Utilizaion of low temperature waste heat at Eramet Norway Kvinesdal

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Abstract

With the increasing energy demand worldwide, it is essential to utilize the energy available to the fullest, not unnecessarily letting energy be wasted. Eramet Norway Kvinesdal is a manganese alloy producer, and they require a lot of energy for their processes. Several production steps lead to waste heat, and investigation regarding their suitability for utilization was made. Filtered sludge water from the cleaning of oven gas was considered to be a viable waste heat source. With its low temperature, two utilization purposes seemed applicable, floor heating and snow/ice melting. Four scenarios were investigated as to how much heat energy that could be utilized. Fluid analysis was based on element concentration in the water, and approximate calculations were done with the little information available. Although the presence of metals was more significant than that of the nearby river water, the calculations made showed that due to the filtration steps, the water characteristics are quite similar to that of normal water. Furthermore, the output potential for the fluid stream was calculated. A plate heat exchanger was looked into as a possible installment to extract the waste heat and calculations regarding flow characteristics, plate gaps, and pressure drop were completed for four different size heat exchangers. The results showed that scenario 1, where the waste heat is used for floor heating, is the most applicable solution, especially in regard to frequent use and energy saving. Size 2 and size 3 heat exchangers with plate gaps of 2-2,5 *mm* were recommended to maintain sufficient velocities and heat transfer while avoiding rapid fouling and clogging.

Preface

This is the master thesis at the end of my studies in the master program Renewable Energy at University of Agder. As there is a growing demand on power in the world, more focus is being placed at utilizing the energy already consumed fully. This project therefore caught my interest, as a possibility to find and make use of energy that otherwise would have gone to waste. My bachelor degree is in Heating, Ventilation and Air Conditioning, and such it was for me interesting to see if waste heat can be used for any of those applications. Due to the covid-19 world wide pandemic, visiting and closely working together with the company were sadly not possible. Measurements and analyses that should have been completed, were thus not done, and assumptions and simplification had to be made instead. This have lowered the depth and quality of the thesis. Therefore some results may turn out to be unrealistic for the application at site due to these uncertainties. Nevertheless i would like to thank my supervisor professor Souman Rudra for guiding me throughout the thesis, and also Stine Skagestad, technical manager at Eramet Norway Kvinesdal, for providing me with the information needed and being encouraging for the project.

Fredrik Haukeland

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Notation

β	Chevron plate angle, [°]
ΔP_c	Channel pressure drop, $[Pa]$
ΔP_p	Port pressure drop, $[Pa]$
ΔP_t	Total pressure drop, $[Pa]$
ΔT	Temperature difference, $[K]$
$\dot{m}$	mass flow, $[kg/s]$
$\dot{V}$	Volume flow, $[\dot{V}]$
μ	Viscosity, $[kg/(m \cdot s)]$
μ_f	Viscosity of fluid containing elements, $[kg/(m \cdot s)]$
μ_w	Viscosity of fluid at wall temperature, $[kg/(m \cdot s)]$
μ_{mix}	Viscosity of fluid mixture, $[kg/(m \cdot s)]$
Φ	Volume fraction
ρ	Density, $[kg/m^3]$
A	Heat transfer surface area, $[m^2]$
A_p	Area of a plate, $[m^2]$
C_h	Constant to calculate Nusselt number
C_p	Specific heat capacity, $[kJ/(kg \cdot K)]$
d_c	Cross section of channel, $[m^2$
d_h	Hydraulic diameter, $[m]$
d_m	Distance temperature travels in a material, $[m]$
D_p	Port diameter, $[m]$
F	Correction factor LMTD

f Friction factor for pressure calculations

G_c Fluid mass velocity in channels, $[kg/(m^2 \cdot s)]$

G_p Fluid mass velocity through ports, $[kg/(m^2 \cdot s)]$

h Convective heat transfer coefficient, $[W/(m^2 \cdot K)]$

k Thermal conductivity, $[W/(m \cdot K)]$

k_f Thermal conductivity of the fluid, $[W/(m \cdot K)]$

k_p Thermal conductivity of the particles, $[W/(m \cdot K)]$

k_{mix} Thermal conductivity of the mixture, $[W/(m \cdot K)]$

L_h Horizontal port distance, $[m]$

L_p Plate length, $[m]$

L_v Vertical port distance, $[m]$

m Mass of a material, element or fluid, $[kg]$

n Constant to calculate Nusselt number

N_c Number of channels

N_cc Number of channels cold fluid

N_ch Number of channels hot fluid

N_p Number of plates

N_pa Number of passes

Nu Nusselt Number

Pr Prandtl number

Q Heat transfer rate, $[kW]$

R_f Fouling resistance of the fluid, $[m^2 \cdot K/W]$

Re Reynolds number

s Plate spacing, $[m]$

t_i Inlet temperature, $[°C]$

t_o Outlet temperature, $[°C]$

t_p Plate thickness, $[m]$

$T_{c,i}$ Inlet temperature cold fluid, $[°C]$

$T_{c,o}$ Outlet temperature cold fluid, $[^\circ C]$

$T_{h,i}$ Inlet temperature hot fluid, $[^\circ C]$

$T_{h,o}$ Outlet temperature hot fluid, $[^\circ C]$

U Overall heat transfer coefficient, $[kW/(m^2 \cdot K)]$

u Velocity, m/s

U_0 Calculated overall heat transfer coefficient, $[kW/(m^2 \cdot K)]$

u_c Velocity in channels, m/s

U_{est} Estimated overall heat transfer coefficient, $[kW/(m^2 \cdot K)]$

W_p Plate width, $[m]$

Abbreviation

CFD Computational Fluid Dynamics.

CHTC Convective Heat Transfer Coefficient.

ENK Eramet Norway Kvinesdal.

GAC Granular Activated Carbon.

HTA Heat Transfer Area.

LMTD Logarithmic Mean Temperature Difference.

NTU Number of Transferred Units.

OHTC Overall Heat Transfer Coefficient.

PAH Polycyclic Aromatic Hydrocarbon.

PHE Plate Heat Exchanger.

SHC Specific Heat Capacity.

1 Introduction

1.1 Background and motivation

There is a growing focus on energy efficiency due to the increase of climate gases in the atmosphere. The industry, in particular, has a large consumption of energy for its processes, one-third of the world's energy consumption, as reported by the International Energy Agency [1]. On top of that, they report that the industry is the source of 36% of the world's CO_2 emissions. However, all the energy is not fully used in the processes, much of it is released into the environment. [2] estimated that 72% of energy in most common sectors is lost.

While trying to achieve a reduction in energy consumption, the demand is ever-growing. Since 1971 the energy consumption from the industry has increased by 61% in 2004 [1]. This is greatly due to the industrialization of underdeveloped countries. Therefore it is all the more reason to achieve energy-efficient processes. If the Paris Agreement's goal is to be kept, keeping the global mean temperature change below 2 degrees above the pre-industrial temperature, utilization of waste heat is a crucial step to take [2].

Utilization of waste heat energy can mainly be used by either changing into other forms of energy, mechanical and electrical, or transfer the heat to another medium. Power cycles such as the organic Rankine cycle and turbines are typical examples of converting the energy, while heat exchangers are a classic example of keeping the energy as heat energy. Therefore [3] categorized it as indirect and direct recovery methods.

In the metal industry, it is often desirable to melt the metal into liquid form, which usually requires high temperatures. However, like most processes, a complete transfer of all energy is almost impossible to achieve, and in such instances, the amount of energy going to waste can be incredibly high. In each process step, energy may be lost along the way, most of this in the form of thermal energy. The utilization of waste heat energy would reduce overall energy consumption.

Eramet Norway Kvinesdal, ENK, is a world leading producer of manganese alloys. The factory is located in the south of Norway, in a low populated area called Kvinesdal. They require a considerable amount of energy for their processes, and waste heat sources occur in several areas. Utilization of these may reduce the overall energy consumption at the factory. Some applications for the waste heat have already been applied, including a steam turbine, hot water for aquaculture, and a low-temperature district heating for nearby buildings.

1.2 Objective and research questions

The main objective of this study is formulated as follows:

- Utilize a waste heat source for an application at Eramet Norway Kvinesdal and consider its feasibility.

Eramet Norway desires to reduce the amount of energy that is going to waste at the factory in Kvinesdal. Utilizing waste heat can supply other applications and their demands, reducing the need to extract energy for isolated purposes. Overall energy consumption may then be reduced. District heating based on some of the waste heat sources has already been installed. For the time being, it does not seem feasible to expand this system due to the low population and infrastructure in the area. Thus other solutions must be considered. To achieve the objective, some research questions have been made. By answering these research questions, a conclusion may be given regarding the viability of the waste heat source, and if it is worth pursuing. The research questions are as follows:

- Which waste heat sources at Eramet Norway Kvinesdal can be utilized?
- For which purposes can the recovered waste heat be utilized?
- How can the waste heat be extracted?
- Is it feasible to implement?

1.3 Limitations and assumptions

In order to properly fulfill the study's objectives, visit and a prolonged working stay at ENK was planned. Solutions could then be based on some of their experience and attempts at waste heat utilization. It would ensure the solutions to be even more case specific for their use. However, due to the covid-19 worldwide pandemic, this was not possible, and close cooperation became challenging. Therefore, simplifications and assumptions had to be made were measurements and inspection were no longer possible.

The determination of fluid flow characteristics and the heat exchanger operation and performance were done by calculations. An in depth analysis of computer programs is required to give more realistic results of how the system operates. The calculations made will, however, give an idea of the operation and performance of the heat exchanger and the potential of the waste heat source.

1.4 Structure of the report

The structure of this report is organized accordingly. First, there is a literature review investigating what other studies have done that could be useful for this study. The review includes waste heat categorization, heat exchanger consideration, water filtration, and fluid analysis.

Secondly, the theory necessary to fulfill the task is presented. The theory includes an introduction to standard heat exchangers, methods of heat transfer, fouling, pressure, and filters. Afterward, the method is presented considering waste heat sources, categorizing utilization scenarios, and calculating fluid characteristics and plate heat exchanger, PHE.

Then results and discussion will be presented together. Here fluid properties are determined, along with the potential heat transfer. Results for each scenario are then presented and discussed before they are compared and considered. Lastly, a conclusion to sum up the findings of the study, followed by further research.

2 | Review of Literature

2.1 Overview of industrial waste heat

Waste energy can be divided into categories: combustible(fuel), thermal(heat), and excess pressure. A combination of these is also possible [4]. Thermal energy, referred to as waste heat, can be categorized further into different temperature ranges. These temperature ranges are often divided into high, medium, and low, as categorized by [5]. For this paper, two further categories will be used, ultra-high and ultra-low, similar to [6]. Categories will be as follows:

- **Ultra-high:** 870°C and above
- **High:** Between 650°C and 870°C
- **Medium:** Between 230°C and 650°C
- **Low:** Between 120°C and 230°C
- **Ultra-low;** 120°C and below

The potential of utilizing waste heat is hugely dependant on the temperature difference, the more substantial difference, the better. A large part of the waste heat from industry, somewhere between 60% and 66%, is what can be categorized as low and ultra-low temperature waste heat [5, 7, 8]. A considerable amount of this is in the ultra-low temperature range. On top of that, another large portion of this is between 30 and 40 °C [9]. Due to the low temperature, the application is limited, especially during the summer. Heat pumps may be an alternative to boost the quality of heat energy extracted.

Characteristics of waste heat can vary greatly from process to process, where some are easier to utilize than others. Some processes may have its waste heat source as clean water, where others are dirty and contaminated. [6] have made a classification of waste heat categories, differentiates waste heat sources, and their level of contamination referred to as waste heat in harsh environments. Among these were flue gases containing combustible, corrosive, and greenhouse gases. Waste heat sources containing byproducts such as solids and sludge were also considered among the harsh environments.

2.1.1 Fluid characteristics

Waste heat sources often contain a lot of impurities, and measures into making them more applicable may be needed. [10] studied the effectiveness of a combination of filters at a landfill leachates. They concluded that a combination of filters was necessary to remove a wide variety of particles. Sand filters removed larger particles, making it a recommended first filter option, increasing the effectiveness of other filters. Furthermore, peat could remove more of PHC's, DEHP, and oxy-PAH. Granular Activated Carbon, GAC, was the most effective filter to remove most particles, in particular PAH, and was therefore recommended to have last among the combination of filters.

[11] compared different models for determining the viscosity of nanofluids. They concluded that Peng-Robinson and Esmaeilzadeh-Roshanfekr equations gave all over more accuracy of viscosity estimations than older models such as Einstein and Brinkman. However, the difference was smaller at lower concentrations and particle sizes. [12] investigated the viscosity of sludge water on a membrane bioreactor. The concentration of suspended solids in their water was between 5,23 g/L and 14,10 g/L. Viscosity varied between 0,0024 $kg/m \cdot s$ and 0,03 $kg/m \cdot s$, dependent on the temperature and concentration of solids. It was observed that when the viscosity was above 0,02 kg/ms, the membrane bioreactor started to clog.

Thermal conductivity and viscosity were studied for 1-4% of bentonite in nanofluids by [13]. They observed that the thermal conductivity increased as the particle concentration of bentonite was increased, and as expected, so did the viscosity. Others have studied nanofluids, and their increased thermal conductivity, among those is [14], who reports a 60% increase in thermal conductivity of water containing 5% copper oxide particles. [15] demonstrated that for nanofluids, theoretical models gave lower values for thermal conductivity than the measured data. They also experienced that a decrease in particle size gave an increase in thermal conductivity.

2.2 Heat exchanger consideration

[16] compared a shell and tube heat exchanger with a plate fin heat exchanger. The paper was about fuel savings on a coffee roasting plant. A numerical analysis of the efficiency was used, with computational fluid dynamics, CFD, as a tool. Analysis with different designs was done for size, baffles, and length of tubes. In their study, the performance of the shell and tube heat exchanger gave slightly better results. However, the size of the plate fin heat exchanger was approximately ten times smaller. The pressure loss in the heat exchanges was almost similar. Therefore, by increasing the plate fin heat exchanger size slightly, that option would be the most beneficial regarding available space at the coffee plant.

[17] compared different types of heat exchangers and to which extent they were affected by fouling. A table is presented, where the PHE performs quite well compared to the other types of heat exchangers, especially the shell and tube. Although different techniques for mitigation and removal of fouling have been investigated over the years, fouling remains among the major unresolved problems in thermal science to this day.

2.2.1 Plate heat exchanger

[18] studied the PHE, its construction, and its operation. Comparison with the shell and tube heat exchanger was made, concluding that the PHE performs excellently within its capability range, particularly for liquid/liquid duties.

[19] investigated the effect of plate spacing in a PHE. They concluded that when increasing the plate spacing, the necessary heat transfer area, HTA, increased as well, while the pressure drop was decreased. [20] also investigated the optimum plate spacing and compared it for counter flow and parallel flow. The results showed that the optimal plate spacing for a counter flow heat exchanger was lower of the two.

[21] compared different pressure drop calculations, and concluded that Mulley and Buonopane & Troupe correlations for friction factor to be the most realistic ones.

3 | Existing waste heat sources at Eramet Norway Kvinesdal

There are several waste heat sources at ENK, while some of them are already wholly or partly utilized, others are still untouched. Some of the promising waste heat sources that can yet be used are presented in this chapter.

3.1 Heat from Slag in piles

Slag is a byproduct of metal production, and are regularly removed from the ovens at ENK. Straight out of the ovens, the slag has a temperature of up to 1500 °C. Extraction of slag from the ovens is not a continuous process, but rather in batches. Each oven has its bin where slag is being transported. There is already a solution in place to make use of some of the heat. Water is flushed over the slag piles to recover some of the heat. Some water evaporates, while a part of it is gathered in a storage tank, where it is used for district heating. However, this is only done for one of the three ovens.

3.2 Cooling of metal in molds

When the metal is in liquid form, it is poured into flat molds where it is cooled down until they solidify. Straight after the metal is cast, it has a temperature of approximately 1300 °C. The metal is laying in the molds for about 30 minutes, until they reach a temperature of 890 °C. Each year it is produced on average 180.000 ton metal.

3.3 Cooling of metal on the floor in the oven hall

After the metal has been cooled down to 890 °C, it is transported to the oven hall for further cooling and storage. The metal is added to the hall in different piles, as the metal is produced. There is still a lot of heat energy emitted from the metal as it cools down to room temperature. Some of the heat is going to the floor, while most of it dissipates upwards as radiation and convection heat.

3.4 Dust filtered exhaust-air from smoke cleaning facility

Whenever metal is extracted from the ovens, a lot of dust particles occur in the air. The air is cleaned in a smoke cleaning facility before it exits through a large chimney. The air in the chimney has a temperature of 44 °C. Airflow is unknown and alternating, but 150.000 m^3/h can be assumed. Ventilation above the molds is a part of this system.

