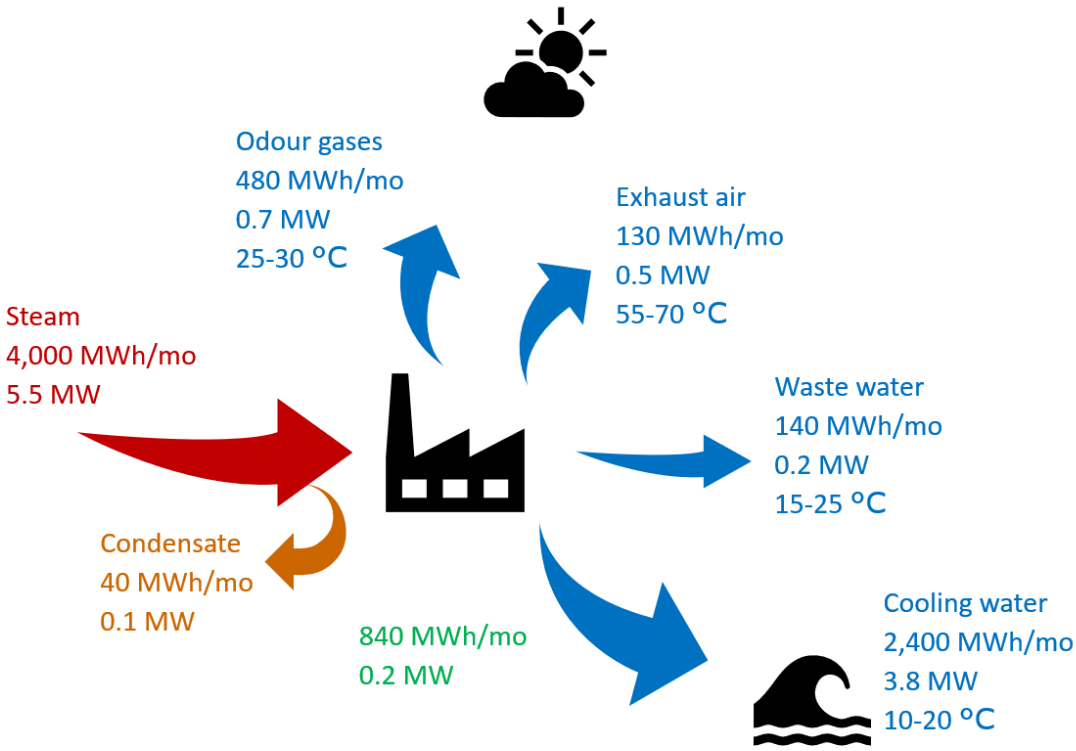

Figure 24 The biggest sources of waste heat and their average heat loads.

The waste heat sources include purified exhaust gas and air streams, waste and cooling waters as well as some final or intermediary products that currently are routed without any cooling from the process to their storage tanks. There are currently multiple storage tanks located outside in the Naantali factory, and a part of these tanks contain liquids that are in relatively high temperatures and there are no heating coils or other means of maintaining the temperature in the liquid. In such tanks, the temperature of the liquid is decreasing over time as the heat energy dissipated to the environment.

The tables in Appendix 2 present the theoretical, technical and economic potentials for each waste heat source. The economic potential was determined either based on the preliminary investment proposals sketched up in this thesis or by evaluating the probability of their economic viability. The total theoretical potential was estimated to be 24 MW, technical potential 23.7 MW and economic potential 22.6 MW. As many of the identified waste heat sources were such that are now requiring cooling in order to carry out a state change, the total potential presented in these tables must be interpreted in such a way that the heat energy can be recovered from one source only, either directly from the individual source replacing the cooling utility, or from the cooling water system as a part of its total heat load.

As can be seen from the Tables 13…15 and from Figure 24, the temperature of identified waste heat sources remains relatively low. The heat loads provided by individual waste heat sources also vary significantly from each other. Despite this large variance between these sources, there are possibilities of finding utilizations for them as the process contains a lot of potential heat sinks with different amounts and temperature demands. Basically, all flows in need of temperature increase are potential heat sinks, but what might become problematic is the fact that many of them require heating to a relatively high temperature, which cannot be achieved with waste heat sources by passive means of heat transfer.

Some preliminary investment proposals were sketched in order to utilize some of these identified waste heat sources and simultaneously decrease the number of heat exchangers that are operated over the Pinch point(s).

4 POTENTIALS FOR UTILIZATION OF WASTE HEAT

For the utilization of waste heat found in the process, several preliminary investment proposals were outlined. All of these proposals include an energy balance to show how the waste heat energy can be utilized, and these balances are based on certain production capacities for different products.

An illustration of the energy balance as well as required modifications or additions to the existing equipment or piping were illustrated on flow sheets in Appendix 3. No additional details such as pressure gauges or by-pass pipelines are presented in these flow sheets, the purpose of them is solely to illustrate what is required and what is achieved temperature-wise with each waste heat source and sink pair.

Investment costs were calculated by utilizing AFRY's average price database for equipment and pipelines. The estimation for mechanical works includes materials and installation work for pipelines, manual valves and all pipe accessories needed for installation of each pipeline. The prices for electrical, automation, and instrumentation works (EIA) were estimated with appropriate percentages (5…50%) from equipment and mechanical work prices. The investment costs were also evaluated in comparison with other investments executed in Naantali to determine how well they are estimated in relation to known costs, and adjusted when necessary.

Following tax free prices for steam and electricity were utilized to estimate potential utility savings or costs:

- Steam 40 €/MWh
- Electricity 43 €/MWh

The payback time for these investment proposals was determined without any interest rate, because a proper interest rate was not yet known in Finnfeeds Finland Co, caused by the on-going change of company owner.

All of the heat transfer calculations were carried out with the basic formula for heat transfer:

$$\dot{Q} = \dot{m} \cdot c_p \cdot \Delta T, \tag{15}$$

where $\dot{m}$ = mass flow [kg/s], c_p = specific heat capacity [kJ/kgK], and ΔT = temperature difference [K]. With streams that included state change, cp in this formula is replaced with specific latent heat capacity L [kJ/kg] and ΔT is not considered the calculation.

As the specific heat capacities for the process streams (raw materials, intermediary flows, or products) are not known, the reference values of either water (4.19 kJ/kgK) or Thermera R (3 kJ/kgK) were utilized. The densities of the process streams range between approximately 970 kg/m^3 and 1,100 kg/m^3.

For each proposal including a heat pump, the cost for increased electricity utilization was evaluated. A value for the required amount of electricity was required for this and these values were calculated as W_{el} [kW or MW] based on the heat energy $\dot{Q}_1$ [kW or MW] provided by the heat pump:

$$W_{el} = 0.35 \cdot \dot{Q}_1 \tag{16}$$

The size of the pumps was preliminary calculated with assumptions on pressure differences over the pump.

$$dp = \sum p_{loss} + p_{required} + p_{head} - p_{hydrostatic} + \sum p_{loss}, \qquad (17)$$

where p_{loss} = pressure losses caused by equipment, piping, etc. [kPa], $p_{required}$ = required pressure achieved with the pump [kPa], and p_{head} / $p_{hydrostatic}$ = $\rho \cdot g \cdot h$ [kPa] with ρ = density [kg/m^3], g = 9.81 m/s^2 and h = surface level / pump head [m].

The electricity required for pump operation with 10% additional design was calculated from

$$W_{el} = \dot{v} \cdot dp \cdot 1.1, \qquad (18)$$

where $\dot{v}$ = volume flow in [m^3/s] and dp = pressure difference between suction and pressure side in [kPa].

A preliminary location for each new equipment was proposed based on space availabilities in the facility, and the piping routed to each destination were estimated based on a 3D model and the plant layout.

4.1 INTERNAL UTILIZATION

The following preliminary proposals include multiple identified waste heat sources, and they were chosen as suitable heat sinks were identified for their utilization. These waste heat streams include individual product or cooling streams as well as the entire facility's cooling utility network, and they were chosen for further investigation because of their potential temperature level or energy contents. Additionally, the hygienic risks these waste heat source and sink pairs contain was expected to remain in a feasible level.

Condensates' expansion steam contains a relatively large amount of energy in a considerably high temperature, which would make its utilization highly beneficial.

However, no investment proposals were sketched out for its utilization as there are previously made designs for improvements in the operation and utilization of condensates and their expansion steams.

The processes produce a varying amount of waste water, but no preliminary investment proposal was sketched up for its utilization because of the highly varying nature. The main equipment for waste water treatment and storage are located outside relatively far away from potential heat sinks, and the temperature dissipates to the environment from these tanks. There are possibilities to circulate the waste water in the waste water treatment system with a relatively steady flow, but the temperature of this source cannot be controlled so this was thought to decrease the efficient utilization of this source, especially as most likely its utilization would also require an expensive heat pump investment. Additionally, the operation of a heat pump usually requires a relatively stable inflow of energy as the efficient operation range is quite narrow, and this stability is not identified in this waste heat source with the materials that were studied.

4.1.1 UTILIZATION OF WASTE HEAT FROM PRODUCT STREAM TO RAW MATERIAL STREAM

A product stream, which is here named as Product 4, is currently flowing in a temperature of 68 °C from the process into its storage tank without cooling, and the heat energy dissipates into the atmosphere from the storage tank as the liquid is waiting to be loaded into trucks. The amount of this flow is relatively small, only 0.7 m^3/h, but as it is a continuous flow, the potentials for waste heat recovery are theoretically and technically feasible. To maximize this potential, a suitable heat sink with a relatively continuous heat energy demand and a sufficiently low mass flow and initial temperature should be found. Such a flow is the intake of Raw Material 1, which is currently stored in ambient temperature in a tank located outside. The mean temperature of Raw Material is 10 °C as it comes into the process with a volume flow of 1.8 m^3/h.

With following temperature changes *dT* from supply temperature T_0 to goal temperature T_1, approximately 27 kW of waste heat energy can be recovered from Product 4 and utilized to pre-heat Raw Material 1, as presented in Table 3:

Table 3 Initial and target temperatures for heat transfer between Product 4 and Raw Material 1

Flow	T_0 [°C]	T_1 [°C]	dT [°C]	$\dot{v}$ [m^3/h]
Product 4	68	35	33	0.7
Raw Material 1	10	21	11	1.8

This pre-heating step can be carried out with a new heat exchanger, which can be located in the vicinity of the current routes of both stream's pipelines. Instruments such as temperature gauges and automatic valves are required for easy operation, but all in all, this investment proposition requires relatively little to be carried out.

Price of the new heat exchanger	3,800 €
Piping and mechanical works	7,500 €
EIA and other investment costs	5,700 €

To estimate the possible savings from steam usage, the heating of Raw Material 1 must be studied in its entirety. After the pre-heating of Raw Material 1, the flow will need to be heated up to 80 °C with the existing heat exchange equipment. Currently there are five heat exchangers coupled in the same hot water circulation, which is heated up with a steam heat exchanger. A simple flow scheme of this pre-heating system is presented in Figure 25 as well as in the flow sheet in Appendix 3. While studying the potential savings for this proposal it was noticed that currently the third heat exchanger HE-1211 is cooling the raw material, transferring the heat energy from the raw material back to the water circulation. This is a remnant of an old operational model, and currently there is no need to utilize this third heat exchanger for cooling purposes.