3.5 Oven gas before scrubbers in cleaning facility

From the chemical reactions in the ovens, Carbon monoxide(CO), Methane gas(CH_4), Hydrogen gas(H_2) as well as some other gases are produced. At the top of the ovens, there is a pipe where the gases exit through. The gas has very varying temperatures, between 200 °C and 800 °C, before it is cleaned in scrubbers. The temperature of the ovens differs. Oven 1 has an average temperature of 591 °C, oven 2 466 °C, and oven 3 504 °C. When all ovens are in full operation, a total of 19.000 Nm^3/h air is going through. Depending on the ovens, the heat effect on this gas is varying a little bit.

3.6 Sludge water from cleaning of oven gas

As the oven gas is full of various particles, the gas is cleaned in scrubbers. Cleaning happens in three steps. The first two steps are water from a sludge tank, while the third step is water from the PAH sand filters, which is cleaner. In the process, the sludge water absorbs some of the heat energy of the oven gas and gets a temperature of 73 °C. After leaving behind some of the impurities in a sludge tank, most of the water is going directly back to step one and two in the scrubbers. Excess water goes through additional cleaning in the PAH facility, where it is sent through sand filters. A portion of this is further cleaned in carbon filters before it is released to the sea. The water that goes through just the sand filters is at 56 °C with a mass flow of 163 m^3/h. The portion of water going through all the cleaning steps has a temperature of 53 °C with a volume flow of 23 m^3/h. These are logged yearly average values of 2014.

3.7 Smoke gas from combustion in gas boiler

In the gas boiler, the oven gas is combusted, and in doing so, converting the gases through chemical reactions. The gas initially contains gases like Carbon monoxide (CO), Methane gas (CH_4), and Hydrogen gas (H_2), among other gases. The resulting smoke gas is mostly carbon dioxide(CO_2), nitrogen gas(N_2), and water steam(H_2O), although other gases are also present. Temperature is quite high, and as a result, it is already being used for some applications. Firstly it heats steam that is going to the turbines. Afterward, it heats water to the gas boiler feeding tank. When these processes have been completed, the smoke is at 110 °C, with an approximate volume flow of 46.000 m^3/h. It is then sent to a mercury-cleaning facility, for later to be released into the environment.

3.8 Flush water tapped from gas boiler

Flush water is tapped from a tank above the gas boiler, at around 100 °C. It is used to heat the inlet air to the gas boiler, and have a temperature of 70 °C afterward. During summer, it is brought out to the sea. Although the temperature is higher than some of the other waste heat sources, the flow is rather low, merely 0,83 m^3/h.

3.9 Waste heat comparison

Figure 3.1 shows the temperatures of each waste heat source presented. Four of the sources much larger temperatures than the four that are below 100 °C. Figure 3.1 shows the heat energy in each waste heat source in MW. Flush water tapped from the gas boiler is barely showing in the figure because the energy level is only 0,063MW. While some sources have much higher energy than others, it may be incredibly hard to extract. Take, for example, the heat from slag in piles. One third of this is tried utilized for district heating. With the solution at hand, only a mere 300 kW of the 3,6 MW is delivered. The temperature provided in the district heating is also much lower, somewhere in between 45-50 °C. The graphs presented are therefore not a good representation for which ones are the best, but rather a presentation of the energy that is present in some waste heat sources.

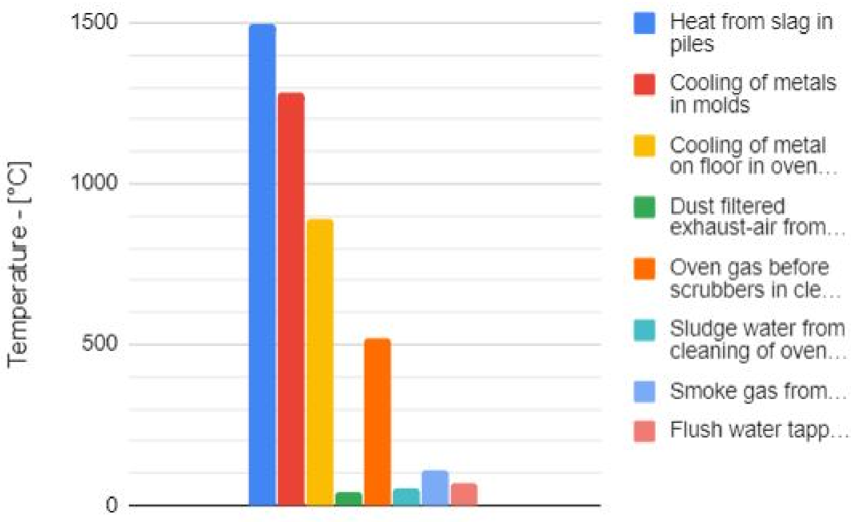


Figure 3.1: Temperatures of the waste heat sources

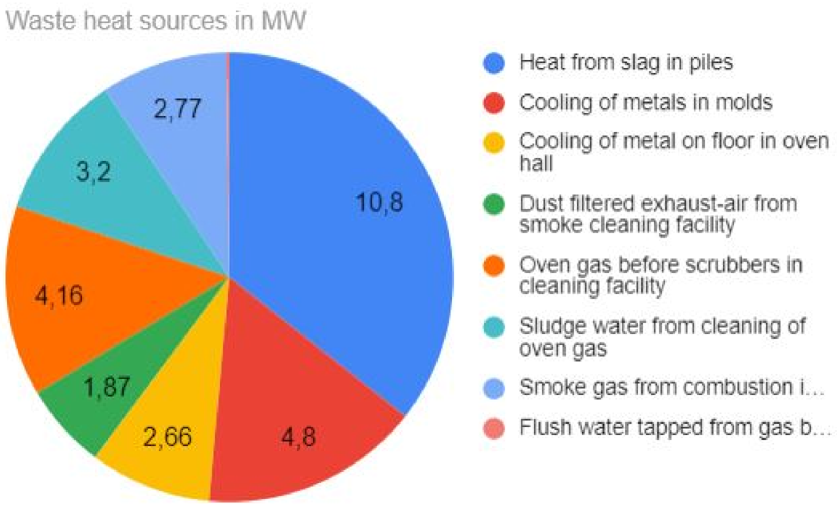


Figure 3.2: Energy in the waste heat sources

4 | Theory

In this chapter theory needed to perform the necessary calculations and considerations is presented. This includes basics about heat exchangers, heat transfer, fouling, flow characteristics, pressure, and water filtration.

4.1 Heat exchangers

A heat exchanger transfers thermal energy from one fluid to a preferable fluid so it can more easily be utilized for heating/cooling purposes. There are many types of heat exchangers, where performance varies depending on the conditions. Primary criteria for selection of a heat exchangers are the type of fluids, temperature, pressure, mass flow, area footprint, and cost. [22] presents a table for the selection of heat exchanger in typical conditions. While the table gives a general idea of what is considered a suitable fit, it may not always be the case. In the table PHE is mentioned as a good fit for low viscosity fluids that are operating on low temperature and pressure. [23, 22, 17] all suggest that PHEs have a lower fouling rate than the commonly used shell and tube heat exchanger. At the same time, a PHE is simple to perform maintenance on, has a considerably lower footprint area, and is considered a cheaper investment cost than the shell and tube heat exchanger [24]. PHE works well with small temperature differences between the hot and cold fluid. ENK had previously installed a heat exchanger that turned out to require frequent maintenance, thus fouling is a central aspect to consider in the selection. A PHE may, in this case, be an adequate choice as a heat exchanger for the filtered sludge water at ENK. There are several types of heat exchangers, standard options are the shell and tube, which is the most common heat exchanger, PHE, and spiral heat exchanger.

4.1.1 Shell and tube heat exchanger

The shell and tube heat exchanger consists of many small tubes, where one fluid passes through, encapsulated by a shell, where the other fluid flows, Figure 4.1. These types of heat exchangers can be configured in different ways. One way has the tubes in a U formation, a U-tube heat exchanger. Here the tubes have a bend inside the heat exchanger, so the fluid is entering and leaving the heat exchanger on the same side. Another type is straight tube heat exchanger, where the tubes are straight all the way through. Here the fluid is entering on one side and leaving on the other side of the heat exchanger. Such a heat exchanger is usually easier to clean. It can come in handy for fluids that tend to foul, leading to regular cleaning. Shell and tube heat exchangers are typically installed with baffles, with even spacing throughout the tube side. Baffles distribute fluid flow evenly in the tube, forcing the flow tog across instead of parallel to the tubes, promoting turbulence in the flow. Baffles improve the heat transfer rate. Unfortunately, these baffles can have an unwanted effect. Flow velocities may vary around the baffles, and stagnant areas may occur. Such areas are very susceptible to fouling [25, 26].

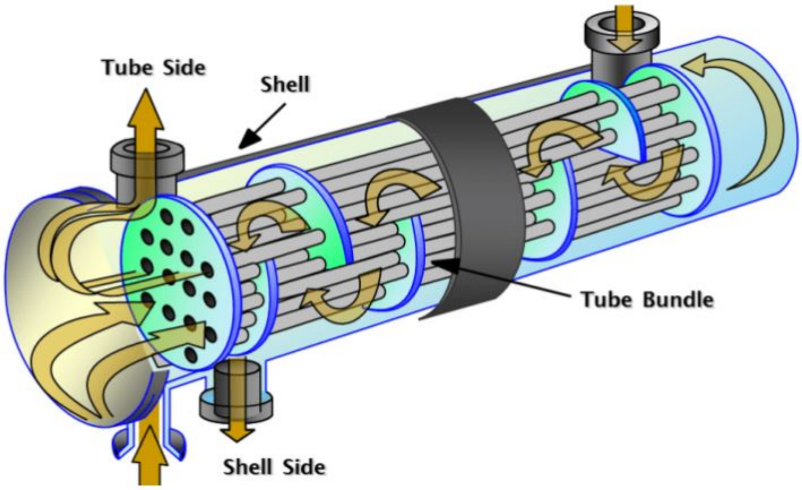


Figure 4.1: Shell and tube heat exchanger [27]

4.1.2 Plate heat exchanger

PHE consists of thin plates in series where hot and cold fluid travel in channels transferring heat through the plates, Figure 4.2. It is generally a simple heat exchanger to deal with and works well with small temperature differences. It is easy to maintenance, repair, and due to its construction, it has the flexibility to increase/decrease heat transfer surface area [22, 28]. There are, in general, four types of PHE: gasketed, brazed, welded, and semi-welded. Welding the PHE will result in possibilities for higher pressure and reduced chance of leaking, but on the other hand, make it more complicated to do maintenance and reduce its flexibility [23]. Gasketed PHE is the most common one, and such PHE usually has operation limitations for temperature and pressure to be less than 160 °C and 25 atm, respectively [23]. Not complying with these limitations puts the PHE at high risk for leaking. Typical thickness of plates usually varies between 0,5 and 1,2 mm, with a gap of 1,5 to 5 mm [29, 18, 19, 20].

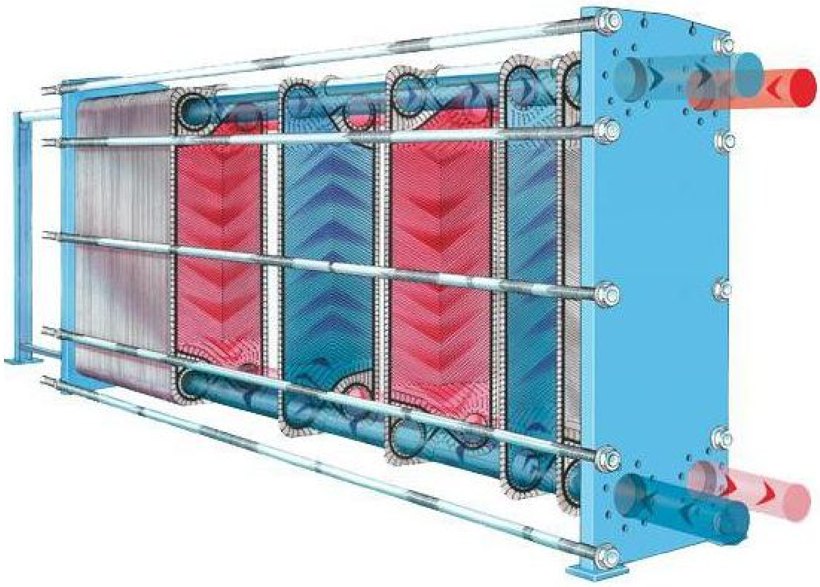


Figure 4.2: Plate heat exchanger [30]

As the plate material greatly affects the performance of the heat exchanger, it is the most important, but also the most expensive part of the PHE [23]. It is desired to find a material that can withstand fouling, such as corrosive resistant materials if corrosion is the case, while still maintaining high thermal conductivity. Being able to withstand the cleaning method is naturally an essential property as well. There are a wide variety of plate materials that can be picked, depending on the situation and use. Standard plate materials are corrosive resistant plates based on stainless steel, Hastelloy, and noncorrosive materials such as titanium alloys [29, 17, 18].

Gaskets are there to make a proper seal between the plates preventing leakage, and generally do not have any other significant function. Therefore, the selection of gaskets is mainly based on them being able to withstand temperature and pressure. Gaskets should also be compatible with the fluid, avoiding corrosion. Typical material of such gaskets are elastomers, such as bytul, nitrile, and silicone. The two first ones can be used for a wide range of organic materials and minerals, with a temperature range of 135 °C and 155 °C, respectively. Silicone gaskets can be used in more corrosive liquids, and function up to 180 °C [23, 18, 29].

The plates may be flat, although in most cases, they are corrugated. The corrugations alter both the HTA and the flow distribution. There are many different types of corrugations, and they may excel in different situations. Some of them are presented in Figure 4.3. The most commonly used design is the chevron plate [23, 21].

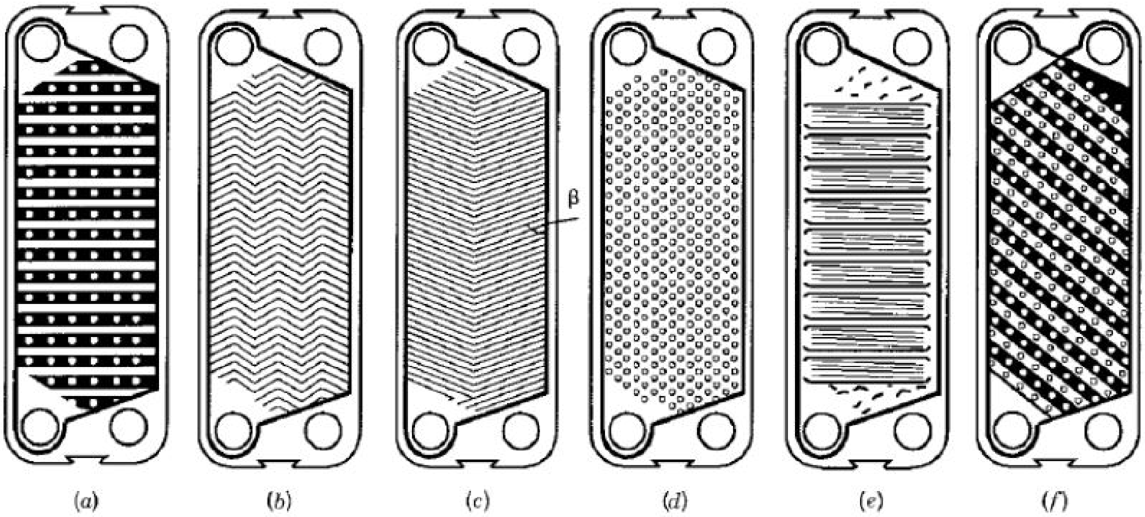


Figure 4.3: Different plate designs. (a) washboard, (b) zigzag, (c) chevron or herringbone, (d) protrusions and depressions, (e) washboard with secondary corrugations, (f) oblique washboard [23]

4.2 Heat transfer

The transfer rate, Q $[kW]$, is how much heat energy can be transferred. Equation 4.1 is used to calculate how much heat can be transferred from a fluid. $\dot{m}$ is the mass flow of the fluid $[kg/s]$, C_p is the specific heat capacity, SHC, of the fluid $[kJ/(kgK)]$, and ΔT is the inlet and outlet temperature difference of the fluid $[K]$.

$$Q = \dot{m} \cdot C_p \cdot \Delta T \tag{4.1}$$

Possible heat transfer through a heat exchanger is calculated by Equation 4.2. Where U is the overall heat transfer coefficient, OHTC $[W/(m^2K)]$, A is the required HTA $[m^2]$, and LMTD is the logarithmic mean temperature difference $[K]$. LMTD is dependant on a correction factor, F, which adjust for the heat exchanger characteristics.

$$Q = U \cdot A \cdot LMTD \cdot F \tag{4.2}$$

4.2.1 Methods of heat transfer

Heat energy can be transferred in three ways, conduction, radiation, and convection. In a heat exchanger, convection and conduction are how most of the thermal energy is transferred. Conduction is the transfer of thermal energy through a material or from one material to another. Fourier's law, Equation 4.3, gives the rate at which thermal energy is transferred by conduction. Where k is the thermal conductivity of the material $[W/mK]$, and d_m is the length of which the heat must travel through the material $[m]$.

$$\frac{Q}{t} = \frac{k \cdot A \cdot \Delta T}{d_m} \tag{4.3}$$

Convection is the process where heat energy is transferred from a moving fluid to another material. It can, for example, be a stream of water flowing along on a piece of metal. Convection can happen by either forced or natural convection. Natural convection, or free convection, is when no external force is present, and the density difference due to temperature changes is controlling the flow. Forced convection is when a pump, fan, or a similar device controls the fluid movement over the HTA, typically in a pipe. The rate of heat transfer by convection can be calculated by Newton's law of cooling, Equation 4.4, where h is the convective heat transfer coefficient CHTC. This value is difficult to determine because its nature is dependant not only on the properties of the fluid but also on the situation. The geometry of the surface, fluid properties, and the nature of the flow of fluid are among the factors affecting the CHTC [31].

$$Q = h \cdot A \cdot \Delta T \tag{4.4}$$

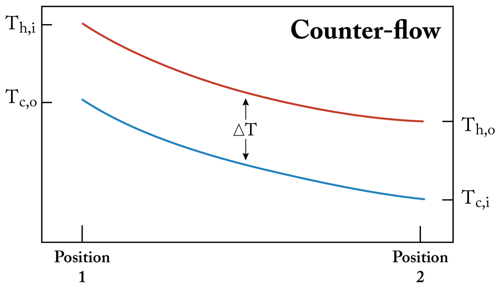
A stagnant layer of fluid film is formed on the material surface, and heat transfer through this layer is happening by conductivity. Turbulent flow usually has a thinner film laminar flow, which is why the CHTC is affected by the nature of the flow. Turbulent flow also forces more of the bulk fluid towards the HTA, increasing heat transfer. The boundary layer is dependant on the viscosity of the fluid, and the type of flow, laminar or turbulent. Due to its difficulty in determining, data is often found empirically.