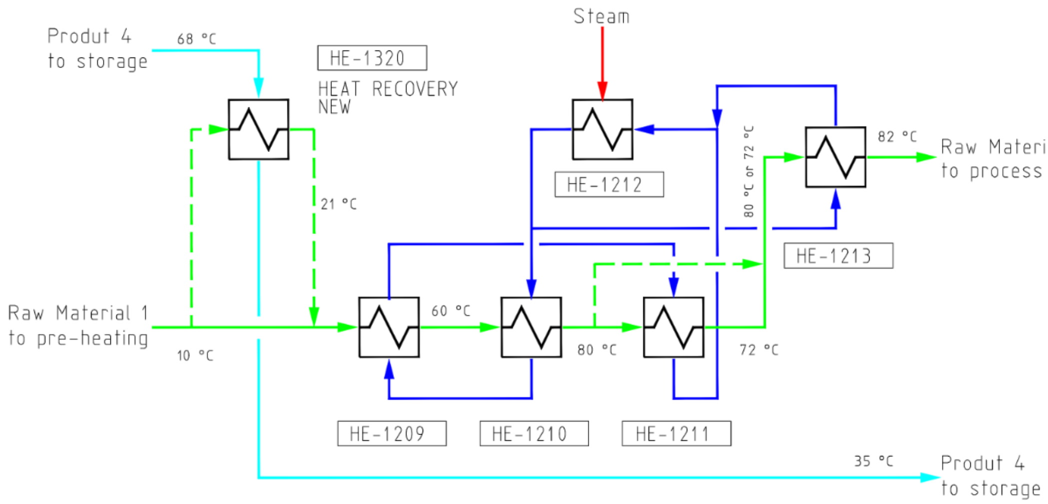

Figure 25 The basic principle for pre-heating of Raw Material 1, including the proposed waste heat recovery and bypassing of HE-1211 (in dashed lines). The bypassing of HE-1211 is here illustrated with a line, even though in reality this can be implemented with automation and valves and without any new pipelines.

If the heat exchanger HE-1211 is bypassed, steam consumption is expected to slightly increase in order to replace the energy that was formerly provided to the water circulation by the heat exchanger. Additionally, as the temperature of Raw Material 1 is higher as it comes to the fourth heat exchanger, the duty for this heat exchanger is expected to decrease, thus decreasing the total heat energy demand of the hot water circulation. The flow sheet in Appendix 3 for this investment proposal is sketched according to heat exchanger duties when HE-1211 is bypassed. Different operational models for heating up Raw Material 1 are evaluated in Table 4.

Table 4 Comparison of steam expenses for alternative operational models for heating of Raw Material 1

Flow	*Q* required [kW]	*Q* to water [kW]	Steam expenses [€/a]	Savings from steam expenses [€/a]
Current operating model	191	-20	34,900	
Bypassing HE-1211	177	0	35,900	-1,000
Pre-heating Raw Material 1 and bypassing HE-1211	150	0	30,500	4,500
Pre-heating Raw Material 1 without bypassing HE-1211	165	-20	29,500	5,500

The viability of the investment proposal can be evaluated by either comparing it to the current situation or to the situation where HE-1211 is bypassed.

Compared to current operating model		
Savings from steam expenses	**5,500**	**€/a**
Investment costs, total	**17,000**	**€**
Payback time	**3.1**	**a**

Pre-heating Raw Material 1 and bypassing HE-1211		
Savings from steam expenses	**4,500**	**€/a**
Investment costs, total	**17,000**	**€**
Payback time	**3.8**	**a**

The potential savings are in a moderate level and there is no significant difference to payback time between these two options. However, the option including bypassing of HE-1211 is recommended as there is no real reason to utilize this heat exchanger as a cooler. The payback time of 3.8 years for an investment of this size is slightly long but remains within economic feasibility nonetheless.

4.1.2 WASTE HEAT RECOVERY FROM DRIER EXHAUST AIR

It is very common for property ventilating units to have heat recovery with a heat pump between the exhaust air and incoming air channels, and a similar principle could be applied to driers D1 and D3, as both of them take in air in ambient temperature and currently utilize steam to heat it up, releasing a warm exhaust air flow. Both driers utilize cold air as well as hot air, which means that heat recovery between the two incoming air channels could also be a possibility, if a heat pump would be utilized to cool down the incoming cool air, and utilize this heat energy in an increased temperature to warm up the incoming hot air. Despite the excellent theoretical potential of this option, it is not easily carried out as the cooling demand for cold air is regulated based on the moisture content in the air, and this varies significantly based on ambient weather conditions as well as seasonal changes.

Another option for utilization of waste heat from drier's exhaust air is to circulate a heat transfer liquid between the exhaust air and incoming (hot) air channel. This requires heat exchanger units in both air channels, which leads to a decreased efficiency in total heat transfer. Also, this is a passive means of waste heat utilization, meaning that the temperature in the exhaust air is a limiting factor for increasing the temperature in the incoming air side, and resulting in continuing utilization of steam in order to increase the temperature in the required level.

4.1.2.1 DRIER D1

Drier D1 utilizes hot air in 85 °C temperature with a volume flow of 6,500 m^3/h, and the same amount of dried cold air in approximately 30 °C. Both incoming air flows have individual channels with necessary heat exchangers and other equipment. The hot air is heated up with two heat exchangers, the first utilizing hot water and the second increasing the temperature into the desired level with steam. The energy needed for heating the hot air is approximately 164 kW. When the drier is in operation, there are no possibilities for sequential operation, resulting in a stable operation of air flow without taking into account whether there is a process flow to be dried or not. The air flow exits the drier through a dust removal cyclone, and the calculated temperature of this exhaust air is calculated to be approximately 50 °C.

If the exhaust air is cooled down from 50 °C to 5 °C, 185 kW of heat energy could be utilized. The optimal location of waste heat recovery would be from the channel after the dust removal cyclone, as then the air has been purified of dust particles. The dust cyclone is in the vicinity of the drier and its incoming air channels, which enables heat transfer between these two flows with either a heat pump or with a direct heat transfer liquid circulation. The amount of potentially recoverable heat energy would be enough to satisfy the heating requirement in total if the temperature would be increased, which would support the idea for utilizing a heat pump as shown in Table 5. Preliminary investment proposals were calculated for both of these options for comparison.

OPTION 1 includes a heat pump installed in the same room as the drier and all of its channels. The evaporator of the heat pump would be installed in the exhaust air channel and the condenser in the incoming hot air channel. As there are very little possibilities to route the channels differently compared to current layout because of space constraints, it would be highly probable that the different parts of the heat pump would have to be installed apart from each other, which most likely would make it more difficult to find a suitable heat pump for this application.

Table 5 Current and proposed heat exchangers for heat recovery with a heat pump. The table shows each streams' initial temperature T_0 and target temperature T_1 as well as heat load and annual expenses from steam consumption.

Heat exchanger	T_0 [°C]	T_1 [°C]	Q [kW]	Steam consumption [€/a]
Current operation	5	85	164	34,200
Cooling of exhaust air	50	5	185	-
Air – Heat pump	5	85	164	-

The heat pump is expected to provide the heat energy in a temperature of 95 °C, which would be sufficient to enable hot air to be fed into the drier at 85 °C. The existing air-water heat exchanger would be demolished, but the air-steam heat exchanger would remain in the channel as an additional or optional heat exchange unit.

Current costs for steam	34,200 €/a
Price of the heat pump	44,400 €
Mechanical works	13,400 €
EIA and other investment costs	8,900 €
Savings from steam	**34,200 €/a**
Expenses from electricity (heat pump)	**13,100 €/a**
Investment costs, total	**66,700 €**
Payback time	**3.2 a**

The value for COP_{th} is:

$$COP_{th} = \frac{T_1}{T_1 - T_0} = \frac{273\ °C + 95\ °C}{(273\ °C + 95\ °C) - (273\ °C + 50\ °C)} = \frac{368\ K}{368\ K - 323\ K} = 8.2$$

Based on the estimated electricity consumption, COP_{el} can be calculated:

$$COP_{el} = \frac{\dot{Q}_1}{W_{el}} = \frac{164\ kW}{58\ kW} = 2.9$$

The value for COP_{th} is quite moderate and the value for COP_{el} is slightly lower than what it would be expected to be. However, these calculations are based on estimations, so the possibility of finding a heat pump with a desired level of $COP_{el} > 3$ is realistic.

OPTION 2 for D1 would be to carry out heat transfer with a liquid circulation between the exhaust air and incoming hot air channels. This option has been calculated in a simplified scheme, where the temperature of the water circulation is always either 10 °C above or below the air temperature on the other side of each heat exchanger. A heat exchanger would have to installed in the exhaust air channel, but at the incoming air side, the same heat exchanger that is currently utilized for pre-heating the air with water could be utilized in this option. A water or other heat transfer liquid circulation loop with a circulation pump would be routed between these two heat exchangers.

The temperature difference over the current air-water heat exchanger is not known, but with assumptions for the operation of this option presented in Table 6, a total of approximately 10,700 € could be saved from steam costs annually. In the current operation, the water in the air-water heat exchanger is also heated with steam, hence the savings in steam expenses when this water would be replaced with an internal circulation containing waste heat energy. The temperature of the incoming air in this scenario is expected to be 5 °C, and it would be heated up to 30 °C with the water circulation, and to 85 °C with the existing steam heat exchanger.

Table 6 Current and proposed heat exchangers for heat recovery with a heat transfer circulation. The table shows each streams' initial temperature T_0 and target temperature T_1 as well as heat load and annual expenses from steam consumption

Heat exchanger	T_0 [°C]	T_1 [°C]	Q [kW]	Steam consumption [€/a]
Current operation	5	85	164	34,200
Cooling of exhaust air	50	5	185	-
Air- Water circulation	5	30	51	-
Air- Steam	30	85	113	23,500

The water circulation temperature would change between 15 °C and 40 °C, and based on the temperature change and amount of waste heat energy available, the circulation volume flow would be 5.1 m^3/h. With an estimated pressure difference of 300 kPa, the circulation pump would require a power of 0.5 kW for its operation.

Current steam expenses	34,200	€/a
Price for a heat exchanger	3,600	€
Mechanical works	18,600	€
Price for a pump	1,400	€
EIA and other investment costs	3,700	€
Savings from steam expenses	**10,700**	**€/a**
Expenses from pump electricity	**100**	**€/a**
Investment costs, total	**27,300**	**€**
Payback time	**2.6**	**a**

The payback times for both of these options are on an economically feasible level. The potential savings, heat transfer efficiency and investment costs go hand in hand in these options so choosing between these options will require a comprehensive evaluation process.

4.1.2.2 DRIER D3

The amount of air fed into drier D3 varies depending on which product it is drying, but the temperatures remain the same. D3 utilizes dried cold air in a temperature of approximately 20 °C and hot air in a temperature of 110 °C. All of the air is taken into the process with a unified channel and heated up with a dT 5 °C from ambient temperature, which in this calculation was assumed to be 5 °C. The air is filtrated and divided into hot and cold air in their individual channels. The hot air is heated up with steam, and also the pre-heating of the whole air intake utilizes steam in its heat exchanger. With Product 1, approximately 3,040 m^3/h of cold air and 4,000 m^3/h of hot air is required in the drier. For Product 3, these figures are 1,600 m^3/h and 1,200 m^3/h, respectively. The drier is operated in a sequential mode, which means that when there is no product to be dried, the air inflow will decrease significantly to approximately 30 % flows.

The exhaust air from D3 goes through a dust removal cyclone and into a wet scrubber. The exhaust air exits from the dust cyclone and the temperature is assumed to be relatively low in this exhaust air because of the steps of dust removal and because of the relatively long distances between the dust removal equipment and the drier. Because of the constraints from the temperature and the distance, the waste heat recovery is proposed to be taken from the exhaust air before it is removed from dust. This is not the optimal location to install a heat exchanger because of potential hygienic and operational risks caused by the dust particles, so careful evaluation of the risk to benefit ratio would be required, as well as determining the required cleaning and inspection equipment for this heat exchanger in order to ensure safe and clean operation.