4.2.2 Logarithmic mean temperature difference

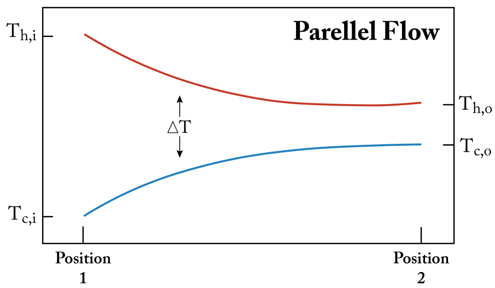
The amount of heat energy available to transfer between two fluids or materials is dependant on the temperature difference. In a heat exchanger, the LMTD is used, Which is calculated by Equation 4.5.

$$LMTD = \frac{\Delta T_2 - \Delta T_1}{ln(\frac{\Delta T_2}{\Delta T_1})} \tag{4.5}$$

A heat exchanger is typically set up as counter-flow or parallel-flow. Counter-flow is when the inlet temperature of the hot fluid is on the same side as the outlet temperature of the cold fluid, Figure 4.4a. Parallel-flow is when inlet temperatures of both fluids are entering on the same side, Figure 4.4b.



(a) Temperature development counter-flow



(b) Temperature development parallel-flow

Figure 4.4: Temperature development counter-flow and parallel-flow

In the Equation 4.5, ΔT_1 and ΔT_2 will differ when calculating for a counter flow or a parallel flow heat exchanger. Due to the nature of the calculations, the LMTD will always be higher for a counter-flow heat exchanger. The equation for heat transfer, Equation 4.2, shows that the larger the LMTD, the larger the heat transfer. Counter-flow heat exchangers are, therefore, more efficient. Equation 4.6 and Equation 4.7 is the calculation of the temperature differences for a counter-flow heat exchanger. Here $T_{h,i}$ is the hot fluid inlet temperature $[°C]$, $T_{h,o}$ is the hot fluid outlet temperature $[°C]$, $T_{c,i}$ cold fluid inlet temperature $[°C]$, $T_{c,o}$ and cold fluid outlet temperature $[°C]$.

$$\Delta T_1 = T_{h,i} - T_{c,o} \tag{4.6}$$

$$\Delta T_2 = T_{h,0} - T_{c,i} \tag{4.7}$$

LMTD method needs a correction factor, F. It is dependant on the geometry of the heat exchanger and the inlet and outlet temperatures of the fluid. The correction factor is to adjust for the departure of true counter-current flow. Its ideal value is therefore 1, where changes from this value are less than 1. A PHE with a single pass is considered as ideal, and the value will, therefore, be 1. However, because of the endplates only exchanging heat in one direction, it is a little bit less than 1. When the number of plates is above 40, the effect of the endplates is negligible [32]. A diagram like Figure 4.5 can find the correction factor. This diagram is for a PHE. The number of transferred units, NTU, is needed to use the chart, calculated by Equation 4.8 as well as the number of passes. In the equation t_o is the outlet temperature $[°C]$ and t_i is the inlet temperature $[°C]$ of the fluid. NTU is based on the fluid with the most considerable inlet/outlet temperature difference [33].

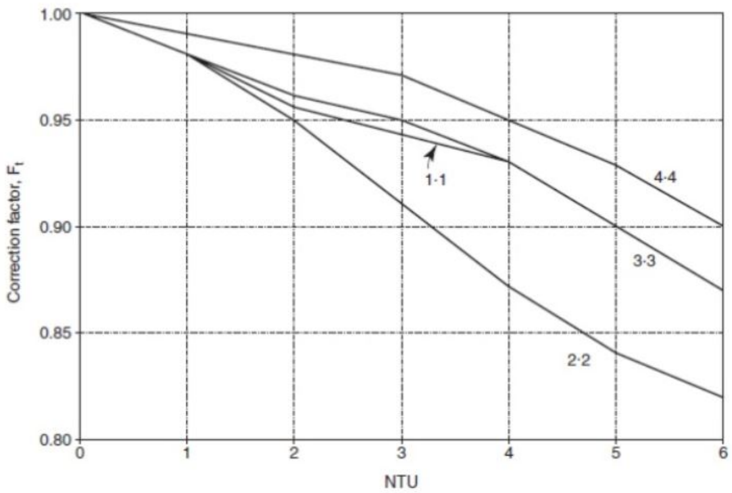


Figure 4.5: Correction factor LMTD [33]

$$NTU = \frac{t_o - t_i}{LMTD} \tag{4.8}$$

4.2.3 Overall heat transfer coefficient

OHTC in a heat exchanger is a value of how much heat energy that can be transferred from one fluid to another, through a material. Additionally to temperature transfer trough the pipe wall, the temperature transfer must go trough a stagnant film layer and fouling layer on both sides. So, its a combination of convection and conduction, see Figure 4.6. Thus it is dependant on how much convective heat the fluids can transfer in the situation and the plate materials ability to transfer heat. It can be challenging to find the correct one. OHTC can be calculated with Equation 4.2. Usually, during the design phase of a heat exchanger, neither the U nor A is known. An estimated U-value can be found with the help of tables for typical U-values for conventional heat exchangers and fluids [34]. Although this will not give an exact U-value, it will provide an idea of how the heat exchanger will perform. A more complicated method of finding the U-value is by studying each flow thoroughly, including calculations of convection heat transfers of each fluid.

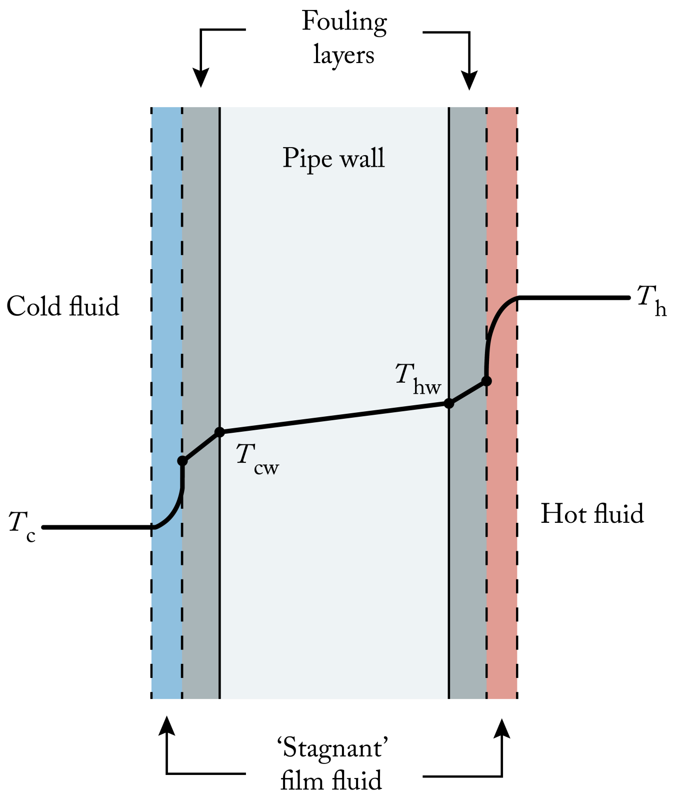


Figure 4.6: Heat transfer and temperature changes through a pipe wall

4.3 Fouling

Fouling can be defined as the deposition of unwanted materials on HTAs. There are different types of fouling, some of the common ones include scaling, growth of algae, particle deposition, corrosion, flocculation, and aggregation. Fouling reduces overall heat transfer but also increases pressure drop due to disruption in fluid flow and narrower flow channels. Fouling produces a layer on top of the HTA. Heat must, therefore, be transferred through the fouling layer by means of conductivity. The thickness and conductivity of the fouling layer are usually unknown, and a fouling factor is therefore used to account for the additional thermal resistance [35, 17]. In some cases, these factors are exaggerated to take into account unforeseen factors that may reduce the transfer capability of the heat exchangers.

As the type of fouling can differ substantially, the measures taken to reduce it will vary depending on the given situation. Generally, high turbulence and velocities, in addition to avoiding areas of stagnant fluid, have a positive effect on the fouling, and smooth surfaces and uniform fluid flow. From a design perspective, typical factors that need to be considered are additional heat transfer surface, additional pressure drop, suitable materials, and easy access cleaning [17]. Depending on the type of fouling, different surface treatment and coatings can be utilized, for example, ultraviolet, acoustic, and electric. If, for instance, biofouling is expected, the use of copper alloys may be implemented, as they are poisonous to some organisms. However, such a solution may not necessarily be suitable, as it can cause increased wear on the system equipment. Also, as it is poisonous to organisms, disposal in the environment is not possible. Some fouling problems may be unpredictable [17].

The rate at which fouling occur is also variable and must be considered in terms of the cleaning frequency of heat exchangers. Cleaning of the heat exchangers is paramount to maintain an acceptable performance level. Cleaning is usually separated into two categories, mechanical and chemical. Mechanical cleaning can be scraping of the surface and pressurized air, and is generally more environmentally friendly, although it might damage the equipment. Examples of chemical cleaning are acids, solvents, and detergents. Cleaning products can be challenging to dispose of due to environmental aspects. The cleaning method depends on the equipment and fouling type. Fouling factors usually are 20-25% compared to the shell and tube heat exchanger due to ease of cleaning and no stagnant areas [17].

4.4 Dimensionless numbers

Reynolds number is a value to describe the flow characteristics of a fluid, if its turbulent, laminar, or transition flow, Equation 4.9. ρ is the density of the fluid $[kg/m^3]$, u is the velocity $[m/s]$, d_h is the hydraulic diameter [m] and μ is the viscosity of the fluid $[kg/ms]$. As seen in the formula, the higher the velocity, the higher the turbulence. The viscosity of fluids is decreasing as the velocity goes up, which enhances the turbulence effect. Higher viscosity gives higher pressure drop and lowers the Reynolds number. Too high viscosity may, therefore, cause the flow to be laminar. For pipes, it is generally accepted that for Reynolds numbers less than 2100, the flow is laminar, while above 4000, it is turbulent. In between these, there is a transition flow. In PHE, turbulent flow can occur at smaller Reynolds number values. Some say the turbulent flow can occur at as low as 100 Reynolds number, while others say it starts at around 400 [21], although this depends on the geometry of the heat exchanger. Another dimensionless characteristic of a fluid is the Prandtl number, which gives an idea of the fluids ability to transfer heat, Equation 4.10, where k is the thermal conductivity of the fluid $[W/m^2K]$.

$$Re = \frac{\rho \cdot u \cdot d_h}{\mu} \tag{4.9}$$

$$Pr = \frac{C_p \cdot \mu}{k} \tag{4.10}$$

Nusselt number is calculated by Equation 4.11. However, in most cases, h is unknown, as this is a value that changes depending on the situation. Alterations of the Nusselt number formula has therefore been made to find the CHTC. Modifications are based on fluid characteristics and its motion, and the Reynolds number and Prandtl number is consequently used. However, these alterations vary in the literature. [23] use a Nusselt number equation where the values are dependant on the angles on the chevron plate, and the Reynolds number, Figure 4.7. For chevron plates with 30 degrees or less angle, it is only differentiated for over and under Reynolds numbers

10. Chevron angles larger than this have specified a transition area, for example between, 10 and 100. It seems then that with 30 degrees or lower chevron angles, a transition phase between laminar and turbulent is negligible. Also, as the Reynolds number is specified as little as 10, it seems that this sort of chevron plates turbulent flow is almost always the case. [18] states that for 30-degree angles on the chevron plate, laminar flow will only occur below Reynolds number 10. Additionally, the formula is only accurate to +− 20%. [32] uses the same table and equation, which originates from [18]. The values m, n, and C_h are constants and are empirical values dependant on chevron angle and Reynolds number. Depending on the chevron plate, turbulent flow in PHE usually occurs from 100-400 Reynolds numbers, which corresponds with values in the table [36].

$$Nu = \frac{h \cdot d_h}{k} \tag{4.11}$$

β	Heat transfer			Pressure drop		
	Re	C_h	n	**Re**	K_p	m
≤ 30°	≤ 10	0.718	0.349	≤ 10	50.000	1.000
	> 10	0.348	0.663	10 - 100	19.400	0.589
				> 100	2.990	0.183
45 °	< 10	0.718	0.349	< 15	47.000	1.000
	10 - 100	0.400	0.598	15 - 300	18.290	0.652
	> 100	0.300	0.663	> 300	1.441	0.206
50 °	< 20	0.630	0.333	< 20	34.000	1.000
	20 - 300	0.291	0.591	20 - 300	11.250	0.631
	> 300	0.130	0.732	> 300	0.772	0.161
60 °	< 20	0.562	0.326	< 40	24.000	1.000
	20 - 200	0.306	0.529	40 - 00	3.240	0.457
	> 400	0.108	0.703	> 400	0.760	0.215
≥ 65 °	< 20	0.562	0.326	< 50	24.000	1.000
	20 - 500	0.331	0.503	50 - 500	2.800	0.451
	> 500	0.087	0.718	> 500	0.639	0.213

Figure 4.7: Constants for heat transfer and pressure for different chevron angles [23]

4.5 Pressure drop

Pressure drop in heat exchangers is naturally desired to keep low, avoiding unnecessarily large and expensive pumps. The pressure drop is due to several reasons. Velocity and geometry of the heat exchanger are two contributors that affect the pressure drop. When the velocity of the fluid increase, the pressure drop rises as well. Narrow channels or pipes can be a reason velocities are high, and thus the pressure drop. Turns and elements such as baffles which force the fluid to change direction is a contributor as well. Extensive fouling is, of course affecting the pressure drop and needs to be dealt with appropriately before the pressure drop is too high. Sometimes it is recommended to design the heat exchanger with minimum shear stress, to account for fouling occurrence. When fouling occurs, the pressure is sufficiently high enough to keep operating the heat exchanger [37]. In some cases, the pressure drop is not only limited by the pump, but also by the fluids. If the total pressure drops to low, fluids may undesirably evaporate at low pressure.

4.6 Economy

The cost of a heat exchanger is mainly divided up into two categories, manufacturing cost, and operating cost. In most cases, the less complicating and the smaller the design, the cheaper the manufacturing cost will be. Operating cost is the cost of pumping the fluids through the heat exchanger, but it also includes the cost of maintenance. Getting the cheapest heat exchanger is not necessarily the most economical solution, a balance between these to must be made [22]. It is usually cheaper to use a smaller number of large plates, than a larger number of small plates. PHE can be inexpensive compared to other heat exchangers as the plates are only pressed together [23].