The temperature of the dust containing exhaust air from D3 is calculated to be approximately 60 °C, and if the temperature of the exhaust air from D3 would be decreased to 10 °C, a total of 111 kW of waste heat energy could be recovered when Product 1 is dried with this drier. The pre-heating of the total air flow requires 11 kW and the heating of the hot air flow requires 121 kW, so the waste heat energy is not sufficient to satisfy all of the energy needed for heating up hot air. Because steam is required in any case as a hot utility, a heat pump is not thought to be a viable option for this waste heat recovery and instead a water circulation similar to that of the other dried D1 is proposed as a means to recover waste heat from the exhaust air and utilize it to heat up the total air intake as well as heat up the hot air from 10 °C to 40 °C. These heat exchange unit operations are presented in Table 7.

Table 7 The required change in temperature, heat loads and steam expenses in current and proposed operational models

Heat exchanger	*dT* [°C]	*Q* [kW]		Steam consumption [€/a]
		Product 1	Product 3	
Current operation, pre-heating of total air intake	5	11	4	2,200
Current operation, heating of hot air	100	121	36	31,300
Water circulation, pre-heating of total air intake	5	11	4	-
Water circulation, pre-heating of hot air	30	38	10	-
Steam, heating of hot air	80	89	26	20,900

Existing heat exchangers could be utilized in both pre-heating of total air intake and pre-heating of hot air. A new heat exchanger could be installed in the exhaust air channel in the vicinity of the air intake channels, and a water circulation with a circulation pump would be routed in between these devices. To estimate the amount of flow in the water circulation, the water temperature is assumed to vary between 15 °C and 50 °C, always remaining 10 °C above or below the air temperature on the other side of each heat exchanger. Based on these temperatures and the heat load from waste heat, the amount of flow in the water circulation can be calculated to 2.2 m^3/h. With a pressure difference of 300 kPa, the circulation pump would require 0.2 kW of power for operation.

Price of the heat exchanger	3,400	€
Mechanical works	5,700	€
Price of the pump	1,200	€
EIA and other investment costs	3,100	€
Savings from steam expenses	**12,600**	**€/a**
Expenses from pump electricity	**100**	**€/a**
Investment costs, total	**13,400**	**€**
Payback time	**1.1**	**a**

The payback time for this investment proposal is calculated based on above-mentioned preliminary estimations on investment costs and potential savings and the payback time appears to be relatively short making this proposal economically feasible. However, as the proposed point of heat recovery from the exhaust air channel is not ideal, the technical feasibility should be carefully considered when making a decision about whether to carry on with this investment proposal.

4.1.3 EVAPORATOR SECONDARY STEAM ENERGY UTILIZATION TO CENTRAL HEATING OF PROPERTIES

The central heating network to heat up the properties is utilizing steam to heat up the water circulating in the radiators. Coupled into this network is a heat exchanger HE-1237, that used secondary steam from an evaporator instead of steam, but this heat exchanger is no longer in use as the evaporator it is coupled into is no longer operated. However, there is another evaporator whose secondary steam could be utilized, reducing the steam expenses significantly in the central heating.

The amount of secondary steam available from this evaporator is approximately 1,800 kg/h, and an additional flow condensate expansion steam 290 kg/h is also directed into this same pipeline. The temperature of this flow is not known, but according to surface temperature measurements, it is approximately 70 °C. Currently this flow is condensed with a surface condenser, thus, creating a vacuum in the evaporator. This phase change alone releases 1,315 kW of energy.

According to measurements made in February, there is a flow of 67,000 kg/h in the central heating network, and the heating demands are dependent on outside temperatures. In this proposal, a temperature difference dT of 10 °C was utilized for the water flow in the central heating network. 8 °C of this temperature difference would be satisfied by the energy in the secondary steam, and the remaining 2 °C would be heated with steam, as shown in Table 8. This dT would not cause the secondary steam to condense, as it the energy released in this heat exchanger would be smaller than what is needed for the phase change to occur. In addition, the potential savings from steam were calculated in such a way that the evaporator would not be used 20% of the time and steam would be required to satisfy the total heating demand.

Table 8 Heat loads and steam expenses for current operation and proposed operation model in central heating

Heat exchanger	dT [°C]	Q [kW]	Steam consumption [€/a]
Current operation	10	780	238,000
Secondary steam, HE-1237	8	624	-
Steam, HE-1163	2	156	85,700

The potentials for savings from steam expenses are quite significant; with this calculation, a total of 152,300 € could be saved annually when this operational model is in use when the factory is in production. However, this calculation does not take into consideration the seasonal changes in the central heating demands. This calculation is based on the heating demand measured during February, so naturally the heating demand decreases with increased ambient temperatures.

In order to utilize the secondary and expansion steams from the evaporator that is in operation, the existing secondary steam lines from the old evaporator to heat exchanger HE-1237 should be demolished and the heat exchanger should be relocated in an empty location near the pipelines that are carrying the current flow of secondary steam to the surface condenser. The pipelines would have to be slightly rerouted to the HE-

1237, both in the secondary steam side as well as in the central heating water circulation side. Because of the relatively big pipeline sizes on both sides, the costs for mechanical works would be quite significant, but on the other hand, no equipment purchases are necessary. The HE-1237 heat exchanger is expected to be in good condition, but it should be inspected before commissioning.

Mechanical works and relocating HE-1237	77,500	€
HE-1237 inspection	2,000	€
Demolitions	3,900	€
EIA and other investment costs	3,900	€
Savings from steam expenses	**152,300**	**€/a**
Investment costs, total	**87,300**	**€**
Payback time	**0.6**	**a**

The payback time for this investment proposal is very short making this investment proposal economically feasible. Provided that the existing heat exchanger is in good condition and could be relocated, no technical barriers were identified for this proposal.

4.1.4 Heat recovery from an individual cooling water stream

There are currently two parallel tanks where Process Stream 18 is cooled down with an individual Cooling Water stream 1. This cooling is currently carried out with the plant's cooling water system, but as this is an individual circulation of water that is flowing in the mantle of the tanks, it could be possible to recover the heat energy to a cold process stream. Such a stream is for example the intake of Raw Material 2, which is taken in to the process in an estimated average temperature of 5 °C.

The cooling water circulation (CWS1) has an estimated flow of 3.5 m^3/h and it needs to be cooled down from 75 °C to approximately 40 °C, resulting in a cooling demand of 138 kW. The intake of RM2 is divided to two different destinations in the process with flowrates of 0.4 m^3/h and 2.7 m^3/h. With a combined flowrate of 3.1 m^3/h, the intake of RM2 could be heated up to approximately 38 °C with heat transfer from CWS1, as shown in Table 9.

Table 9 Heat transfer temperatures between CWS1 and RM2

Flow	T_0 [°C]	T_1 [°C]	dT [°C]	$\dot{v}$ [m³/h]
Cooling water circulation 1	75	40	35	3.5
Raw Material 2	5	38	33	3.1

Although this is not enough to replace steam utilization in heating of RM1 in its entirety, it would be able to decrease it in following process steps. In these following steps, the RM1 is mixed into other flows before increasing the temperature, so the benefits from the increased temperature of RM1 will have to be evaluated based on the increased initial temperature of these combined streams. Table 10 presents the current situation for these combined streams as well as how the initial temperature would be increased with this proposed waste heat recovery.

Table 10 Initial and target temperatures and heat load for current operational model as well as proposed operational model

	T_0 [°C]	T_1 [°C]	dT [°C]	$\dot{v}$ [m³/h]	Q [kW]	Steam savings [€/a]
Current situation, heating of PS17	37.5	85	48.5	10.6	458	-
Heating of PS17, if RM1 stream pre-heated	44.7	85	40.3	10.6	381	3,800
Current situation, heating of PS19	50.5	85	34.5	3.5	150	-
Heating of PS19, if RM1 stream pre-heated	54.3	85	30.7	3.5	129	3,400

In addition to savings from steam expenses, there is a possibility to obtain savings from reducing the volume flow in the plant's cooling water network. If the amount of water that is pumped from the sea to the cooling water system is reduced, the electricity the cooling water pump requires for its operation will decrease.

$$W_{el,current} = \dot{v} \cdot dp \cdot 1.1 = 350\ \frac{\mathrm{m^3}}{\mathrm{h}} \cdot 450\ \mathrm{kPa}\ \cdot 1.1 = 0.097\ \frac{\mathrm{m^3}}{\mathrm{s}} \cdot 450\ \mathrm{kPa}\ \cdot 1.1 = 43.8\ \mathrm{kPa}$$

$$W_{el,decreased} = \dot{v} \cdot dp \cdot 1.1 = (350 - 16.7)\ \frac{\mathrm{m^3}}{\mathrm{h}} \cdot 450\ \mathrm{kPa} \cdot 1.1 = 0.093\ \frac{\mathrm{m^3}}{\mathrm{s}} \cdot 450\ \mathrm{kPa} \cdot 1.1 = 41.7\ \mathrm{kPa}$$

Based on these calculations, the current electricity consumption in approximately 14,600 €/a and with a decreased pumping volume, the electricity consumption would be approximately 12,100 €/a.

This waste heat recovery could be carried out with an additional heat exchanger, which could be located in the same room as the equipment that are currently cooling down the process tanks. The pipeline that is used to take in RM2 is also already routed in that same room, so only a moderate amount of new piping is required to route the cooling water from the existing circulation to a new heat exchanger, and return it to the same location. The existing heat exchangers in each tank's cooling circulation would remain in operation, as the intake of RM2 is non-continuous and there might be a need to utilize the existing equipment as a backup or to accomplish additional cooling.

Price of heat exchanger	3,700	€
Mechanical works	22,600	€
EIA and other investment costs	6,800	€
Savings from steam expenses	**7,200**	**€**
Savings from pumping electricity	**2,500**	**€**
Investment costs, total	**33,100**	**€**
Payback time	**3.4**	**a**

With a payback time of 3.4 years, this investment proposal appears to still be within economic feasibility. If the potential savings from cooling water pumping electricity are not taken into consideration, the payback time naturally increases to 4.6 years, which is close to being economically unfeasible.

4.1.5 WASTE HEAT RECOVERY FROM COOLING WATER WITH A HEAT PUMP

As mentioned before, a relatively low temperature is common to most of the waste heat sources found in Finnfeeds Finland Naantali processes. These relatively low temperature sources are tough to utilize as such, as most of the streams requiring heating must be heated into a relatively high temperature. The waste heat from low temperature sources can be utilized in these heat sinks with the help of a heat pump, and in this thesis, the waste heat source is proposed to be the cooling water system, which utilizes sea water and returns it back to the sea in an elevated temperature. Even though the temperature in this stream is low, approximately 10…20 °C, it contains a large amount of energy because of its large flow, 350 m^3/h on average.

Another option would be to directly recover waste heat by condensing secondary steams from crystallization or evaporation, as these are the biggest unit processes to create a large amount of waste heat. All of this energy is now gathered into the cooling water in a low temperature, but obtaining the energy directly from the secondary steam would provide for a higher temperature in the recovered waste heat. This could be executed either directly or with a heat pump, and the waste heat sink should also be a hot water circulation with a properly dimensioned buffer tank. As all of the crystallization unit operations are operated batch-wise, this would mean that also the waste heat source would not be continuous, but the waste heat sink could be operated continuously if the hot water in it would be stored in a buffer tank. A comparison of the advantages and disadvantages of these two options are presented in Table 11.

Table 11 Comparison of cooling water and secondary steam as a source of waste heat to a heat pump

Waste heat source	Advantages	Disadvantages
Cooling water	- Collects energy from multiple sources - Heat pump operation probably more stable - Heat pump operation probably with a higher utilization rate	- Low source temperature - Lower COP for heat pump caused by low temperature of waste heat - Waste heat recovery not reasonably if the temperature of cooling water from processes remains too low (no cooling in processes)
Secondary steam	- Higher source temperature - Higher COP for heat pump caused by higher temperature of waste heat - Recovering of latent energy directly from its source	- Recovery possible only from individual / specific waste heat sources: the amount of waste heat recovered may be smaller - A buffer tank is required to stabilize the availability of hot water

Besides these two options, it could also be possible to gather energy from multiple sources into a same heat sink, thus, stabilizing the amount of waste heat energy provided to the circulation. However, this would require multiple heat pumps which would increase both the investment and operational costs.