4.7 Sand filters

Sand filters are a technology that lets you remove solids and reduce the turbidity in water. Such filters can be used for several reasons, one of them achieving clean enough water according to regulation so it can be dispersed off. It can also be used as a step before the water undergoes further cleaning treatments. A sand filter usually functions by pumping water into the top of a tank filled with sand. The water entering is spread evenly across the top layer of sand, to avoid channeling of water. Water will then sink through the sand layers. Larger particles get stuck in between sand grains, while the surface of the sand grains can adsorb smaller particles [38]. [10] recommend using sand filters to remove larger particles so that the finer filters function more effectively.

Occasionally the sand filters will have to be cleaned to stay effective. It is done by something called backwashing. Water is now pumped into the tank from below, rising through the sand and removing the contamination. This water exits at the top of the tank where it can be disposed.

4.8 Carbon filters

Carbon filtering is a method that utilized chemical adsorption to remove sediments and organic compounds, among others. Activated carbon filters can be infused with metals such as magnesium dioxide to react with hazardous compounds like carbon monoxide [39, 40]. Carbon filters are most commonly used for treating water and air. A chemical bond between the surface of the carbon and the matter in the fluid occurs. Carbon is an element with a very high surface area per granule, especially activated carbon, which makes it possible for large amounts of such chemical bindings to occur [41]. Once a binding ha occurred, it is not easily reversible and may require high energy to reverse the process [42]. After a while, the number of bindings possible for each carbon granule will decrease. The carbon in the filters must be then replaced. The particle size of carbon filters can range between 50 microns to 0.5 microns [43]. Carbon filters are among the most effective filters, and work great in combination with other filters, as the last filter, removing PAH among other particles [10].

5 | Methods

The approach and execution of how the objectives and research questions is answered are explained in this chapter. Consideration of the waste heat sources is presented, followed by an introduction of scenarios and finally, a method of calculations.

5.1 Selection of waste heat source

When choosing one of the waste heat sources, each waste heat was considered and compared, with possible solutions and applications. Table 5.1 shows the suggested temperatures that are needed for an application at ENK.

Table 5.1: Suggestive temperatures for possible applications

Application	**Suggestive temperature**
District heating	40 °C
Floor heating	40 °C
Snow and ice melting	40 °C
Ventilation heat	40 °C
Water to air heatfans	40 °C

5.1.1 Heat from slag in piles

It should be possible to expand this to the other ovens as well for increased capacity. A better drainage system underneath the slag piles would also help gather more water, which in turn will increase capacity. For this to be feasible, usage for the extra heat must be in place. As the process is happening in batches, the heat gathered will fluctuate accordingly. This makes it challenging to find good use of the energy gathered, and storage of the heat energy may be necessary.

5.1.2 Cooling of metal in molds

There is a lot of potential energy to be gathered at this point, and as the temperature is very high, it is suitable for many end usages. If the extracted energy could remain reasonably high temperatures, it could for example be used to heat steam, working as a supplement for the steam turbines. Another usage could be heating for both internal heating and district heating, without using low-temperature principles. An assessment of this waste heat has previously been done. In the assessment, it was stated that a solution involving water pipes must take into consideration the high temperature, to avoid the pipes expanding, and be suited for fluctuating temperatures. Even though the energy potential here is excellent, the assessment also concluded that it was not feasible without a proper end-use. Another way of extracting the heat is by using a fan, which directs the hot air above the molds towards water pipes in a set direction. For instance, into a ventilation duct, leading to a heat exchanger. Similar to the slag piles, the heat energy from this step is fluctuating.

5.1.3 Cooling of metal on floor in oven hall

This waste heat source suffers from the same issues as the previous sources, alternating temperatures. Water pipes in the floor might absorb some heat, but the heat absorbed will only focus at one spot at a time. Dividing the floor into several areas may be possible, as water can be circulated in the area where the metal is dissipating the most heat.

5.1.4 Dust filtered exhaust-air from smoke cleaning facility

While the flow is excellent, the temperatures of this waste heat are minuscule, limiting possible applications. Airflow in the ventilation duct is fluctuating as its purpose is to remove dusty air from ovens and molds. Therefore, the waste heat source is inadequate in most cases without installations of heat pumps and such.

5.1.5 Oven gas before scrubbers in the cleaning facility

As the gas from the ovens contains a mixture of different gases and a lot of particles, it might be challenging to utilize it. This is partly due to maintenance, but also because carbon monoxide is a poisonous gas. Cleaning of the gas through the scrubbers makes it easier, however, that also removes a lot of the heat energy. If heat energy is extracted and utilized at this point, heat energy will be decreased for a lot of the following steps of the process, reducing their potential of utilization.

5.1.6 Sludge water from cleaning of the oven gas

A heat exchanger was installed after PAH sand filters at some point but was removed because maintenance turned out to be necessary 1-2 times a week. The water after the coal filters is cleaner and should be easier to utilize, although the flow is way less. Note that if heat energy is extracted from the oven gas, the potential in this step will reduce. After the coal filters, the water only has a temperature of 53°C, limiting the possible uses. Snow and ice melting is a nice usage for this kind of heat along with floor heating and low temperature based district heating. If more coal filters could be installed after the PAH filters, more heat energy could be utilized.

5.1.7 Smoke gas from combustion in gas boiler

Smoke gas might be a great source of waste heat, because of the high temperature, it can be applicable for a wider variety of purposes. It is, however, containing some dangerous gases, which may lead to maintenance and expensive equipment. Further utilization of the heat will result in the steam to condense into water. Condensation in itself is a great deal of energy released, however, for such a gas combination, acidic dew point must be taken into consideration, and more expensive materials may be required.

5.1.8 Flush water tapped from gas boiler

Even though the potential heat energy is lower than other waste heat sources, the water is rather clean, making it easier to utilize from a maintenance perspective. It might not be enough to supply systems solely, but it can supplement the total capacity. It was at some point used as a supplement for district heating, however, the water contained salts that damaged the pipes. A change in how the system is operated now should have removed this problem. Some of the pipes for district heating are still in place, and it should be possible to start utilizing this source again.

5.1.9 Conclusion of the waste heat source selection

A reoccurring problem for many of the WH sources is to find proper end usage. Even though high temperatures may be recovered, it is not feasible to invest in such solutions without anyone or any system needing it. As some of these processes are happening in batches, a solution to continuously deliver a set amount of flow and temperature would have to be made. Water storage in an accumulation tank, set to hold a specific temperature might be a possible solution. It might be necessary with an electric-boiler to keep the temperature up. Such a solution primarily revolves around the cooling of metal and slag, which are happening at fixed periods.

The most promising waste heat source seems to be sludge water from the cleaning of the oven gas after it has been through the carbon filtering process. The water does not have as large temperature as the other sources but is reasonably clean. The flush water tapped from the gas boiler is also clean, and on top of that even hotter. However, the volume flow of this is much lower than that of sludge water. The process can predictably deliver constant heat energy, and as it is a liquid use in a heat exchanger is reasonably straightforward. Filtered sludge water is, therefore, further looked into in this report.

5.2 Waste heat utilization

There are mainly two areas where heat energy can be conveniently utilized at ENK, indoor heating and snow/ice melting. Roughly demands of each application are known. To determine what is possible to achieve, four scenarios will be investigated. The demand of the scenarios can be seen in Table 5.2.

Scenario 1: Indoor heating - 150kW Scenario 1 will solely cover the current indoor heating demand. Filtered sludge water does not have a high temperature, and thus the indoor heating will be based on low temperature solutions, such as floor heating. The required temperature for this scenario is therefore 40 °C, with a $\Delta 5$ °C difference between supply and return water [44, 45]. In many areas, pipes have already been installed and just need a source of hot water to be operated.

Scenario 2: Snow/ice melting - 220kW Scenario 2 will cover the necessary heat for keeping outside areas deemed essential such as walk paths and gates. Temperatures for supply and return water in a snow/ice melting application are 40 °C and Δ 15 °C, respectively [46]. Pipes in most desired areas are already in place.

Scenario 3: Both indoor heating and snow/ice melting - 370kW The third scenario examines the possibility to cover both indoor heating and snow/ice melting at the same time. Both applications can function with the same supply temperature, 40 °C. Return temperature will be a mix of the two.

Scenario 4: Possibility of a larger demand - 600kW Scenario 4 explores the possibility of covering a higher demand than what is currently required for the proposed applications. Supply temperature will still be 40 °C, as a higher temperature than this is difficult to achieve from the filtered sludge water.

Table 5.2: Heat demand scenarios 1-4

	Scenario 1	**Scenario 2**	**Scenario 3**	**Scenario 4**
Heat demand [kW]	150	220	370	600

Covering district heating is a viable solution as well, expanding its current supply, or replace the existing solution in place. The system in place is a low temperature based district heating, with a supply temperature little above 40 °C. Currently, the district heating averagely supplies a capacity of 140 kW. However, a peak value of 300 kW can be expected in the coldest months. These values correspond approximately with scenario 1 and 3, which opens up for the possibility to change the application if needed.

For each scenario, four different sized heat exchangers will be tested, which should give a good indication of what size is needed, and what is possible to achieve. Consideration of which scenario is the most feasible will be done. Thereafter the most beneficial scenario and its associated heat exchanger to invest in can be concluded.

5.3 Sludge water from cleaning of oven gas

For the application, a PHE is chosen. This is because it has a great design to handle conditions where fouling can be an issue, which is a central concern at ENK. PHE is also great at handling low temperatures, which is the situation with the filtered sludge water. When designing a PHE, an approach method is visualized in Figure 5.1. The approach is based on a similar design method explained in [19].

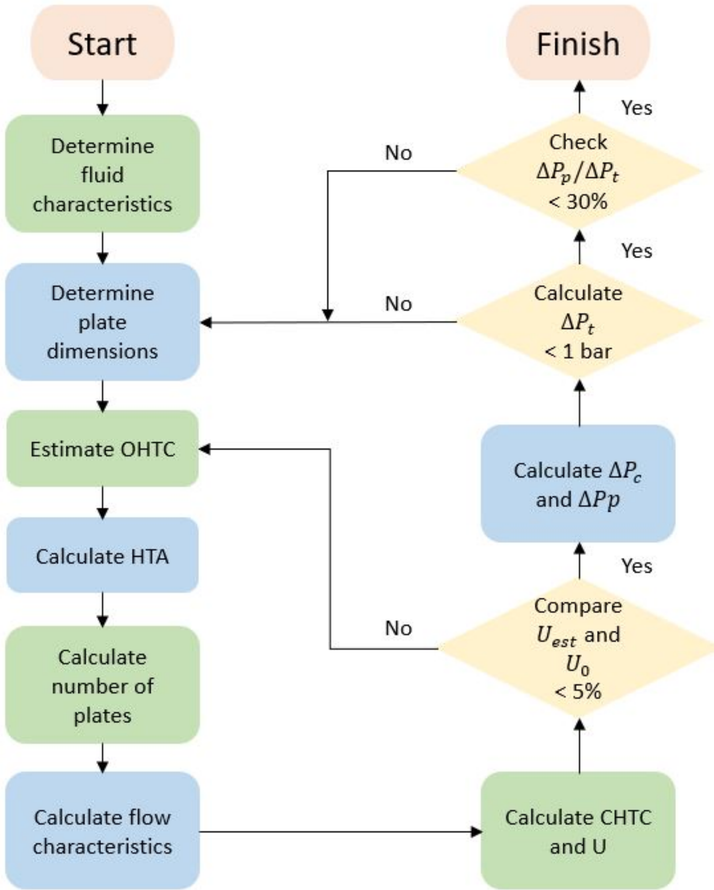


Figure 5.1: Design method PHE flow chart

When designing a PHE, calculations for both fluids must be done. In this report, the following will be the case:

Fluid 1 is the filtered sludge water, the waste heat source being utilized. This is the fluid that will transfer heat.

Fluid 2 Water from the nearby river Kvina will be referred to as Fluid 2, and the fluid receiving the heat energy. This water is used as process water at the factory.

Some assumptions were made to make the calculations more manageable. The assumptions should not have a significant impact on the results.

- Physical properties of fluids are constant. This assumes that specific heat capacity, density and viscosity remain the same throughout the process.

- No heat loss to the surroundings.

- All heat transfer is perpendicular. This applies to both the fluids and the material.

5.3.1 Heat transfer

For each scenario, there is a different heat demand. Rearranging Equation 4.1, the outlet temperature of fluid 1 can be found, which is the filtered sludge water. While the volume flow is known, the mass flow is not. Neither is the SHC and density. Even after going through sand and carbon filtering, the remaining sludge water will still contain some impurities. The altered SHC and density of the fluid are therefore required to be found. As measurements of the stream were not possible, roughly calculations with appropriate assumptions were done instead.

Contents of fluid 1 were based on a few samples from the water after carbon filtering, giving element concentrations in the water. The amount of elements of each sample was varying quite a lot, so an average of the three samples was set as a standard. The samples were given in mg/L and $\mu g/L$. As there were many different elements in the water, only the elements containing more than 0,1 mg/L were considered to have a noticeable effect on the fluid properties. The volume fraction of each element, Φ, was found with Equation 5.1, where m is the mass of the element $[kg]$.

$$\Phi = \frac{m}{\rho} \tag{5.1}$$

Determining properties like SHC, density, and temperature of a fluid Equation 5.2, the rule of mixtures can be used. C_p value can be swapped with either density or temperature, depending on the desired outcome. It can be used for any number of compounds [47]. SHC changes with the temperature, although the change is minimal because of the low temperature differences that are operated. A calculation was done, taking into consideration the SHC temperature change. The change was minuscule, and thus, the SHC is considered to be constant for both fluids. Tables can be found for most common gases and substances, giving different values at different temperatures [48, 49, 50]. The mass flow was calculated when density was determined, Equation 5.3, where $\dot{V}$ is the volume flow $[m^3/h]$.

$$C_{ptot} = C_{p1} \cdot \frac{m_1}{m_{tot}} + C_{p2} \cdot \frac{m_2}{m_{tot}} + ... + C_{pn} \cdot \frac{m_n}{m_{tot}} \tag{5.2}$$

$$\dot{m} = \dot{V} \cdot \rho \tag{5.3}$$

5.3.2 Logarithmic mean temperature

The next step in the approach method is to determine the LMTD. Temperatures of both fluids must then be settled. The second fluid is process water at the factory, which is taken from the nearby river, and will be referred to as fluid 2. The outlet temperature of fluid 2 needs to be at least the temperature of the required application. Inlet temperature will vary depending on the scenario. For floor heating, scenario 1, supply temperature is 40 °C, with a return temperature of 5 °C lower [44, 45]. For scenario 2, snow/ice melting, supply temperature is as well 40 °C but with a return temperature of 15 °C lower [46]. Scenario 3, where both floor heating and snow/ice melting are included, supply temperature will still be 40 °C, but return temperature will be a mix of the two. The inlet temperature will then be found by the rule of mixture, Equation 5.2. The same goes for scenario 4.

Once temperatures of both fluids are settled, LMTD is calculated by Equation 4.5. Heat exchangers in this report will be counter-flow. The LMTD must be altered by the correction factor. For a PHE, the correction factor is found by Figure 4.5 and Equation 4.8. If the number of plates exceeds 40, the correction factor can be neglected [32]. The mass flow of fluid 2 can also be calculated once the fluid temperature is determined, Equation 4.1. The number of passes was set to 1 for all scenarios and configurations.

5.3.3 Heat transfer Area

First time calculating the required HTA with Equation 4.2, both U and A are unknown. A preliminary U is therefore used. Typical u-values for different applications and heat exchangers can be found in tables [34, 51]. Liquid to liquid PHE commonly has a U-value between 1000 - 4000 $W/(m^2 \cdot K)$.

5.3.4 Number of plates

When determining the necessary number of plates, it is required to know the characteristics of the PHE. How many passes will the fluid have, plate material, dimensions of the plate, length and width, and the spacing between the plates. For this report, four sizes of heat exchangers will be compared to determine which fit best for each scenario. Table 5.3 shows the dimensions of the four heat exchanger sizes that are tested. The parameters are visualized in Figure 5.2. In the figure β is the chevron plat angle [°]. Chevron plates with an angle of 30°and 1 mm plate thickness were used for all heat exchangers. A thinner plate thickness will result in better OHTC. Plate thickness is set to 1 mm for all heat exchangers.

Table 5.3: Heat exchanger plate dimensions

	Size 1	**Size 2**	**Size 3**	**Size 4**
Length - Lp $[m]$	0,42	0,74	1,04	1,61
Width - Wp $[m]$	0,11	0,33	0,36	0,47
Port distance vertical - Lv $[m]$	0,45	0,83	1,16	1,78
Port distance horizontal - Lh $[m]$	0,08	0,24	0,24	0,30
Port diameter - Dp $[m]$	0,035	0,091	0,116	0,171

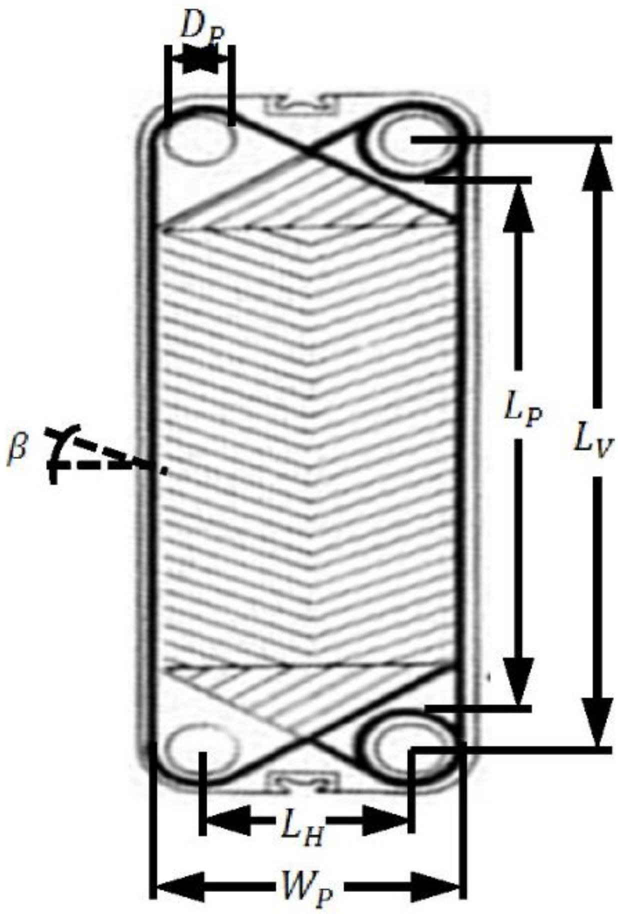


Figure 5.2: Parameters of a chevron plate [23]

The number of plates, N_p, is calculated by Equation 5.4, where A is the necessary HTA $[m^2]$, and A_p is the area of one plate $[m^2]$. The number of plates is always rounded up to the closest whole number.

$$N_p = \frac{A}{A_p} \tag{5.4}$$

The number of channels can be calculated straight afterward. If the number of plates is odd, there will be just as many channels with cold fluid as hot fluid, and the amounts of channels can be calculated by Equation 5.5, where N_c is the number of channels. It is usually cheaper to use a smaller number of large plates, than a larger number of small plates.

$$N_c = \frac{N_p - 1}{2} \tag{5.5}$$

If the number of plates is even, the number of channels is calculated by Equation 5.6 for hot fluid and Equation 5.7 for cold fluid.

$$N_{ch} = \frac{N_p - 2}{2} + 1 \tag{5.6}$$

$$N_{cc} = \frac{N_p - 2}{2} \tag{5.7}$$

5.3.5 Reynolds and Prandtl numbers

To find OHTC, CHTC must be found first, and therefore, flow characteristics are to be calculated. The width of channels affects the flow characteristics and must be settled. Plate gaps from 1,5mm to 4mm with 0,5mm incrementations was investigated. The goal was to find a suitable plate spacing for each size and scenario combination, maintaining a high heat transfer rate and velocity without having too much pressure drop. Flow characteristics are then calculated with dimensionless numbers. Reynolds number is calculated with Equation 4.9. In this formula, d_h is the hydraulic diameter $[m]$, which for a rectangular duct is calculated by Equation 5.8, where s is the plate spacing [m], and W_p is the plate width [m].

$$d_h = \frac{2 \cdot W_p \cdot s}{W_p + s} \tag{5.8}$$

However, to be able to do the Reynolds calculations, the fluids viscosity must be found. As both fluids were mixtures of different components, a simple value could not be found in a table but had to be calculated, similar to the SHC and density. Fluid viscosity can be done with Einstein's viscosity equation, Equation 5.9. For small concentrations of particles, $\Phi < 0,02$, it gives satisfactory results for this report [11]. μ_{mix} is the fluid viscosity of the mixture, while μ_f is the viscosity of the fluid containing particles, which is, in this case, water. The viscosity of water was calculated at the bulk mean temperature of the fluids.

$$\mu_{mix} = \mu_f \cdot (1 + 2,5 \cdot \Phi) \tag{5.9}$$

Velocity in each channel u_c is calculated by Equation 5.10. $\dot{m}$ is the total mass flow $[kg/s]$. Cross-section of the channel is d_c $[m^2]$, ρ is the density $[kg/m^3]$, and N_c is the number of channels. Higher velocities give higher pressure drops. Finding a balance between acceptable pressure drop and velocity is key to a good heat exchanger. Typically velocities may be between 0,3 and 0,9 m/s, which is used as guiding limits in the calculations [36]. Reynolds can now be calculated.

$$u_c = \frac{\dot{m}}{\rho \cdot d_c \cdot N_c} \tag{5.10}$$

Further on, the Prandtl number is to be found, Equation 4.10. Therefore, the fluids thermal conductivity must be calculated with the Maxwell thermal conductivity equation, Equation 5.11 [15, 52]. Where k_{mix} is the conductivity for the mixture $[W/m \cdot K]$, k_p the conductivity of the particles $[W/m \cdot K]$, k_f conductivity of the fluid $[W/m \cdot K]$ and Φ is the volume fraction of the particles.

$$k_{mix} = (\frac{k_p + 2 \cdot k_f + 2 \cdot (k_p - k_f) \cdot \Phi}{k_p + 2 \cdot k_f - (k_p - k_f) \cdot \Phi}) \cdot k_f \tag{5.11}$$

5.3.6 Overall and convective heat transfer coefficient

Formulas for the nusselt number varies depending on the literature, partly due to the different corrugations patterns available for the plates. Multiple correlations were tested, and Equation 5.12 was reoccurring in several articles and took into consideration plate design. Therefore it will be the one used for further calculations in this report. μ_w is the fluid viscosity at wall temperature $[kg/m{\cdot}s]$. C_h and n are constants that depend on the Reynolds number and chevron angle, given in Figure 4.7. Chevron angle in the calculations was set to 30°. For simplicity, μ_w is assumed to be the mean temperature between the mean temperature of both fluids. When the nusselt number is found, Equation 4.11 can be rearranged to find CHTC.

$$Nu = C_h \cdot Re^n \cdot Pr^{1/3} \cdot (\frac{\mu}{\mu_w})^{0,17} \tag{5.12}$$

OHTC is calculated by Equation 5.13. h_1 and h_2 are the CHTC for fluid 1 and 2 $[W/(m^2 \cdot K)]$, t_p is the thickness of the plate [m], k_p is the thermal conductivity of the plate $[W/(m \cdot K)]$, and R_{f1} and R_{f2} are the fouling resistances for the fluids $[(m^2 \cdot K)/W]$. With the available information and calculations made for the fluid properties, the fluids seem quite similar, and fouling resistance to that of river water is used for both of them. [53].

$$U = \frac{1}{\frac{1}{h_1} + \frac{t_p}{k_p} + \frac{1}{h_2} + R_{f1} + R_{f2}} \tag{5.13}$$

Plate material chosen for the PHE is titanium, which has a thermal conductivity of 24,5 $W/m{\cdot}K$ [48]. This material is non-corrosive, and should, therefore, reduce fouling by corrosion. The thickness of the plate is set to 1 mm for the calculations. Thickness of the plate impacts the OHTC. The lower the thickness is, the higher the OHTC.

When all calculations are completed, the final U-value is compared to the estimated one, Equation 5.14. U_{est} is the estimated OHTC, while U_0 is the calculated one. All calculations up to this point must be repeated if the calculated and estimated value is diverting too much. For the calculations, a difference of less than 5% was desired.

$$\frac{U_0 - U_{est}}{U_0} \tag{5.14}$$

5.3.7 Pressure drop

The pressure drop across channels ΔP_c and pressure drop across the ports in the distribution pipes ΔP_p make up the main contributors to pressure drop in a PHE. The total pressure drop is calculated by 5.15 $[Pa]$. ΔP_c is the channel pressure drop $[Pa]$, and ΔP_p is the port pressure drop $[Pa]$. ΔP_c is found by Equation 5.16. f is the friction factor, L_v is the vertical distance between the ports $[m]$, G_c is the fluid mass velocity in the channel $[kg/(m^2 \cdot s)]$, N_{pa} is the number of passes and μ_w is the viscosity at wall temperature $[kg/ms]$.

$$\Delta P_t = \Delta P_c + \Delta P_p \tag{5.15}$$

$$\Delta P_c = 4 \cdot f \cdot \frac{L_v \cdot N_{pa}}{d_e} \cdot \frac{G_c^2}{2 \cdot \rho} \cdot (\frac{\mu}{\mu_w})^{-0,17} \tag{5.16}$$

Different methods for determining friction factor, f, is varying in literature. Equation 5.17, which is the mulley correlation, is the one [21] recommends and will be used in this report.

$$f = (\frac{\beta}{30})^{0,83} \cdot [(\frac{30,2}{Re})^5 + (\frac{6,28}{Re^{0,5}})^5] \tag{5.17}$$

In a PHE, the pressure drop across all channels is similar, and calculations for one of them are sufficient. This is because a stream goes to the path of least resistance, least pressure required. If several parallel paths need different pressure, the stream will only be able to flow through the path where the pressure drop is less than the available pressure. If several parallel paths require the same pressure, the stream will be able to flow through all paths simultaneously, distributed evenly between them.

Pressure drop through the ports is calculated by Equation 5.18. Where N_{pa} represents the number of passes, and G_p is the mass velocity of the fluid through the ports $[kg/(m^2 \cdot s)]$. G_p is calculated by Equation 5.19. D_p is the port diameter $[m]$, and $\dot{m}$ is the total mass flow through the channels $[kg/s]$. The pressure drop across the ports is recommended to be less than 10% of the total pressure drop, although it may be up to 30% [54, 23, 37].

$$\Delta P_p = 1,4 \cdot N_{pa} \cdot (\frac{G_p^2}{2 \cdot \rho}) \tag{5.18}$$

$$G_p = \frac{\dot{m}}{\pi \cdot \frac{D_p^2}{4}} \tag{5.19}$$

6 Results and Discussion

The results are presented and discussed in this chapter. First results concerning the fluid properties and capabilities are presented. Then each scenario is gone through before they are all compared in the end.

6.1 Sludge water from cleaning of oven gas

6.1.1 Fluid properties

When comparing the river water to the filtered sludge water, most of the element concentrations are quite similar. Table 6.1 shows the concentration ratio of some elements between the fluids. The table only shows the ratios of the elements that have more than 0,1 mg/L. It is clear that for some metallic elements such as magnesium and potassium, the values have increased quite a bit. However, there was only one sample measurement of the process water and three of the filtered sludge water. Among the few filtered sludge water samples, values of some elements varied by three times the amount. The same would most likely be the case for the river water. An increase of element concentration in fluid 1 is still apparent.

Table 6.1: Ratio of element concentration fluid1/fluid2

Element	**Ratio (Fluid1/Fluid2)**
Ca (Calcium)	0,92
Cu (Copper)	2,20
Fe (Iron)	1,02
K (Potassium)	6,43
Mg (Magnesium)	7,82
Mn (Manganese)	1,78
Na (Sodium)	3,38

When designing a PHE, it is essential to know the properties of the fluids that are to be used. The characteristics of the fluids are paramount to the PHE operation, both in terms of heat transfer rate and maintenance. Thus the properties of fluid 1 and fluid 2 have been calculated and shown in Table 6.2, where Φ is the total volume fraction. For fluid 1, there is only a small change in most of the fluid properties, compared to pure water. Density, viscosity, mass flow, and thermal conductivity have all been slightly increased. The minuscule change can be due to the volume fraction of the elements were only 0,6% in total. SHC has, on the other hand, decreased, but these values are also similar to water. The properties of fluid 2 are pretty much the same as clean water. The elements in the water are only 0,1% compared to 0,6% in fluid 1. For these fluids, it seems the volume fraction of elements is too small to noticeably increase the thermal conductivity, compared to nanofluids, where the volume fraction might be 5% [14]. Mass flow of fluid 2 is not shown in this table, as this value will change depending on the heat demand.

Table 6.2: Fluid 1 properties

Properties	**Fluid 1**	**Fluid 2**	**Unit**
Φ elements	0,621	0,111	%
Density	996,28	996,89	kg/m^3
Viscosity	0,000797	0,000787	$kg/(m \cdot s)$
Thermal conductivity	0,611	0,602	$W/(m \cdot K)$
SHC	4,165	4,182	$kJ/(kg \cdot K)$
Mass flow	6,365	-	kg/s

With the use of Maxwell thermal conductivity equation, Equation 5.11, the conductivity value resulted in barely above the thermal conductivity of water, for both fluid 1 and 2. The small increase is likely due to the low particle concentration of the elements, even though they have a significantly higher thermal conductivity. Although theoretical calculations have proved to underestimate the measured thermal conductivity, the calculated value of conductivity will be used, as the change in thermal conductivity was small [15, 13].

6.1.2 Energy potential filtered sludge water

In Table 6.3, energy potential, Q, in the filtered sludge water is calculated, Equation 4.1. In the calculation, the water temperature is decreased from the inlet temperature of 53 °C to that of seawater, 6 °C, to see the fluid stream's maximum potential. As shown in the table, the energy potential is calculated to 1246kW. This is twice the value that is needed for scenario 4, which was 600kW.

Table 6.3: Energy potential filtered sludge water

Properties	**Value**	**Unit**
Mass Flow	6,365	$kg/(s)$
Δ T	47	°C
SHC	4,165	$kJ/(kg \cdot K)$
Q	1246	kW

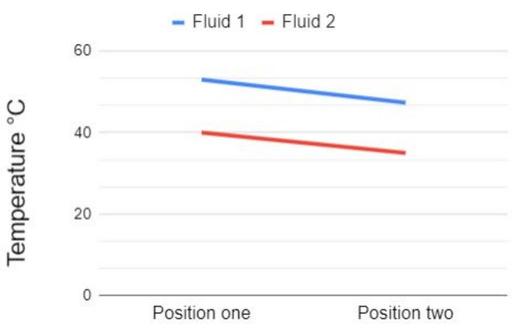
6.1.3 Fluid temperatures

Table 6.4 shows the inlet and outlet temperatures of the fluids for the four demand scenarios. For the first two scenarios, the outlet temperature of fluid 1 is not much smaller than the inlet temperature, 53 °C. However, for the last two, a more significant difference can be seen as the demand is noticeable higher. The outlet temperature of fluid 2, the stream going out to the application, is 40 °C for all scenarios. Inlet temperatures depend on the heat loss in the given applications, Equation 5.2.

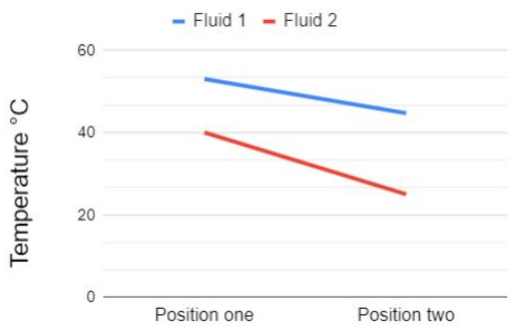
Table 6.4: Inlet and outlet temperatures for the fluids

	Fluid 1 inlet	**Fluid 1 outlet**	**Fluid 2 inlet**	**Fluid 2 outlet**
Scenario 1	53 °C	47,34 °C	35,00 °C	40,00 °C
Scenario 2	53 °C	44,70 °C	25,00 °C	40,00 °C
Scenario 3	53 °C	39,03 °C	31,72 °C	40,00 °C
Scenario 4	53 °C	30,35 °C	29,31 °C	40,00 °C

To have a proper heat transfer between the fluids, the outlet temperature of fluid 1 must be higher than the inlet temperature of fluid 2. For scenario 1, 2, and 3, this is the case. However, for scenario 4, the outlet temperature of fluid 1 is barely above the inlet temperature of fluid 2. The small difference means that, at that part of the heat exchanger, almost no heat is transferred between the fluid. Figure 6.1a, Figure 6.1b, Figure 6.2a, and Figure 6.2b shows the temperature development in the heat exchanger for scenario 1-4. As these are counter-flow heat exchangers, position 1 is the inlet of fluid 1 and outlet of fluid 2. Position 2 is the outlet of fluid 1 and inlet of fluid 2. It is clear that in scenario 4, the heat transfer effect at one end of the heat exchanger is minuscule. Thus it is not reasonable to get the desired heat transfer without requiring colossal heat exchangers.