The amount of waste heat energy recovered from the cooling water returning from the processes is estimated with an average flow of 350 m^3/h and by decreasing the temperature in it from 15 °C to 7 °C, thus, releasing approximately 3 MW of energy. This could be recovered with a heat pump and used to heat up a hot water circulation to a temperature of 90 °C. The hot water circulation would be used in internal heating demands, and the surplus of energy could be sold to district heating. After all of these internal and external uses, the temperature in the hot water circulation would have decreased to 60 °C as it returns to the heat pump. The amount of flow in the hot water circulation, as well as the amount of hot water provided into the district heating network, were calculated based on the chosen temperature differences and heat loads. The basic principle of this proposed waste heat recovery and utilization is illustrated in Figure 26.

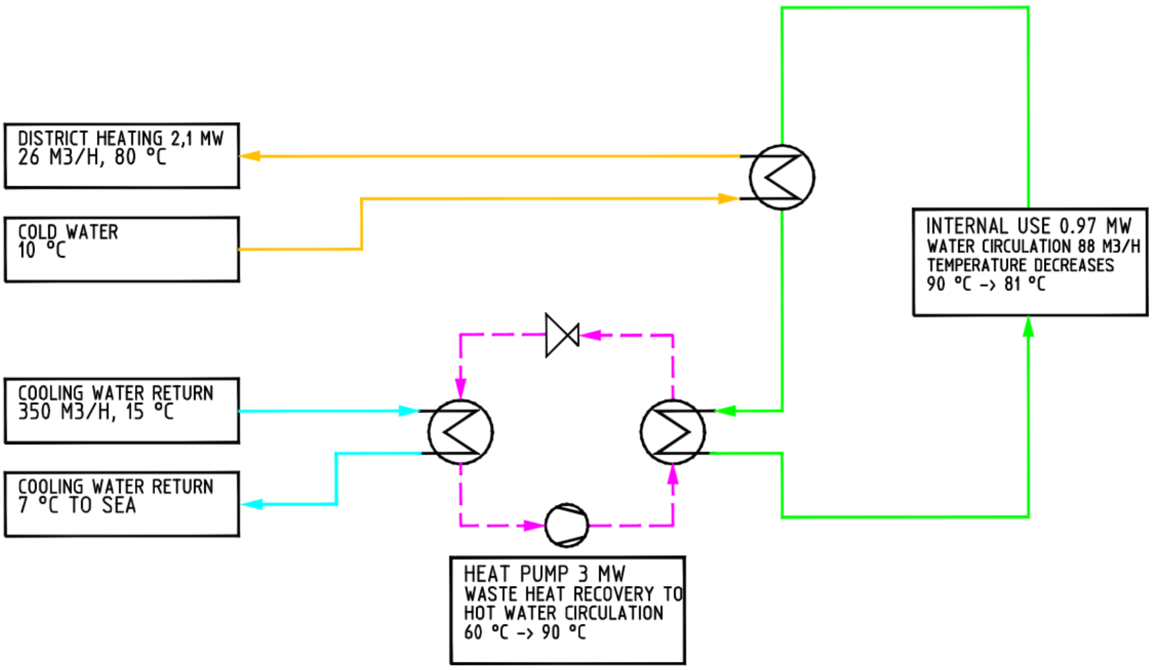

Figure 26 An illustration of the basic principle for the heat pump investment proposal with internal and external utilization.

Internal uses are listed in Table 12. T_0 is the initial temperature of the flow and T_{target} is the target temperature determined by process requirements. As it is not possible to increase the temperature as high as T_{target} in all of these uses, T_1 illustrates the temperature reached with heat transfer from the hot water circulation.

Approximately half of these flows could utilize their existing heat exchangers if the hot water circulation would be routed to these locations, replacing steam usage in its entirety. The other half would require new heat exchangers to be purchased and installed in the vicinity of the current locations of these flows. A few of these flows would still require additional heating with steam by their existing heat transfer equipment.

Some of these flows are continuous, but some are operated occasionally based on what product is being made in the processes. This leads to the amount of internally used energy variating according to production demands, but the heat pump could nevertheless be operated in a stable manner, as the amount of heat energy provided to district heating could fluctuate. The controllability of a heat pump works on a relatively narrow range, so adjusting the amount going to district heating according to internal demands would prevent problems arising from unstable operation of heat pump.

Table 12 The internal flows to utilize heat transfer from the hot water circulation. The last column shows which streams can utilize existing heat exchangers and which require a new one

Flow to be heated	T_0 [°C]	T_{target} [°C]	T_1 [°C]	$\dot{v}$ [m^3/h]	Q [kW]	New / existing
Process stream 10	72	80	80	25.0	167	Existing
Process stream 11	60	85	70	4.7	56	New
Process stream 12	78	80	80	6.1	14	Existing
Process stream 13	79	80	80	2.1	3	Existing
Raw material 1 intake	10	80	80	1.8	160	Existing
Process stream 15	70	75	75	0.9	6	Existing
Process stream 14	75	80	80	26.8	156	New
Process stream 16	75	94	82	11.0	69	New
Process stream 17	37	81	73	10.6	340	New

The pipeline for cooling water that returns from the process flows through the facility floors and the heat pump and its auxiliary equipment should be located in the vicinity of this existing route. By demolishing an existing old CIP tank and replacing it with pipelines from another CIP tank, and relocating a pump as well as some pipelines, a clear space could be generated right next the cooling water pipeline. The costs for these demolitions and replacements are considered in the investment proposal.

The prices for all of the equipment that are proposed to be purchased are calculated based on AFRY's price database. The sizes of the pumps are estimated based on preliminary evaluations for pressure differences. The power of the hot water circulation pump is estimated to be 20 kW and the power of the district heating pump is estimated to be either 10 kW or 19 kW depending on the coupling point. In addition to heat exchangers and pumps, the hot water circulation should have at least a small expansion tank, even though there would not be a demand for a bigger buffer tank in the circulation.

Implementing this waste heat recovery would generate significant savings from steam expenses. The amount of the savings can, in a simplified manner, be calculated directly from the sum of the internal heat loads listed above in Table 12. However, as all of these flows and their heat exchangers are not in continuous operation, the estimate for steam savings should be evaluated by taking into consideration an evaluation on the annual production plans for each product, which affects directly to the operation of each heat exchanger. The difference of these two ways to evaluate steam savings is approximately 20,000 €/a. The savings from steam are estimated based on the production plan when calculating the payback time for this investment proposal.

There are two alternative options for the location where the water provided to district heating could be coupled into, and both of these options are taken into consideration in the investment calculations. Cooperation with a district heating company is presented further in Chapter 4.2.

	Option 1		Option 2	
Price of heat pump	829,500	€	829,500	€
Price of heat exchangers, total	21,700	€	21,700	€
Price of pumps, total	19,700	€	27,600	€
Price of expansion tank	5,000	€	5,000	€
Mechanical works	197,500	€	296,800	€
Demolition works	30,000	€	30,000	€
Automation and instrumentation investment costs	29,700	€	44,600	€
Electricity investment costs	29,700	€	44,600	€
Savings from steam	**276,000**	**€/a**	**276,000**	**€/a**
Income from district heating	**259,100**	**€/a**	**259,100**	**€/a**
Electricity expenses from heat pump and pumping	**369,200**	**€/a**	**373,000**	**€/a**
Investment costs, total	**1,162,800**	**€**	**1,299,800**	**€**
Payback time	**7.1**	**a**	**8.0**	**a**

As the temperature of the waste heat (cooling water) provided for the heat pump's evaporator, and the temperature of the hot water leaving the heat pump's condenser differ significantly from each other, COP_{th} remains relatively low:

$$COP_{th} = \frac{T_1}{T_1 - T_0} = \frac{273\ °C + 90\ °C}{(273\ °C + 90\ °C) - (273\ °C + 15\ °C)} = \frac{363\ K}{363\ K - 288\ K} = 4.8$$

COP_{el} can be calculated based on the estimated electricity consumption of the heat pump, and also this value is not as high as it should be:

$$COP_{el} = \frac{\dot{Q}_1}{W_{el}} = \frac{3\ MW}{1.1\ MW} = 2.9$$

As these calculations are based on estimations and not calculated for a specific heat pump or its properties, $COP_{el} > 3$ should be well within reach for this application.

If there is no interest in cooperating with a district heating company by providing excess waste heat energy to their district heating system, a smaller heat pump would suffice for internal use. With similar additions, modifications and estimations, the principle of this smaller 1 MW heat pump would be as illustrated in Figure 27.

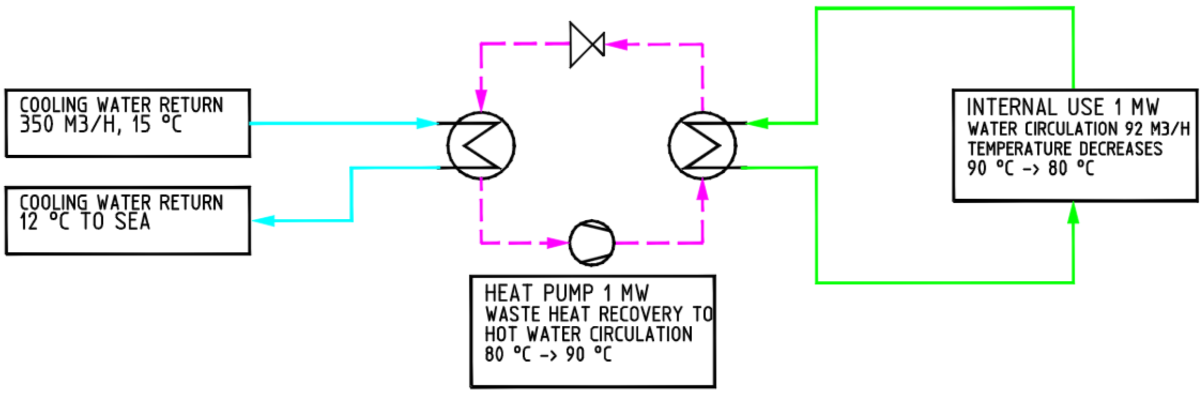

Figure 27 An illustration of the basic principle for waste heat recovery with a heat pump for internal utilization.

This investment would require a significantly smaller investment, but there would be no additional income from district heating and the whole waste heat energy potential in the cooling water would not be utilized.

Price of heat pump	261,800	€
Price of heat exchangers, total	15,300	€
Price of pump	13,800	€
Price of expansion tank	5,000	€
Mechanical works	165,500	€
Demolition works	30,000	€
Automation and instrumentation investment costs	24,900	€
Electricity investment costs	24,900	€
Savings from steam expenses	**276,000**	**€/a**
Electricity expenses from heat pump and pumping	**119,580**	**€/a**
Investment costs, total	**541,200**	**€**
Payback time	**3.5**	**a**

The payback time for the options including district heating is relatively long and not necessarily within the traditional limits of economic feasibility. The feasibility of an energy efficiency investment is not solely reliant on the duration of the payback time, so this preliminary investment proposal should be considered whether it would be otherwise viable and there would be willingness for such new cooperation. To improve its feasibility, more internal utilization should be incorporated to the same hot water circulation, thus, increasing also the size of the heat pump. The same approach could also be applied to the option for internal use only, as increasing the utilization of the energy in the hot water circulation would create more savings from steam consumption and decrease the investment's payback time.

The internal process streams chosen to be implemented in this investment proposal were chosen based on their temperature profiles, volume flows, and current location in the process. More streams that could be coupled into a hot water circulation can be found from the Naantali plant, so in order of finding the ideal approach for a heat pump investment, the places where the energy would be utilized would have to be carefully chosen in order to determine the ideal amount of heat recovery.

4.2 EXTERNAL UTILIZATION AS DISTRICT HEAT

The possibility of delivering waste heat to the district heating system was preliminarily examined in this case study. The district heating system of the city of Naantali is in the near proximity of the Finnfeeds Finland Co. plant, and the prerequisites for supplying heat energy into this network was examined in cooperation with both Turku Energia Co. and Turun Seudun Energiantuotanto Co. They have a long history with similar waste heat cooperation with external parties dating back to the 1990s, and were interested in this possibility of finding a mutually beneficial operational model.

As it has been previously mentioned, the amount of waste heat produced in the Finnfeeds Finland Naantali plant's processes are heavily dependable on the production plans and production phases. Different alternatives of waste heat sources were discussed with Turku Energia and Turun Seudun Energiantuotanto, including secondary steam and cooling water.

It was concluded that most probably the most effective and effortless way for this cooperation would be achieved with an individual water flow provided with an indirect coupling to the network, and this water flow would be heated to a sufficient temperature by the waste heat source. This is not necessarily the advisable way to couple a waste heat source to the district heating network, as the coupling should always be conducted indirectly in order to avoid potential impurities in the network water. The equipment (heat pump, heat exchanger, and pump) and piping necessary for this coupling would be owned, operated and maintained by Finnfeeds Finland.

In the preliminary discussions, a couple of alternative coupling points to the network were identified and the design values for these points were determined. The temperature of the supplied water is dependable on ambient temperature and the pressure in both of these points were approximately either 250 kPa or 450 kPa. The first option would be a point from where district heating goes to the centre of Naantali. This location is approximately 500 meters from the Finnfeeds Finland plant, it has a lower pressure in the network and the heat load provided by Finnfeeds Finland would be utilized relatively quickly so there would not be any significant heat losses caused by long distances. The other option for a possible coupling point is in the vicinity of the power plant which is the primary heat source to the district heating. This point is approximately 1.2 km from the Finnfeeds Finland plant and has a larger pressure in the network, but the estimated supply temperature of approximately 80 °C would better dissipate to the network in cold ambient temperatures.

As the amount of heat energy that could be provided to the district heating system by Finnfeeds Finland is relatively small compared to the total energy in the network, especially during high demand operation, its varying nature would not cause problems in the network operation and there would not be a need to operate this district heating coupling with a strictly steady heat flow. The preliminary estimated temperature of 80 °C was estimated to be sufficient until ambient temperature decreases to a few degrees Celsius below zero, but a higher supply temperature would be desirable during colder weather, especially if the coupling point would be located in the first option.

The demand for district heating is heavily dependable on outside temperature, which means that there are also great seasonal differences. The district heating network supply temperature must be higher during winter and the demand for the heating is significantly higher then, and this also reflects to the prices the district heating company would be willing to pay for waste heat. If sufficient supply temperature is not met, the price of the waste heat provided to the district heating is lower. The investment calculations were calculated with average prices taking into account the seasonal changes.

5 Conclusions

The relatively low temperature of different waste heat sources as well as their relatively varying production rates complicates the efficient utilization of this heat energy, especially as there are both continuous and batch-wise unit processes and heat energy demands in the Finnfeeds Finland Co. Naantali factory.

Heat storage would be necessary to stabilize the supply of waste heat energy to heat sinks, but the low temperature in the sources causes storage as such to be inefficient and pointless. The temperature would have to be increased prior to storing in order to provide the heat energy in a temperature that would be sufficient for multiple utilizations and that would be able to heat up process streams to a relatively high temperature. Without a sufficient temperature level, the investment costs are expected to be too high compared to the profits obtained from this heat recovery.

The relatively low temperature of waste heat sources also prohibits the use of internal electricity production with Organic Rankine Cycle, as no sufficiently high (both in terms of temperature and flow) waste heat sources were identified.

The applicable means for heat recovery with these waste heat sources are heat exchangers and increasing the temperature with a heat pump prior to utilization. The aforementioned investment proposals were sketched up to provide alternatives for heat recovery and to reduce the amount of heat exchangers that are operated over the Pinch point in different scenarios. The investment proposals aimed at reduction of steam utilization as well as increase of process integration in suitable targets to improve energy efficiency and waste heat utilization. Scenario 1 included the most of such heat exchangers (16 out of 50) and out of these heat exchangers, 9 are included in the investment proposals for waste heat utilization. As 16 out of 50 heat exchangers represented 37% of the heat energy demands in Scenario 1, these 9 heat exchangers included in the preliminary investment proposals would be able to change into waste heat instead of steam as an energy source, this waste heat recovery representing approximately 9% of the scenarios heating and cooling demands. However, some of them would still be operated over the Pinch point if these investment proposals would be implemented, as they would be included in a hot water circulation creating a situation where the heat transfer media would be on the wrong side of the Pinch point compared to the heat sink. For scenarios 2 and 3, the number of heat exchangers that are currently operated over the Pinch point and that are included in the investment proposals are 1 and 3, respectively.

The payback times for the preliminary investment proposals were quite varying, but the payback time is not the only criteria to evaluate an investment’s feasibility, especially with energy efficiency investments. Utilization of waste heat provides the factory means to improve their sustainability with increased energy efficiency. All of these preliminary proposals should be assessed in relation to one another in order to determine whether or not they could be implemented and in which order. Especially the ones with short payback times (such as the waste heat recovery to central heating) or significant potential (waste heat recovery with a heat pump) are advised to be prioritized in the potential implementation order. Additionally, any means for process integration and simultaneous reduction in steam expenses are strongly advised, especially by implementing heat recovery technologies to the heating of raw material flows.

While studying the materials and interviewing Finnfeeds Finland personnel it became clear that there are limited means of monitoring energy consumption, especially in individual equipment. There are relatively few measurements for flow or temperature which complicated the compiling of initial data for this survey but also makes it more difficult or even impossible for operators to monitor and adjust the process according to energy effectiveness when the process allows it. There is a tradition of thorough and regular monitoring and reporting of total energy consumption, but that is carried out from history data. The most effective ways to affect to energy effectivity in the process is by adjusting it in real time instead of making corrective actions afterwards. Adding measurements especially in suitable locations especially in the unit operations requiring significant amount of heating or cooling would be a relatively easy way to improve both the understanding of where energy is consumed and perhaps even adjust it if the production process allows it.

As the energy consumption is not that known in detail, neither is the real-time quantities of waste heat released from the process. There are measurements and means to monitor some sources of waste heat, but these apply mainly to the already identified largest sources such as cooling water, secondary steam or waste water. Additional measurements for such waste heat sources which have varying natures in terms of temperature and flow are recommended prior to making decisions about heat recovery investments, as these investment proposals in this thesis are based on the initial data that was available. They are simplifications of the process as the calculations are carried out with one exemplary situation instead of carrying out simulations to determine the typical ranges of each proposal.

6 SVENSK SAMMANFATTNING

UNDERSÖKNING AV SPILLVÄRMEKÄLLOR OCH DERAS ANVÄNDNING I PROCESSINDUSTRIN

Varje gång mekaniskt arbete utförs i någon process, är det oundvikligt att den primära energin som matas in i processen inte kan utnyttjas fullt. Spillvärme är värmeenergi som inte utnyttjades i processen, och den kan hittas från många olika källor inom industrin. De flesta litteraturkällor definierar spillvärme som värmeenergin som förloras för miljön, utan att ta ställning till denna energis temperatur eller om den kan användas eller inte. Användningen av denna värmeenergi är dock mycket viktig för att säkerställa en fabriks energieffektiva drift.

Förbättring av energieffektivitet är alltså en viktig del av olika nationella och internationella strategier, till exempel Europeiska Unionens Green Deal och finska statens strategi mot för ett klimatneutralt Finland 2035. Industriell produktion regleras med direktiv och lagar och det är möjligt att få finansiella stöd till investeringar i energieffektivitet. En ökning av den cirkulära ekonomiprincipen är gemensam för alla strategierna och den betyder mera användning av spillvärme istället för att låta den gå till spillo.

Spillvärme kan vara bunden i de olika flöden som lämnar en fabrik och sådana källor kan kategoriseras enligt deras temperatur:

- Temperatur under 50 °C
  - till exempel kylningsvatten, kondensat från mekanisk kylning, avgaslufter
- Temperatur mellan 50 °C och 100 °C
  - till exempel kylningsvatten, avlastningsgaser, kylningsolja från smorda kompressorer
- Temperatur över 100 °C
  - till exempel rökgaser, varma avgaslufter

För att förbättra energieffektivitet i industri, är det viktigt att finna sätt att använda spillvärme antingen internt eller externt. Teoretisk, teknisk och ekonomisk potential måste definieras för varje spillvärmekälla när en lämplig värmesänka finns för användningen. Interna sätt skulle vara primära sätt för användning, och först efter att alla möjliga interna sätt har evaluerats, borde externa sätt studeras. Ett externt sätt att använda spillvärme är säljning av spillvärme till fjärrvärme, vars genomförande behöver samarbete med ett fjärrvärmeföretag.

Systematisk undersökning av spillvärmekällor kan utföras med Pinch-analys, som använder enkla inmatningsdata i analysen. Initialvärden (start- och måltemperatur, specifik värmekapacitet och massflöde) definieras för varje flöde som behöver värme eller kyla, och från dessa initialvärden sammanställs en kurva som representerar alla heta eller alla kalla flöden. De två kurvorna ritas i en graf som uttrycker temperatur och förändringen i entalpi på sådant sätt att de aldrig överlappar och skiljer med åtminstone Pinch-punkten (ΔT_{min}) från varandra. Baserat på analysen kan man bestämma minimala behovet för heta och kalla bruksnyttigheter, jämföra nuvarande situationen i fabriken med minimala behovet, och utforma värmeöverföringen så att Pinch-punkten inte överskrids. Pinch-analysen ger också resultatet för den teoretiska potentialen för värmeöverföring mellan heta och kalla flöden, vilken kan används att förbättra energieffektivitet och intern användning av spillvärme.

Sätt för användning av spillvärme kan kategoriseras i passiva och aktiva sätt baserat om temperaturen hålls på samma nivå eller om den ändras. Interna sätt att använda spillvärme är till exempel användning som sådan, värmeöverföring med värmeväxlare eller värmelagring. Värmeöverföring baserar sig på samma principer som med traditionella källor: enligt termodynamiska lagar kan värmeenergi överföras och överföring sker alltid från högre till lägre temperatur. Värmelagring kan utföras på olika sätt, enklast med lagring av energin i vatten i en behållare av lämplig storlek, men också med användning av värmeenergin för att växla lagringsmaterialens tillstånd. Värmelagring kan också utföras med termokemisk lagring, var energin lagras som resultat av en kemisk reaktion i lagringsmaterial.

Aktiva sätt att använda spillvärme ändrar temperaturen eller växlar värmeenergin till en annan form av energi, till exempel till elektrisk energi. Det mest använda sättet är ökning av temperatur med en värmepump. Det finns många olika tekniker för värmepumpning, antingen med öppen eller stängd cykel, eller antingen mekanisk eller termisk drift, vilket betyder att extern energi i form av elektricitet eller till exempel ånga behövs när värmepumpen används. Användnings-principen för värmepumpar baserar sig på kylmedlets kokpunkts tryckberoende. Alla värmepumpar är planerade individuellt för varje användning. För att stödja investeringsbeslut, kan koefficienter för energieffektivitet (*EER*) och utförandet (*COP*) räknas för att bestämma hur användbar en värmepump är vid en specifik användningsplats.

Organic Rankine Cycle eller ORC är ett sätt att omvandla värmeenergi till elektrisk energi, och det kan använda även 100 °C spillvärme till småskalig elektricitetsproduktion. ORC använder också ett kylmedel som är evaporerad med energin från spillvärme innan det förgasade kylmedlet går till turbinen. Turbinen återhämtar mekanisk energi när den förgasade fluiden expanderas till ett minskat tryck. Den mekaniska energin omvandlas till elektricitet och kondensation av kylmedlet ger ut värmeenergin till en lämplig värmesänka.