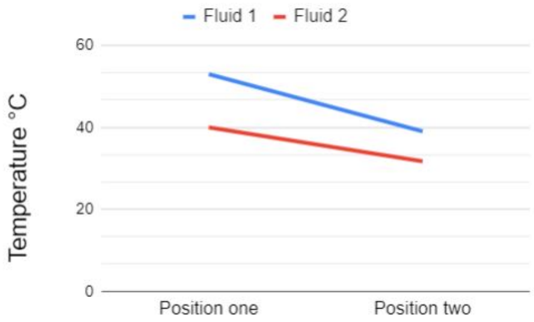


(a) Temperature development scenario 1

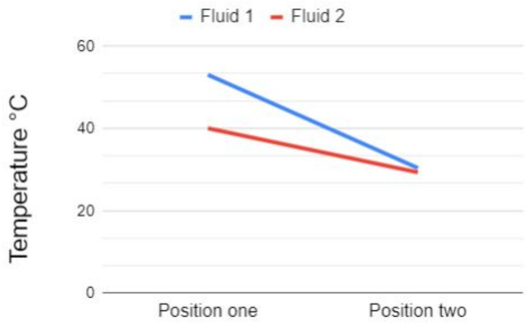


(b) Temperature development scenario 2

Figure 6.1: Temperature development scenario 1 and 2



(a) Temperature development scenario 3



(b) Temperature development scenario 4

Figure 6.2: Temperature development scenario 3 and 4

The temperature development is reflected in the LMTD for the scenarios, Table 6.5, shows that scenario 4 has a LMTD value of just 4,6 °C. Except for scenario 4, the LMTD is pretty even for the other scenarios. Although scenario 3 has twice as high LMTD as scenario 4, it is quite low. Scenario 2 has the largest LMTD, due to snow/ice melting having a much lower return temperature than the other scenarios, $\Delta 15$ °C. The low LMTD indicates that heat exchangers for those scenarios might be undesirable large. Correction factor F is rather stable for all configurations, varying between 0,97 and 0,99. Therefore it does not have too much impact on the LMTD, which is expected for a single pass counter-flow PHE.

Table 6.5: Energy potential filtered sludge water

	LMTD	**Mass flow fluid 2**
Scenario 1	12,54 °C	7,17 kg/s
Scenario 2	15,79 °C	3,51 kg/s
Scenario 3	9,69 °C	10,68 kg/s
Scenario 4	4,60 °C	13,42 kg/s

A direct result of the LMTD is the mass flow of fluid 2. In Table 6.5, it is seen that the larger the LMTD, the smaller the required fluid 2 mass flow. When the mass flow of both fluids are more or less the same, equal plate gaps can be installed without concerns of one of the fluid having drastically larger velocities and pressure drop than the other. However, for scenario 2, the mass flow of fluid 2 roughly half to that of fluid 1, which is 6,37 kg/s. For scenario 4, the opposite is the case, where fluid 2 has approximately double the mass flow to that of fluid 1. Instead of alternating plate gaps for fluid 1 and fluid 2, scenario 2 can lower the mass flow of fluid 1, although this would result in a larger HTA.

6.2 Scenario 1 selection

When going through the results for scenario 1, all four heat exchanger sizes were tested with plate gaps from 1,5-4 *mm*. A plate gap could then be found while still maintaining velocities, pressure drop, etc. within their desired values. This procedure was done for each scenario.

6.2.1 Required heat transfer area

In Figure 6.3, the required HTA is compared for the different heat exchanger sizes and plate gaps. The heat exchanger sizes can be found in Table 5.3. The HTA differences were rather small, and in most cases, the area varied roughly from 10 m^2 to 12 m^2. It is seen that the size 1 heat exchanger required the highest HTA. When the heat exchanger got larger, the HTA decreased.

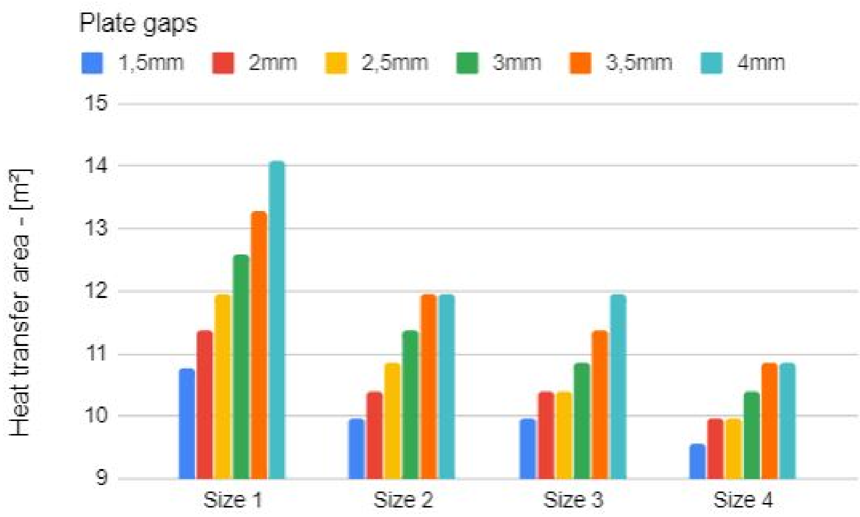


Figure 6.3: Required surface area scenario 1

6.2.2 Number of plates

Figure 6.4 shows the number of plates for scenario 1 heat exchangers. It clearly shows that there is a significant increase in the number of plates, the smaller the heat exchanger. The rise in plate number is especially seen in the case for the size 1 heat exchanger, where the number of plates is above 200 for all plate gaps. Due to the big difference, the graph is limited to show a maximum of 75 plates. Even though the required HTA was not that different, the plate size difference significantly impacted the number of plates. The other heat exchanger sizes have roughly the same amount of plates regardless of the plate gaps. It is clear that the size 1 heat exchanger may not be suitable for this scenario. On the other hand, the size 4 heat exchanger has very few plates, with very little change when the plate gap increases, only in the range of 12-14 plates. It might indicate that it is most likely too large, leaving an unnecessarily large footprint.

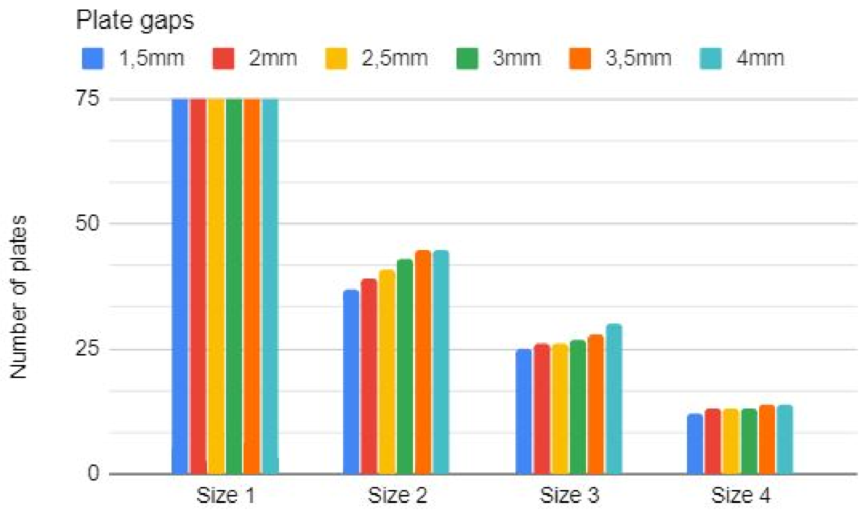


Figure 6.4: Number of plates for scenario 1

6.2.3 Overall heat transfer coefficient

OHTC is compared in Figure 6.5. The majority of OHTC has a value in the range of 1000 to 1200 $W/(m^2K)$. Reducing the plate thickness will increase the OHTC considerably. The results show that OHTC is always the largest for size 4 for similar plate gaps. The larger the plate size, the fewer plates are needed to reach the necessary heat transfer surface area, which means the same amount of fluid needs to be divided on fewer channels. Fewer channels, higher velocities. When the velocity increases, so do the turbulence, which gives a higher OHTC. It also shows that the small plate gaps provide the best OHTC, which is for the same reason, tighter channels, higher velocities. Typical OHTC values for liquid/liquid PHE are 1000-4000 $W/(m^2K)$, so in general, the heat exchangers configurations are performing in the lower end of the spectrum.

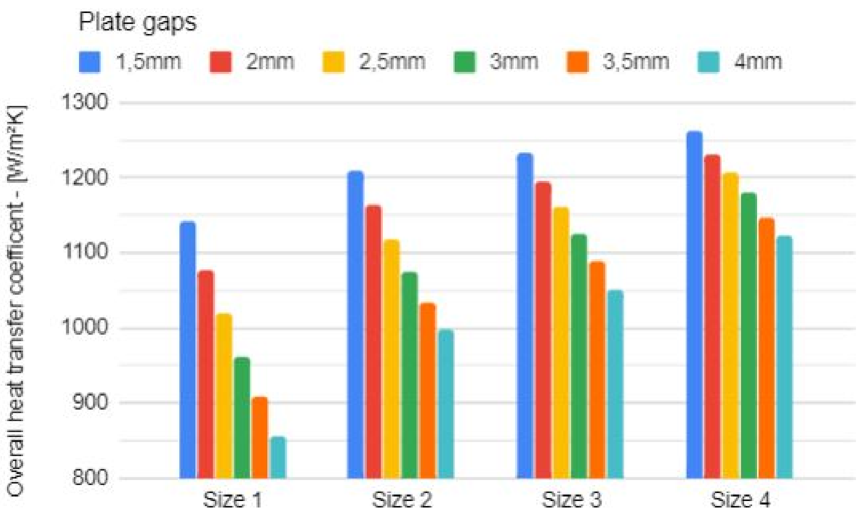


Figure 6.5: Overall heat transfer coefficient for scenario 1

6.2.4 Velocities

Velocities for fluid 1 and 2 are presented in Figure 6.6 and Figure 6.7, respectively. In the figures, max and min limit for the velocities has been set to 0,9 and 0,3 m/s to maintain high turbulence and avoiding high pressure drop [36]. Velocity profiles for the two fluids are quite similar, with slightly higher values for fluid 2. Yet again, the size 1 heat exchanger performs poorly, with almost all velocities below the desired 0,3 m/s for both fluid flows. Although the velocities were low for this size, the Reynolds number was still above the threshold to keep the fluid turbulent. The low velocities are a result of the high amount of plates that the size 1 heat exchanger required. The graph reveals that the velocity changes quite a bit, depending on the plate gaps. Size 3 of the heat exchangers seems to perform well for all plate gaps in this scenario. Size 2 needs small plate gaps, and size 4 needs bigger plate gaps to be within the desired values. High velocities in itself are not necessarily bad, but it may cause problems as it results in a rise in pressure drop.

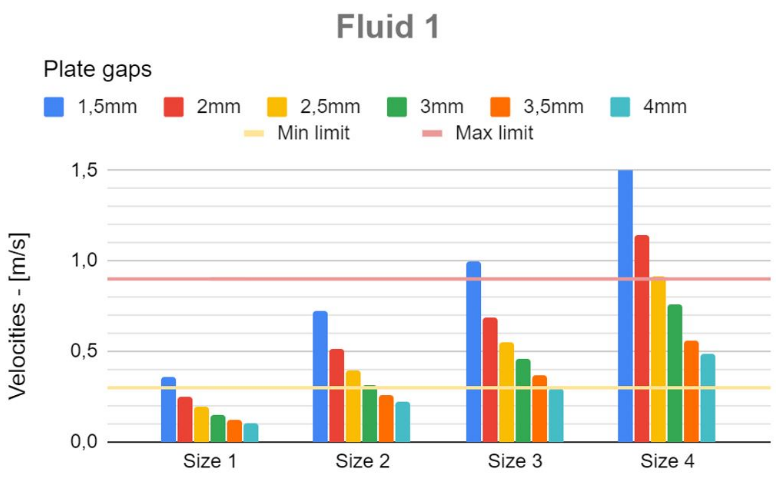


Figure 6.6: Velocities fluid 1

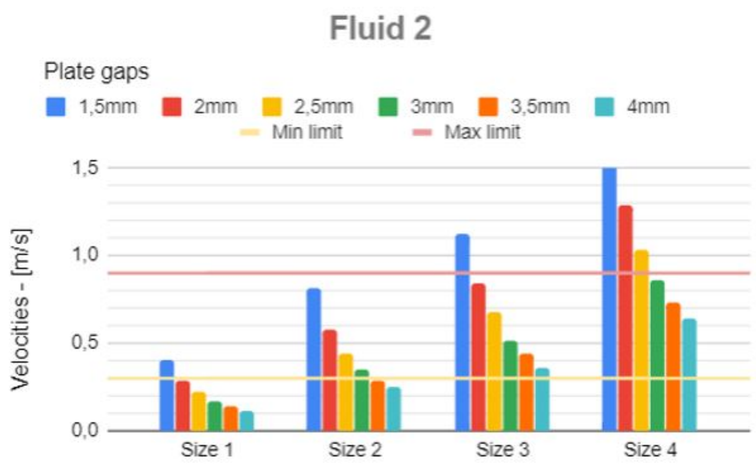


Figure 6.7: Velocities fluid 2

6.2.5 Pressure drop

The total pressure drop for fluid 1 can be seen in Figure 6.8, and for fluid 2 in Figure 6.9. The maximum limit in the figure is 1 *bar* to avoid the use of large pumps. Pressure drop for the size 4 heat exchanger, with a 1,5 *mm* plate gap, exceeded the desired 1 bar pressure drop for both fluids, with values of 1,76 *bar* and 3 *bar*, respectively. The graph is limited to 1,1 bar to enhance the variations of the other heat exchangers. All other cases are comfortably below the 1 *bar* limit, although fluid 2 is quite close for a 2 *mm* plate gap. For the size 1 heat exchanger, the pressure drop development is pretty stable. The other sizes have a more noticeable decrease of pressure drop as the plate gap increases. For this scenario, most heat exchangers deliver within the given parameters for most of the cases.

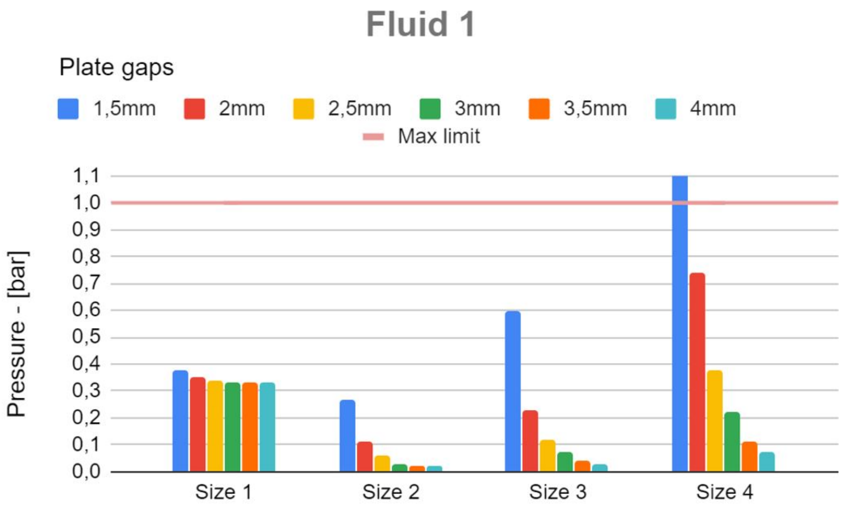


Figure 6.8: Total pressure drop fluid 1

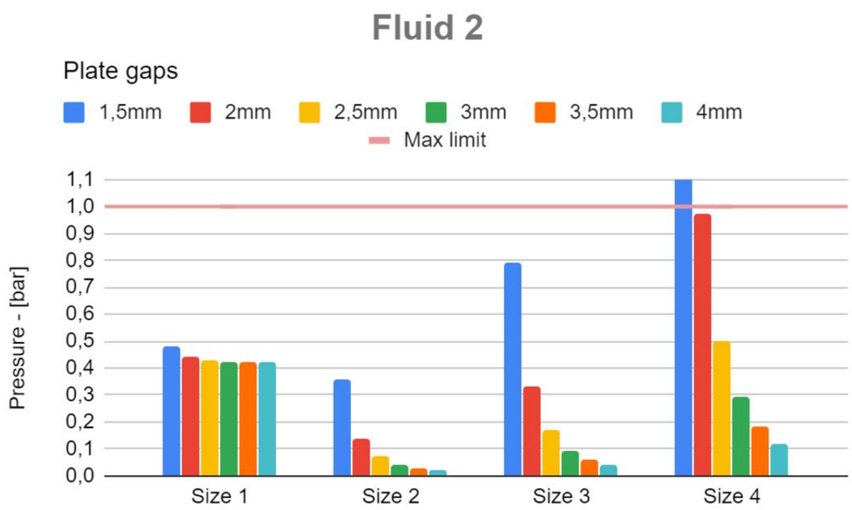


Figure 6.9: Total Pressure drop fluid 2

Finally, the pressure drop across the ports relative to the total pressure drop can be seen in Figure 6.10 and Figure 6.11. There are two limits in this graph, one for 10% and one for 30%. Ideally, this should be as low as possible, but may be up to 30% in some cases [23]. For the size 1 heat exchanger, the value is way too high, close to 100% of the pressure drop. Although this is not the desired value, it actually doesn't matter too much, as the total pressure drop is still lower than the recommended on those cases. However, it may indicate that the dimensions of the heat exchanger are not suitable. Because of the high ratio for heat exchanger 1, the graph is limited to 45%. Looking at size 2, the value is quite frankly higher than desired, but at the same time, the total pressure drop is minuscule, which then means that the pressure drop over the ports also will be low, even though it is higher than that of the channel pressured drop. For the size 3 and 4 heat exchangers, the ratio is below 10% for all plate gaps. Thus this graph generally only needs to be taken into consideration if the total pressure drop is close to the maximum desired pressure drop.

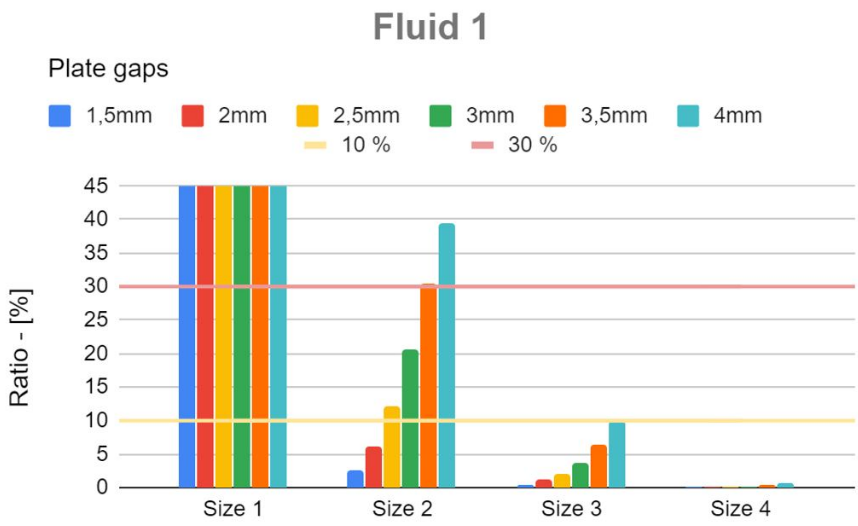


Figure 6.10: Port pressure drop relative to total pressure drop for fluid 1

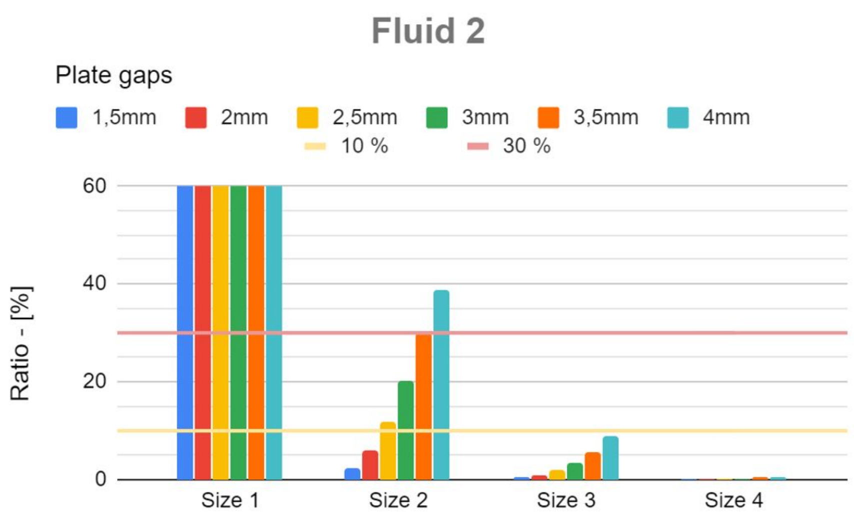


Figure 6.11: Port pressure drop relative to total pressure drop for fluid 2

6.2.6 Selection

Summing it all up, both size 2 and 3 heat exchangers have several plate gap options that fit this scenario well. Size 2 will have possible plate gaps for 1,5 - 3 mm. However, when getting close to a 3 mm plate gap, the port pressure drop is between 10-30 % of the total pressure drop. The total pressure drop for those configurations is really low, so it is not an issue. To avoid clogging of the pipes, 2 mm or 2,5 mm plate gaps seem best for this size. Velocities are still satisfactory high, above 0,4 m/s for both fluids. For the size 3 heat exchanger, all plate gaps are viable. Even with the velocities over the recommended 0,9 m/s, the total pressure drop is still below the desired limit. When considering the risk of fouling in narrow channels, 2-3,5 mm plate gaps could be excellent options for this heat exchanger. There are also some viable configurations for the size 4 heat exchanger in this scenario. When the plate gaps are larger than 2 mm, the pressure drop is below the desired limit of 1 bar. Velocities are overall quite high even for large plate gaps, due to the few channels. However, this heat exchanger may give an unnecessary large footprint while still having the same performance as the smaller sizes. A high amount of plates and low channel velocities makes the size 1 heat exchanger unsuited for this scenario. The possible heat exchangers for this scenario is as follows:

- Size 2 - Plate gap: 2 - 2,5 mm
- Size 3 - Plate gap: 2 - 3,5 mm
- Size 4 - Plate gap: 2,5 - 4 mm

6.3 Scenario 2 selection

Graphs for scenario 2 can be found in Appendix A.1. Heat exchanger 1 has the same problems for this scenario, as it did for scenario 1. Size 2 is starting to struggle with low velocities as well, especially on fluid 2 side. Fluid 2 in this scenario has half the mass flow of fluid 1, and the velocities are consequently much lower if no concerns to varying the plate spacing are being taken. This is reflected in the velocity profiles for the fluids, roughly half the velocities for fluid 2. A plate gap of 1,5 mm does have velocities that are above the 0,3 m/s limit. However, as the plate gap is narrow, there is a higher risk of clogging. Size 3 is marginally better, with a plate gap of up to 2,5 mm can be accepted without the velocities being too low. Neither of these two exchangers is subjected to high pressure drops. Heat exchanger 4 surpasses the 1 bar limit with a pressure drop of 1,4 bar when the plate gap is 1,5 mm. Plate gaps of 2-3 mm, however, have great performances. Possible heat exchanges for this scenario is as follows:

- Size 3 - Plate gap: 2 - 2,5 mm
- Size 4 - Plate gap: 2 - 3 mm

6.4 Scenario 3 selection

Graphs for scenario 3 can be found in Appendix A.2. The OHTC value is starting to drop in this scenario compared to the others. LMTD is also lower, which of course, lowers the effectiveness. Comparing the development from scenarios 1 to 2, it does not surprisingly indicate that the larger the demand, the larger the necessary heat exchanger. When the two scenarios' heat demand is combined, the development continues, and as a result, both size 1 and 2 heat exchangers struggle with both high plate numbers and low velocities. The size 3 heat exchanger performs slightly better but needs narrow plate gaps to keep the velocity at acceptable levels. Even then, it is only barely above the 0,3 m/s limit. The size 4 heat exchanger is not performing that much better. A maximum plate gap of 2,5 mm can be used, but then fluid 1 velocities are a little below the comfortable limit. Thus the 2mm plate gap for this exchanger is suitable for the application. It is observed that for scenarios 1 and 2, the recommended heat exchangers have a number of plates in the range of 25-45. The size configuration for size 4 heat exchanger in this scenario is 42, so it seems that this is a reoccurring trend. Simultaneously the OHTC for the recommended heat exchanger configurations has been at around 1100 W/m^2K. Possible heat exchanges for this scenario is as follows:

- Size 4 - Plate gap: 2 mm

6.5 Scenario 4 selection

Graphs for scenario 4 can be found in Appendix A.3. For scenario 4, none of the selected sizes is a suitable fit, as none of them are large enough. There has been an increase in heat exchanger size for each scenario, and scenario 4 has a considerably higher heat demand than in scenario 3, with 600 kW compared to 370 kW. This reflects in the values of scenario 4. Where in scenario 1 and 2 required HTA has been around 10-13 m^2, and scenario 3 with 35 m^2, scenario 4 have easily 4-6 times the amount of scenario 3. A considerably larger heat exchanger is therefore needed to be able to keep velocities up and plate numbers down. This is ultimately due to the low temperature and mass flow of fluid 1. As it stands in the calculations, the LMTD is a merely 4,6 °C, compared to 10-15 °C for the other scenarios. Increasing the inlet temperature for fluid 1 from 53 to 60 °C, the required HTA is reduced to 1/3, size 4 plate gap 2mm. Thus it does not seem like a viable option to extract much more than what is needed for the combination of floor heating and snow/ice melting from this waste heat source. Calculations for a larger heat exchanger were therefore not done.

6.6 Scenario comparison

From the results presented, it is possible to utilize the waste heat sludge water from the cleaning of the oven gas, although both the temperature and mass flow is quite low. Scenario 1, 2, and 3 all gave some results for possible utilization. Scenario 3 may seem to be the more favorable scenario as a more substantial portion of the waste heat is utilized. However, some situations may need to be considered. The number of days floor heating is required is much higher than the number of days snow/ice melting is necessary. Thus, if the water in the snow/ice melting circuit is not circulated through the heat exchanger simultaneously with the floor heating circuit, the mass flow drops significantly. That will result in reduced velocities for fluid 2, and consequently reduced heat transfer and a higher risk of fouling. Keeping the water circulated in the snow/ice melting circuit at all times is obviously a waste of energy as well. Energy saved from reusing the waste heat for floor heating must then cover the extra energy used for the circulation to be feasible. If the periods where snow/ice melting is predictable, a solution might be to increase/decrease the number of plates in the heat exchanger accordingly, which is one of the advantages of the PHE.

When considering usability, scenario 1 has an actual effect, where the energy needed to heat rooms is already being spent, regardless of the source of heat energy. Snow/ice melting is more of an extra benefit rather than a necessity, in addition to not being required as many days. Thus the gain and payback time for a heat exchanger for this purpose will naturally be worse than for scenario 1. Nevertheless, if there is a similar area that needs roughly the same temperature as floor heating, and has the same uptime, the waste heat source is capable of covering a demand similar to that of scenario 3. The waste heat source is also capable of supplying the district heating at a temperature of 40 °at current demand. Installation of new carbon filters could increase the heat energy gained, and more applications could be covered.

For the time being, scenario 1 is recommended to implement, with a heat exchanger comparable to one presented in Table 6.6.

Table 6.6: Recommended heat exchangers for scenario 1

Heat exchanger	**Plate gap**	**Number of plates**
Size 2	2 mm	39
Size 3	2 - 2,5 mm	26

7 | Conclusion

In this study utilization of waste heat at ENK was considered. With the intention of reducing the amount of energy going to waste, and hopefully reduce the overall energy consumption. Low maintenance was seen as paramount when providing a method of heat recovery. Fluid characteristics were determined based on fluid samples with element concentration. Four different application scenarios were investigated. For each scenario, four different sized heat exchangers were tested, and the effect of varying plate gaps was explored.

Filtered sludge water from the cleaning of the oven gas was considered the most promising waste heat source. The determination of the filtered sludge water characteristics was based on the element concentration of a few samples. The presence of elements in the water was low, 0,6%, and did not seem to have a significant impact on the characteristics of the water, and was quite similar to the nearby river water. Due to the low temperature of 53 °C, only two applications stood out as possible to supply, floor heating, and snow/ice melting, as it was desired to avoid the use of heat pumps. A PHE was chosen for the task, as its design promotes low fouling tendencies, easy maintenance and can operate at low temperature differences.

The results from the four scenarios showed that pressure drop in the channels reduced drastically when the plate gaps increased, even for 0,5 mm increments at a time. The pressure drop was reduced at a much higher rate than both velocities and the OHTC. Therefore the plate gap could be kept at 2-2,5 mm in most cases while keeping velocities above 0,3 m/s and the OHTC satisfactory. This plate gap was desired to reduce the risk of clogging and rapid fouling. The optimal number of plates turned then out to be in the range of 25-45.

A comparison of the scenarios showed that the heat exchanger size grew rapidly as the demand increased. This was due to the low temperature available. Scenario 4 was discarded as the required heat exchanger size was larger than deemed necessary for its purpose. Scenario 2 and 3 were both possible to implement with reasonable heat exchanger sizes, however with little gain due to snow/ice melting is rarely needed. Among the scenarios, scenario 1, floor heating, was the scenario considered most feasible to implement, because the indoor heating is needed and frequently used regardless of it being supplied by waste heat or not. Heat exchangers similar to size 2 and 3, with plate gaps of 2-2,5 mm were recommended.

8 Further Research

In this study, only one of the waste heat sources were closely investigated. There are more waste heat sources at ENK then those presented here, with different potential. Some have a large mass flow but low temperature, while others have a higher temperature and little pollution, but low mass flow. As such, there can be done several small and large installations at ENK to reduce the amount of waste heat. A further assessment would have to be made for each of them to determine if its economically feasible, and to what degree it is useable.

It is entirely possible to extract heat energy from several areas at ENK. However, finding an application for the waste can be problematic, as most of the waste heat is low temperature. Creating a demand may be a solution, similar to the aquaculture that has been developed. Other solutions may be increasing the temperature with the help of heat pumps, to widen the application possibilities. A cascade solution where heat energy is combined from several places might be a possibility.

The utilization proposed in this study is based on heating during the winter. During summer, these systems are not necessary, but the waste heat is still present. Thus finding a use for this and similar energy sources in the summer can significantly reduce the heat going to waste and should be investigated. Such a solution might very well be applicable to many other companies and factories than just ENK.

The company desired a low maintenance solution. Nevertheless, maintenance must be done. What type of cleaning that is best suited for the PHE should be determined. Closer investigation to operation time and cost should be investigated as well.