Målsättningen med det här diplomarbeten var att göra en undersökning av alla olika spillvärmekällor i processer i Finnfeeds Finland Oy Naantali fabrik, och hitta sätt att utnyttja denna energi både internt och externt. Undersökningen började med flera intervjuer och bekantgörande av materialet som beskriver den nuvarande situationen i fabriken. Massbalanser, produktionsplaner, processdata och mätningar användes för att utgöra energibalanser till de olika avdelningarna i fabriken. Energibalansen används för att utföra Pinch-analys med Aspen HYSYS Energy Analyzer för tre olika produktionsscenarier. Fabrikens nuvarande användning av bruksnyttigheter räknades med värden från rapporter eller mätningar, och jämförelse med Pinch-analysresultat upptäcker att fabriken använder mera heta och kalla bruksnyttigheter än vad är nödvändigt.

Alla fabrikens värmeöverväxlare gicks igenom för att bestämma om de opererar antingen över Pinch-punkten eller under den. I olika scenarier, överskrider följande antal av värmeöverväxlare Pinch-punkten: 16 av 50, 9 av 46 och 9 av 46. I alla scenarierna var området för värmeöverföring mellan heta och kalla flöden ganska smalt, och det finns mycket mera flöden som behöver upphettning än vad det finns flöden som behöver kylning. Det betyder att det finns många olika spillvärmekällor i processen men också att temperaturen av spillvärmen inte nödvändigtvis är tillräckligt hög för användning av spillvärmen som sådan. För mer effektiv användning, måste temperaturen ökas till exempel med en värmepump.

De större källorna av spillvärme var kylningsvatten, avloppsvatten, och olika avgaslufter. Alla källor förenas av att de har en relativt låg temperatur och energiinnehållen varierar beroende av produktion. Mängden av spillvärmeflöden kan variera kraftigt, särskilt i avloppsvatten.

Fem olika investeringsförslag var skisserade för att använda spillvärme. Alla förslag ska minska användning av ånga, eftersom de alla använder spillvärme för att hetta upp råmaterialen. Förslagen innehåller sådana som använder ett processflöde för att hetta upp ett råmaterial, sådana som använder avgasluft att hetta upp inkommande luft, och ett större förslag som använder spillvärme från kylningsvatten och kräver en värmepump för att öka temperaturen. Det här heta flöde används för att hetta upp olika interna flöden såväl som erbjuda värmeenergi som fjärrvärme. Enkla beräkningar av värmeöverföringen och flödesscheman gjordes för varje förslag, såväl som enkla beräkningar av de förändringar som krävs och utrustningen som ska köpas för att implementera varje investeringsförslag. Baserat på de här beräkningarna och potentiella besparingar från ångkostnader, räknades återbetalningstider för att evaluera ekonomisk potential av varje förslag.

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8 APPENDICES

Appendix 1

List of heat exchangers, their heat loads and a comparison with the Pinch points of each Pinch Analysis scenario.

Equipment tag	Over Pinch 56 °C	Over Pinch 30 °C	Over Pinch 45 °C	Q [kW]
HE-1095	No	No	No	1
HE-1001	**Yes**	**Yes**	**Yes**	102
HE-1264	No	No	No	11
HE-1072	No	No	No	11
HE-1296	No	No	No	11
HE-1011	Not in use	No	Not in use	255
HE-1155	Not in use	No	Not in use	62
HE-1269	Not in use	No	Not in use	10
HE-1185	Not in use	No	Not in use	31
HE-1263	**Yes**	**Yes**	**Yes**	131
HE-1081	No	No	No	91
HE-1194	No	No	No	37
HE-1137	Not relevant	Not relevant	Not relevant	197
HE-1201	Not in use	No	Not in use	39
HE-1155	**Yes**	Not in use	No	97
HE-1086	No	No	No	34
HE-1283	Not in use	No	Not in use	64
HE-1195	No	No	No	88
HE-1222	No	No	No	45
HE-1151	No	No	No	109
HE-1132	**Yes**	Not in use	No	scenario 1 102 kW scenario 3 5 kW
HE-1209	**Yes**	**Yes**	**Yes**	253
HE-1210	**Yes**	No	No	117
HE-1211	No	No	No	37
HE-1212	No	No	No	108
HE-1213	No	No	No	33
HE-1034	**Yes**	Not in use	Not in use	138
HE-1263	**Yes**	Not in use	**Yes**	265
HE-1120	**Yes**	Not in use	**Yes**	121
HE-1082	No	Not in use	No	5
HE-1005	No	Not in use	No	71
HE-1288	**Yes**	Not in use	Not in use	138
HE-1108	No	**Yes**	No	492
HE-1226	No	No	No	147
HE-1204	No	No	No	192
HE-1121	No	No	No	48
HE-1163	**Yes**	No	No	780

HE-1014	No	No	No	3
HE-1089	**Yes**	Not in use	No	486
HE-1232	No	**Yes**	No	492
HE-1250	Not in use	Not in use	Not in use	Not relevant
HE-1289	No	No	No	3,902
HE-1237	Not in use	Not in use	Not in use	Not relevant
HE-1062	No	No	No	48
HE-1171	No	No	No	33
HE-1204	No	No	No	214
HE-1092	**Yes**	**Yes**	**Yes**	309
HE-1079	No	No	No	147
HE-1079	No	No	No	286
HE-1032	Not in use	Not in use	Not in use	Not relevant
HE-1122	No	Not in use	Not in use	Not relevant
HE-1167	Not in use	Not in use	Not in use	Not relevant
HE-1170	No	No	No	221
HE-1281	No	No	No	61
HE-1289	No	No	No	51
HE-1183	**Yes**	**Yes**	**Yes**	2,832
HE-1241	No	No	No	3,177
HE-1104	No	No	No	3,177
HE-1016	**Yes**	**Yes**	**Yes**	1,061
HE-1040	**Yes**	**Yes**	**Yes**	1,061
HE-1172	No	No	No	301

Appendix 2

Tables describing the identified sources for waste heat

Table 13 Theoretical potential of identified waste heat sources

	Theoretical potential 24 MW					Notes
Waste heat source	Temperature [°C]	Flow [m^3/h]	Heat load [MW]	State	Theoretically available	
Product 4	68	0.7	0.06	Liquid	Yes	
Product 5	65	0.6	0.05	Liquid	Yes	
Product 6	77	2.3	0.23	Liquid	Yes	
Product 7	85	0.2	0.02	Liquid	Yes	
Cooling water stream 1	40-75	3.4	0.24	Liquid	Yes	Batch process, temp. controlled
Process stream 1	47	6.3	0.36	Heat radiation	No	Batch process
Process stream 2	95	5.9	0.72	Heat radiation	No	Batch process
Process stream 3	92	5.9	0.68	Heat radiation	No	Batch process
Exhaust air from dryer D3, scenario 1	70	7,040	0.27	Gas	Yes	
Process stream 4	100	4.5	0.58	Heat radiation	No	Batch process
Process stream 5	95	4.5	0.55	Heat radiation	No	Batch process
Exhaust air from dryer D1, scenario 2	60	13,000	0.46	Gas	Yes	
Process stream 6	80	0.7	0.07	Heat radiation	No	Batch process
Process stream 7	75	0.7	0.07	Heat radiation	No	Batch process
Process stream 8	80	0.6	0.06	Heat radiation	No	Batch process
Process stream 9	75	0.6	0.06	Heat radiation	No	Batch process
Exhaust air from dryer D3, scenario 3	60	2,800	0.10	Gas	Yes	
Secondary steam 1	60	5.5	1.36	Gas / Liquid	Yes	
Secondary steam 2, scenario 1	50	4.6	2.89	Gas / Liquid	Yes	Batch process
Secondary steam 3, scenario 1	70	5.5	3.46	Gas / Liquid	Yes	Batch process
Secondary steam 4	70	3.4	2.14	Gas / Liquid	Yes	Batch process
Secondary steam 2, scenario 3	50	2.3	1.45	Gas / Liquid	Yes	Batch process
Secondary steam 3, scenario 3	50	2.2	1.38	Gas / Liquid	Yes	Batch process
Condensate expansion steam	83	1,1	0,7	Gas / Liquid	Yes	Estimate
Odour gases	25	16,632	0.73	Gas	Yes	Low temperature
Odour gas purifying water	30	38	1.79	Liquid	Yes	Temperature controlled
Cooling water (sea water)	10...20	350	6.28	Liquid	Yes	Low temperature
Waste water	15...25	3...20	0.35	Liquid	Yes	Low temperature

Table 14 Technical potential of identified waste heat sources

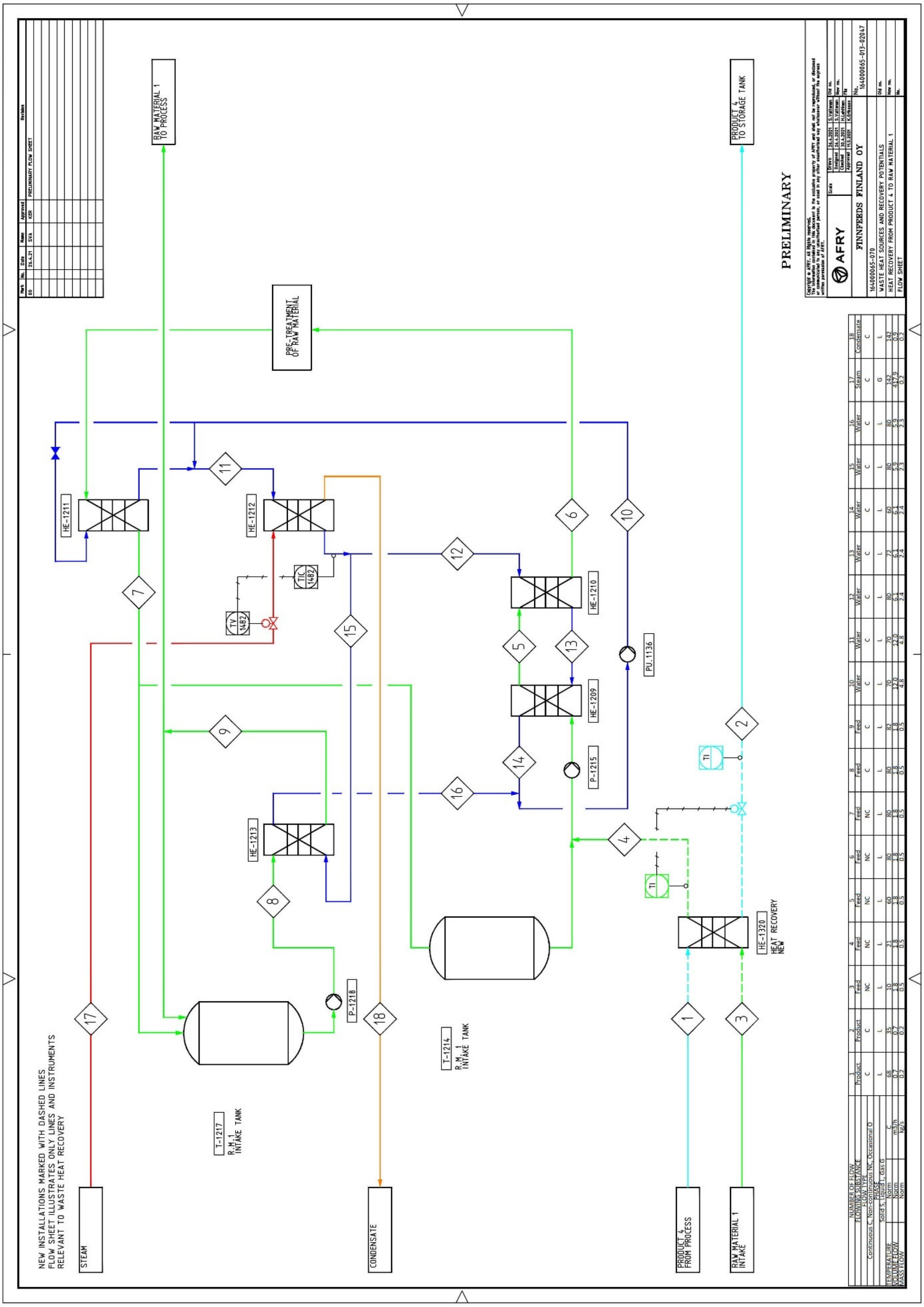
		Technical potential 23.7 MW
Waste heat source	Technically available	Justification
Product 4	Yes	Liquid flow with recoverable heat energy
Product 5	No	Precipitates when cooled down
Product 6	No	Intermediary product, requires further heating in process
Product 7	No	Must be kept in a specific temperature
Cooling water stream 1	Yes	Liquid flow with recoverable heat energy
Process stream 1	No	Temperature cannot decrease too much or too fast
Process stream 2	No	Temperature cannot decrease too much or too fast
Process stream 3	No	Temperature cannot decrease too much or too fast
Exhaust air from dryer D3, scenario 1	Yes	Gas flow with recoverable heat energy
Process stream 4	No	Temperature cannot decrease too much or too fast
Process stream 5	No	Temperature cannot decrease too much or too fast
Exhaust air from dryer D1, scenario 2	Yes	Gas flow with recoverable heat energy
Process stream 6	No	Temperature cannot decrease too much or too fast
Process stream 7	No	Temperature cannot decrease too much or too fast
Process stream 8	No	Temperature cannot decrease too much or too fast
Process stream 9	No	Temperature cannot decrease too much or too fast
Exhaust air from dryer D3, scenario 3	Yes	Gas flow with recoverable heat energy
Secondary steam 1	Yes	A significant amount of energy in state change
Secondary steam 2, scenario 1	Yes	A significant amount of energy in state change
Secondary steam 3, scenario 1	Yes	A significant amount of energy in state change
Secondary steam 4	Yes	A significant amount of energy in state change
Secondary steam 2, scenario 3	Yes	A significant amount of energy in state change
Secondary steam 3, scenario 3	Yes	A significant amount of energy in state change
Condensate expansion steam	Yes	A significant amount of energy in state change
Odour gases	Yes	Gas flow with recoverable heat energy
Odour gas purifying water	Yes	Liquid flow with recoverable heat energy
Cooling water (sea water)	Yes	Liquid flow with recoverable heat energy
Waste water	Yes	Liquid flow with recoverable heat energy

Table 15 Economic potential of identified waste heat sources

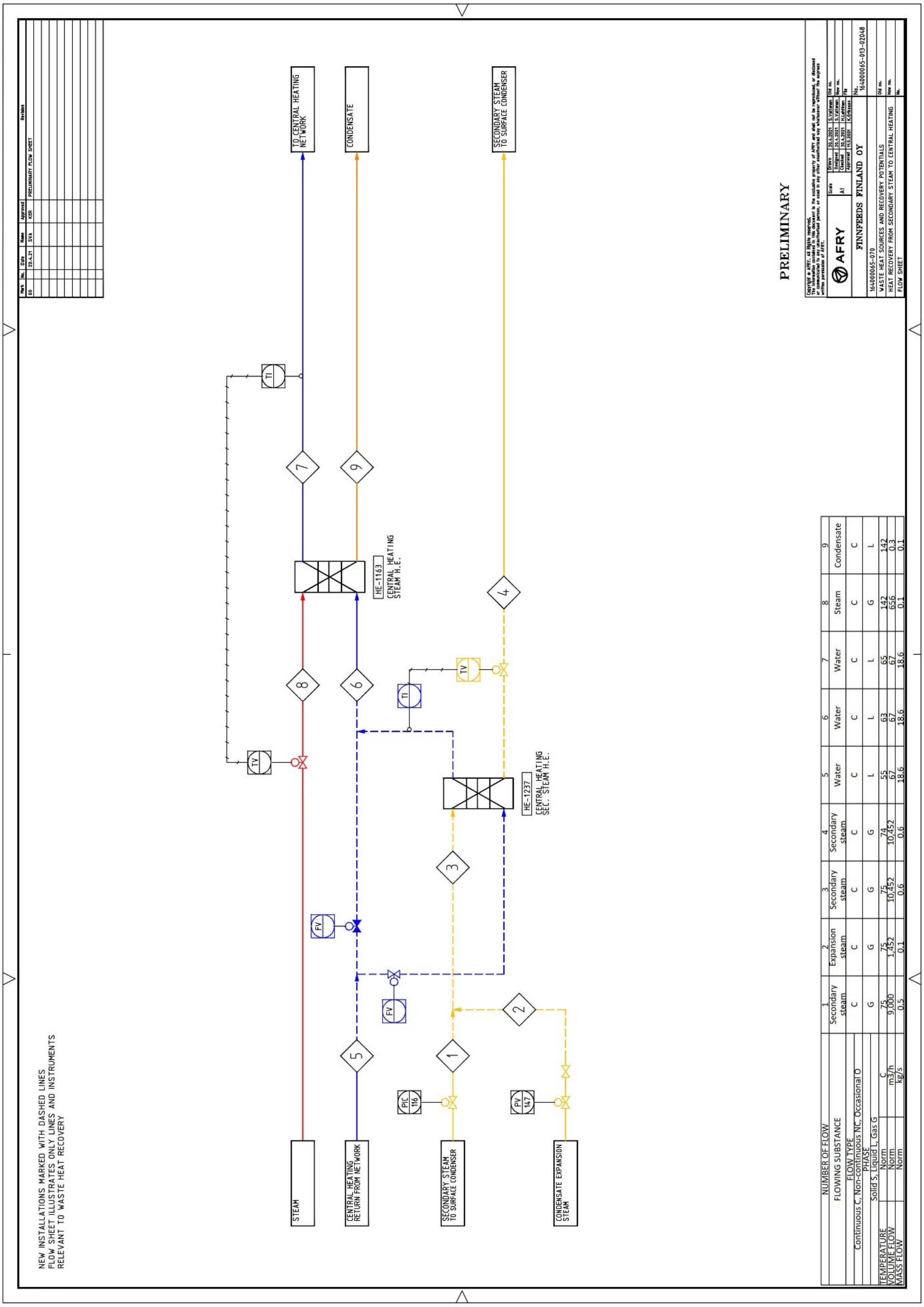
Waste heat source	Economically available	Economic potential 22.6 MW Justification
Product 4	Yes	When paired up with a suitable heat sink
Product 5	No	Not technically available
Product 6	No	Not technically available
Product 7	No	Not technically available
Cooling water stream 1	Yes	When paired up with a suitable heat sink
Process stream 1	No	Not technically available
Process stream 2	No	Not technically available
Process stream 3	No	Not technically available
Exhaust air from dryer D3, scenario 1	Yes	Heat recovery to incoming air possible
Process stream 4	No	Not technically available
Process stream 5	No	Not technically available
Exhaust air from dryer D1, scenario 2	Yes	Heat recovery to incoming air possible
Process stream 6	No	Not technically available
Process stream 7	No	Not technically available
Process stream 8	No	Not technically available
Process stream 9	No	Not technically available
Exhaust air from dryer D3, scenario 3	Yes	Heat recovery to incoming air possible
Secondary steam 1	Yes	Heat recovery through cooling water
Secondary steam 2, scenario 1	Yes	Heat recovery through cooling water
Secondary steam 3, scenario 1	Yes	Heat recovery through cooling water
Secondary steam 4	Yes	Heat recovery through cooling water
Secondary steam 2, scenario 3	Yes	Heat recovery through cooling water
Secondary steam 3, scenario 3	Yes	Heat recovery through cooling water
Condensate expansion steam	Yes	When paired up with a suitable heat sink
Odour gases	No	Requires a heat pump
Odour gas purifying water	Yes	Heat recovery through cooling water
Cooling water (sea water)	Yes	Requires a heat pump
Waste water	No	Irregular flow and alternating temperature

Appendix 3

Flow sheets for investment proposals

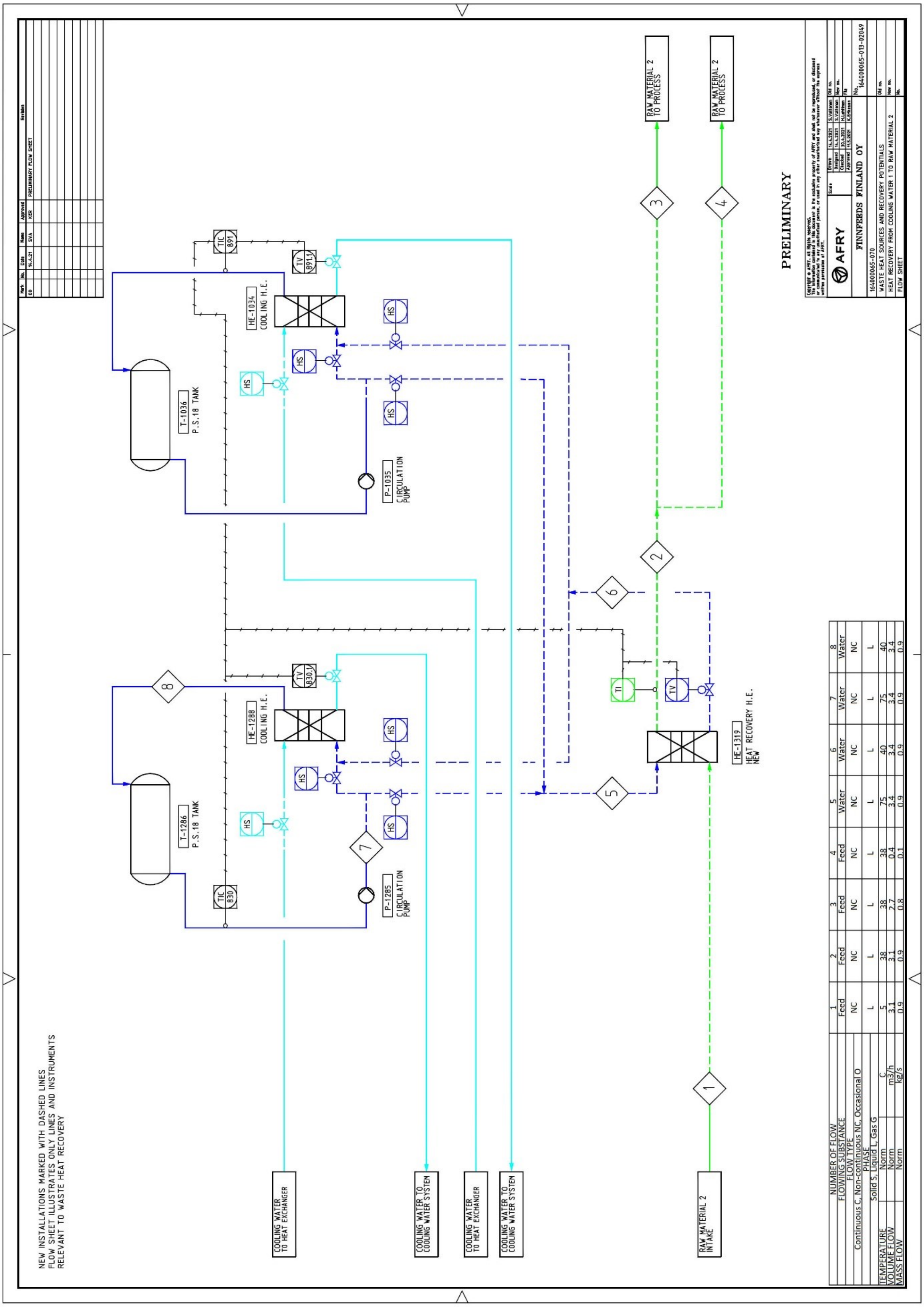


Appendix 3



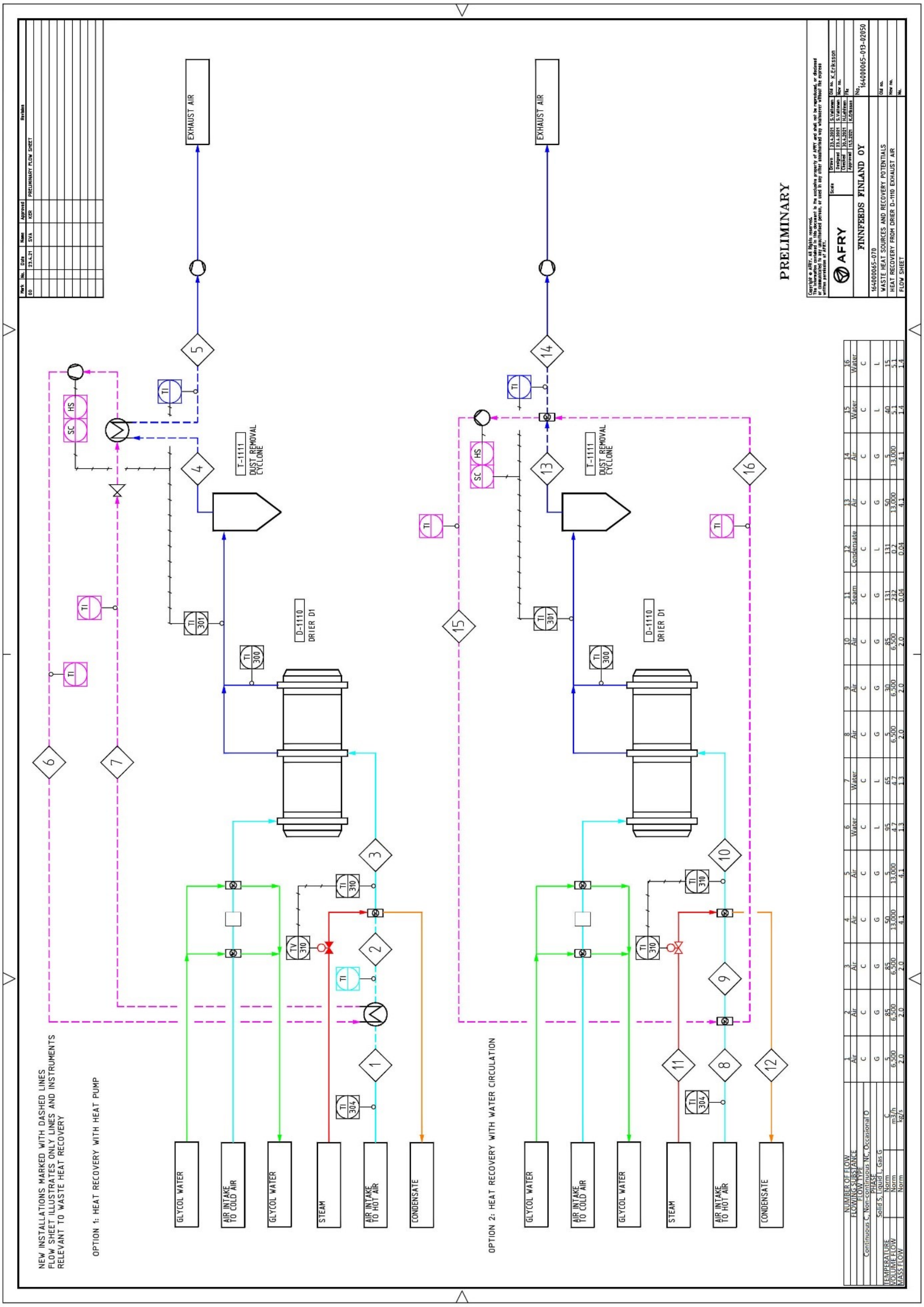
NUMBER OF FLOW		1	2	3	4	5	6	7	8	9
FLOWING SUBSTANCE		Secondary steam	Expansion steam	Secondary steam	Secondary steam	Water	Water	Water	Steam	Condensate
FLOW TYPE Continuous C, Non-continuous NC, Occasional O		C	C	C	C	C	C	C	C	C
PHASE Solid S, Liquid L, Gas G		G	G	G	G	L	L	L	G	L
TEMPERATURE Norm	C	75	75	75	74	55	63	65	142	142
VOLUME FLOW Norm	m3/h	9,000	1,452	10,452	10,452	67	67	67	656	0.3
MASS FLOW Norm	kg/s	0.5	0.1	0.6	0.6	18.6	18.6	18.6	0.1	0.1

Appendix 3

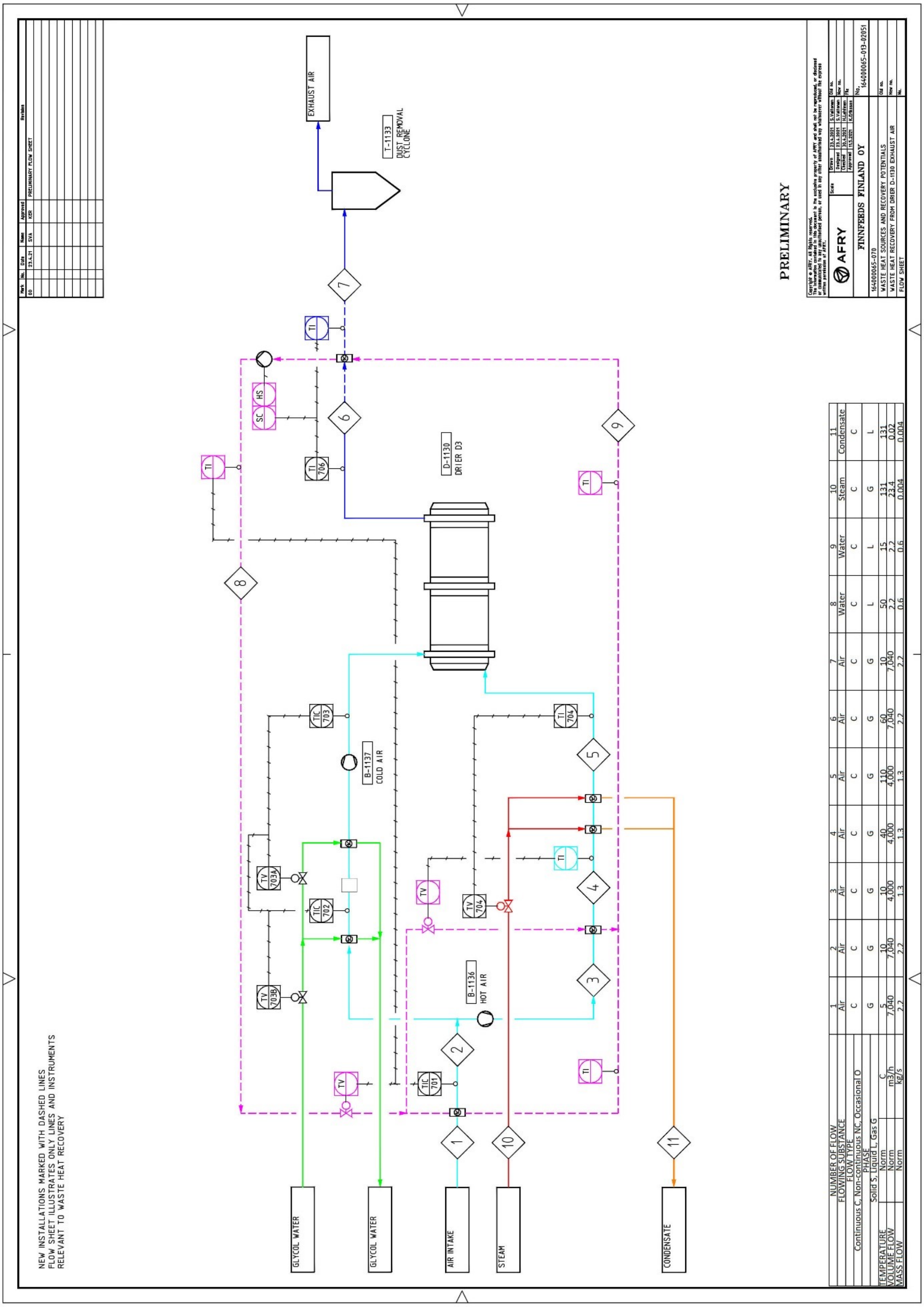


NUMBER OF FLOW			1	2	3	4	5	6	7	8
FLOWING SUBSTANCE			Feed	Feed	Feed	Feed	Water	Water	Water	Water
FLOW TYPE Continuous C, Non-continuous NC, Occasional O			NC	NC	NC	NC	NC	NC	NC	NC
PHASE Solid S, Liquid L, Gas G			L	L	L	L	L	L	L	L
TEMPERATURE	Norm	C	5	38	38	38	75	40	75	40
VOLUME FLOW	Norm	m3/h	3.1	3.1	2.7	0.4	3.4	3.4	3.4	3.4
MASS FLOW	Norm	kg/s	0.9	0.9	0.8	0.1	0.9	0.9	0.9	0.9

Appendix 3


NEW INSTALLATIONS MARKED WITH DASHED LINES
FLOW SHEET ILLUSTRATES ONLY LINES AND INSTRUMENTS
RELEVANT TO WASTE HEAT RECOVERY
OPTION 1: HEAT RECOVERY WITH HEAT PUMP
OPTION 2: HEAT RECOVERY WITH WATER CIRCULATION
GLYCOL WATER
AIR INTAKE TO COLD AIR
STEAM
AIR INTAKE TO HOT AIR
CONDENSATE
EXHAUST AIR
D-1110 DRIER D1
T-1111 DUST REMOVAL CYCLONE
TI 304
TV 310
TI 310
TI 300
TI 301
TI
SC
HS
PRELIMINARY
NUMBER OF FLOW: 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16
FLOWING SUBSTANCE: Air Air Air Air Air Water Water Air Air Air Steam Condensate Air Air Water Water
FLOW TYPE (Continuous C, Non-continuous NC, Occasional O): C C C C C C C C C C C C C C C C
PHASE (Solid S, Liquid L, Gas G): G G G G G L L G G G G L G G L L
TEMPERATURE Norm C: 5 85 85 50 5 95 65 5 30 85 131 131 50 5 40 15
VOLUME FLOW Norm m3/h: 6,500 6,500 6,500 13,000 13,000 4.7 4.7 6,500 6,500 6,500 232 0.2 13,000 13,000 5.1 5.1
MASS FLOW Norm kg/s: 2.0 2.0 2.0 4.1 4.1 1.3 1.3 2.0 2.0 2.0 0.04 0.04 4.1 4.1 1.4 1.4
AFRY
FINNFEEDS FINLAND OY
164000065-070
WASTE HEAT SOURCES AND RECOVERY POTENTIALS
HEAT RECOVERY FROM DRIER D-1110 EXHAUST AIR
FLOW SHEET
164000065-013-02050
00 23.4.21 SVA KER PRELIMINARY FLOW SHEET

Appendix 3



NUMBER OF FLOW			1	2	3	4	5	6	7	8	9	10	11
FLOWING SUBSTANCE			Air	Air	Air	Air	Air	Air	Air	Water	Water	Steam	Condensate
FLOW TYPE Continuous C, Non-continuous NC, Occasional O			C	C	C	C	C	C	C	C	C	C	C
PHASE Solid S, Liquid L, Gas G			G	G	G	G	G	G	G	L	L	G	L
TEMPERATURE	Norm	C	5	10	10	40	110	60	10	50	15	131	131
VOLUME FLOW	Norm	m3/h	7,040	7,040	4,000	4,000	4,000	7,040	7,040	2.2	2.2	23.4	0.02
MASS FLOW	Norm	kg/s	2.2	2.2	1.3	1.3	1.3	2.2	2.2	0.6	0.6	0.004	0.004

Appendix 3