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Appendices A | Scenarios

A.1 Scenario 2

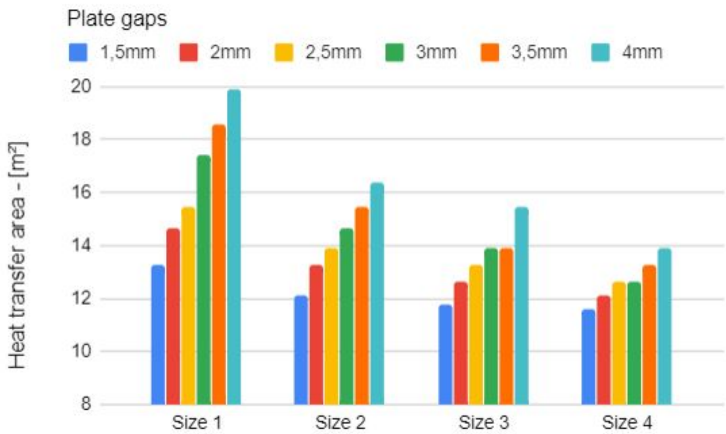

Figure A.1: Required surface area scenario 2

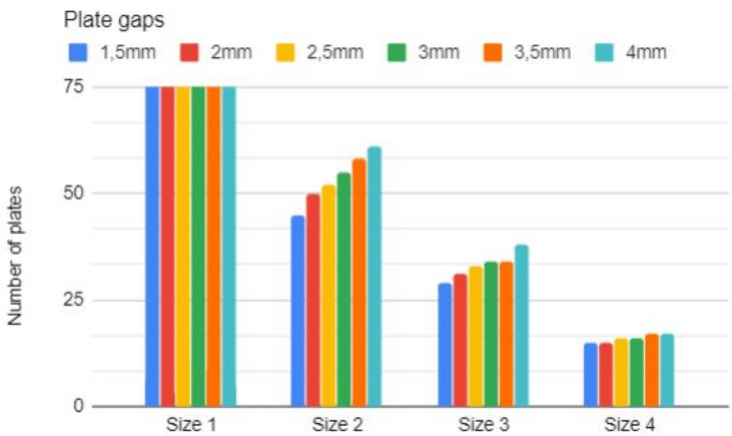

Figure A.2: Number of plates for scenario 2

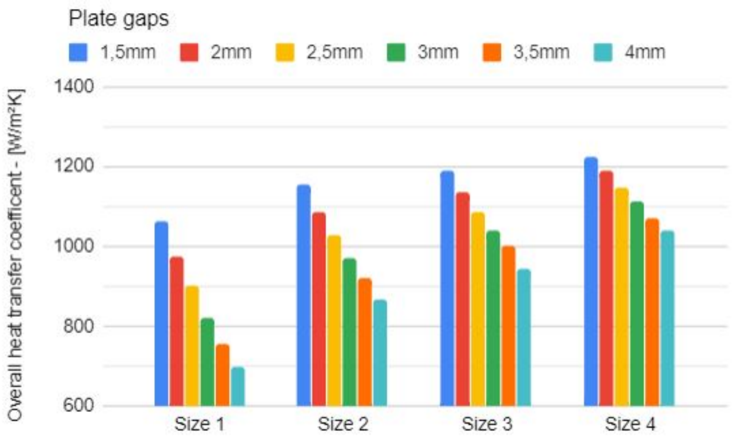

Figure A.3: Overall heat transfer coefficient for scenario 2

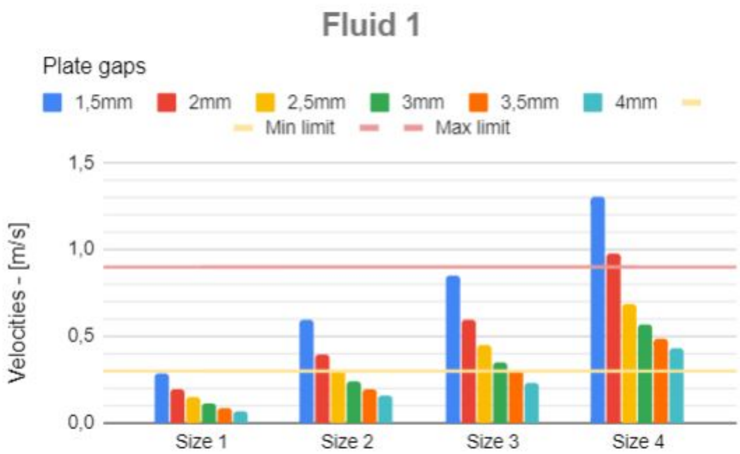

Figure A.4: Velocities fluid 1 scenario 2

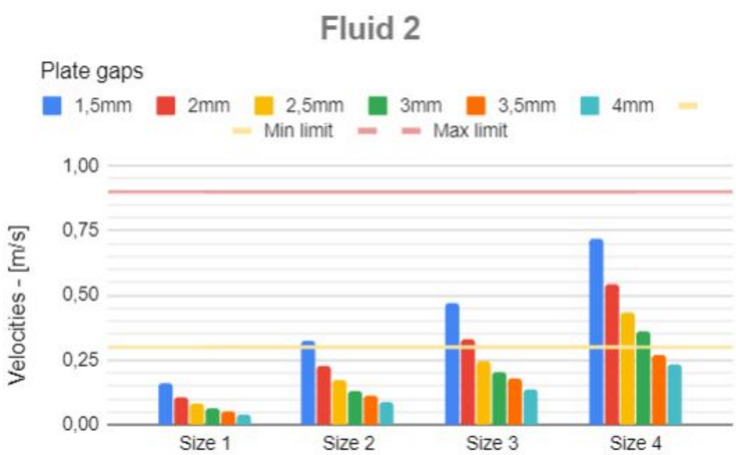

Figure A.5: Velocities fluid 2 scenario 2

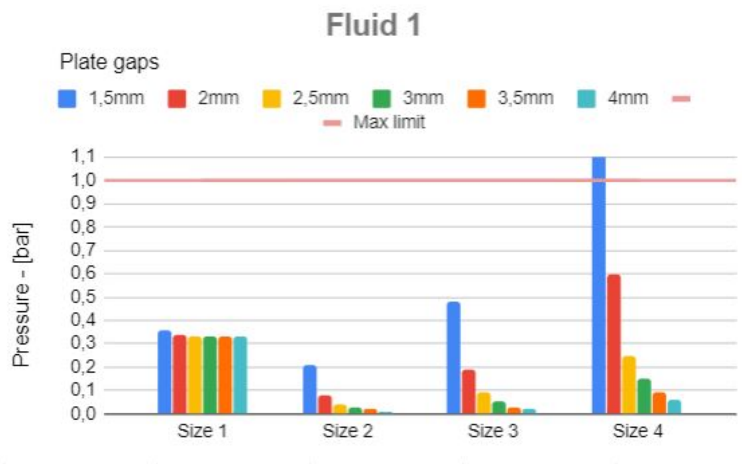

Figure A.6: Total pressure drop fluid 1 scenario 2

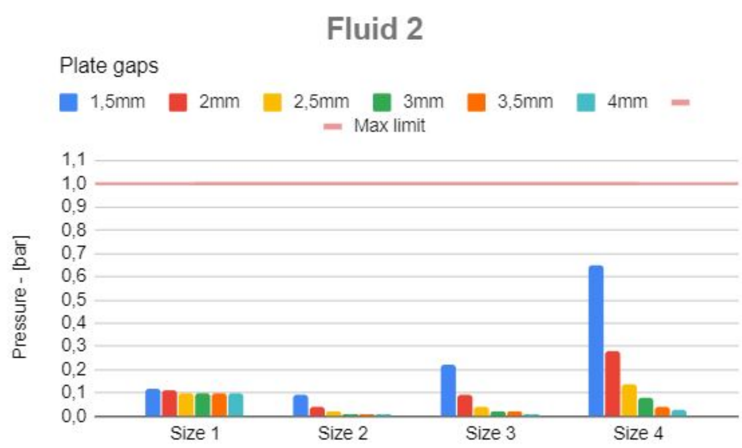

Figure A.7: Total Pressure drop fluid 2 scenario 2

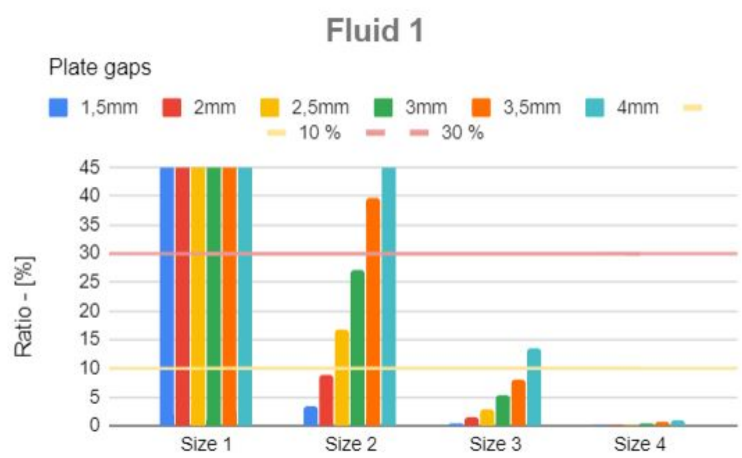

Figure A.8: Port pressure drop relative to total pressure drop for fluid 1 scenario 2

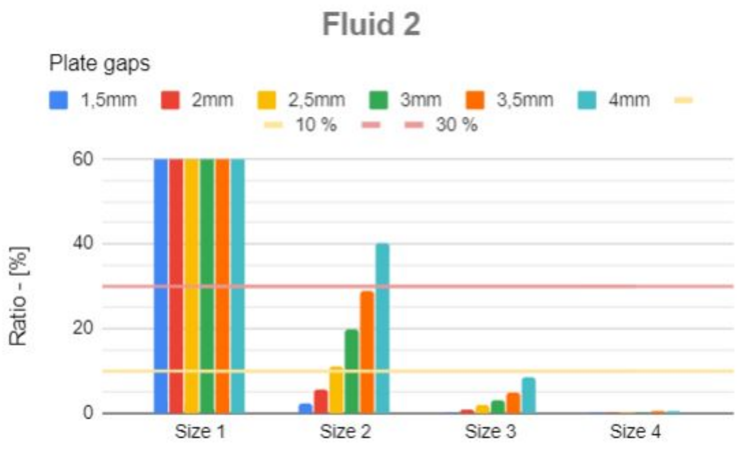

Figure A.9: Port pressure drop relative to total pressure drop for fluid 2 scenario 2

A.2 Scenario 3

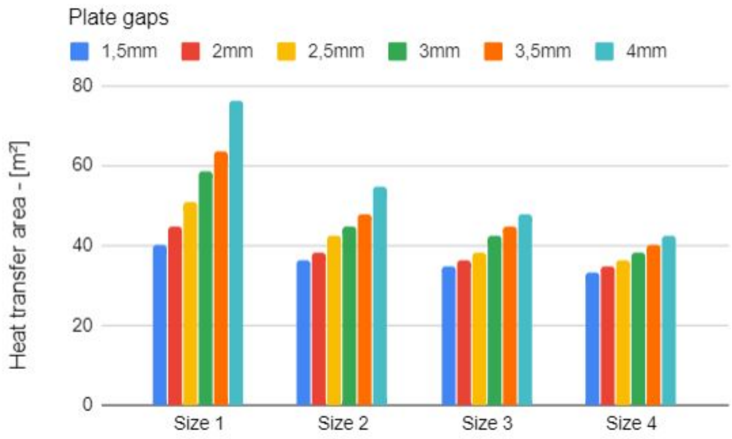

Figure A.10: Required surface area scenario 3

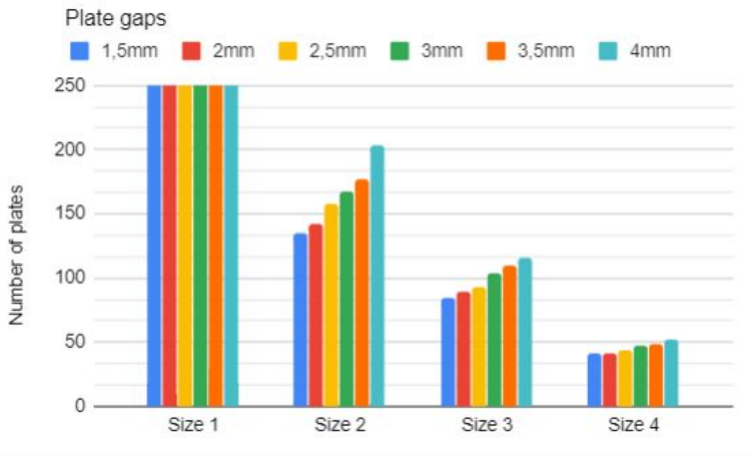

Figure A.11: Number of plates for scenario 3

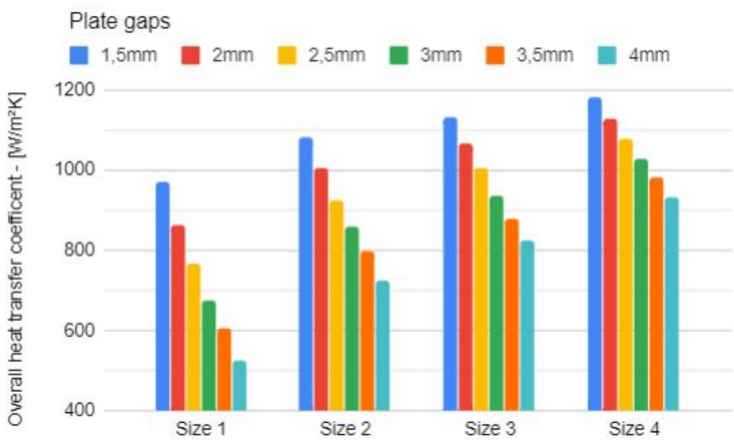

Figure A.12: Overall heat transfer coefficient for scenario 3

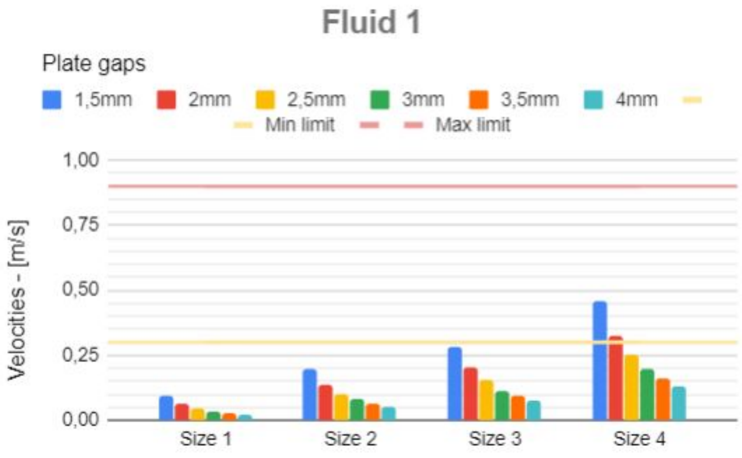

Figure A.13: Velocities fluid 1 scenario 3

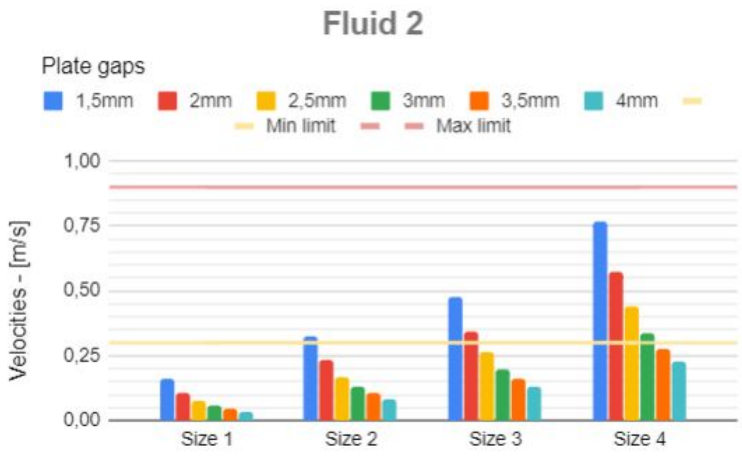

Figure A.14: Velocities fluid 2 scenario 3

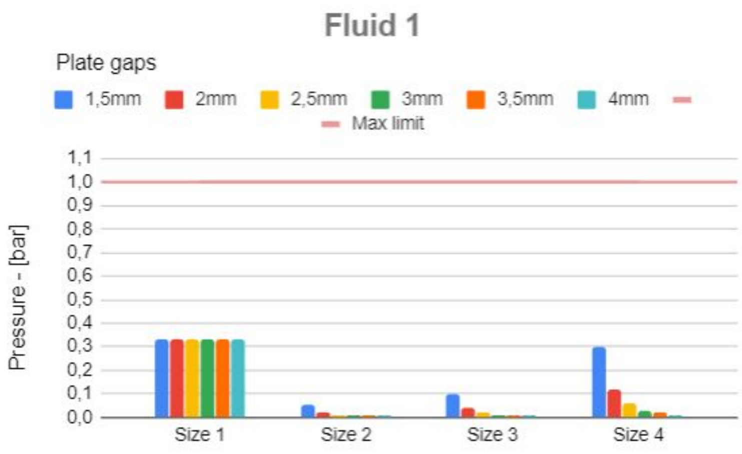

Figure A.15: Total pressure drop fluid 1 scenario 3

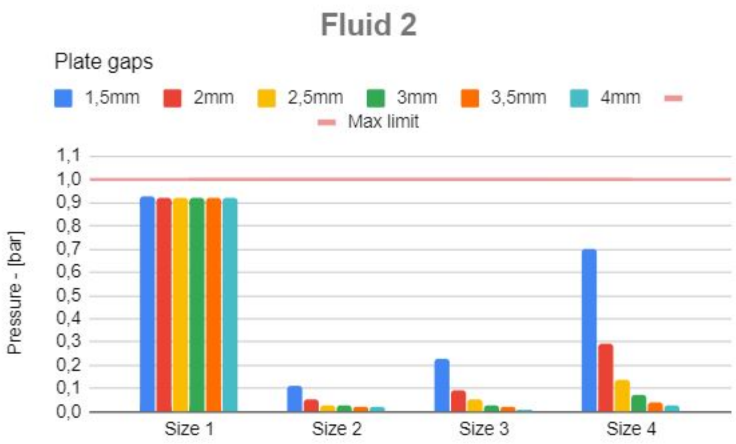

Figure A.16: Total Pressure drop fluid 2 scenario 3

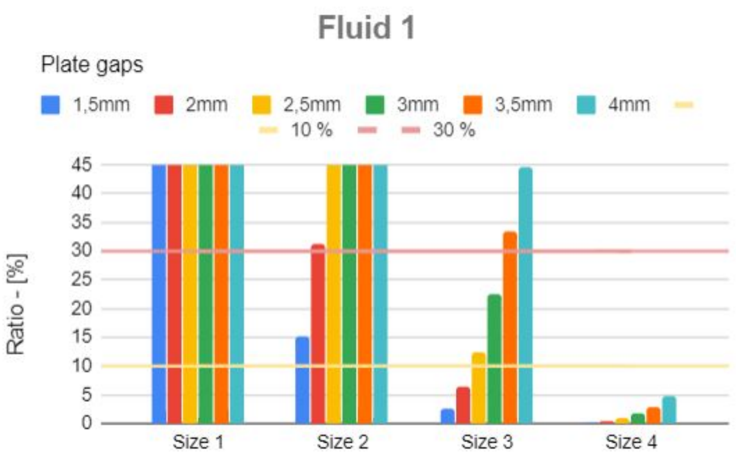

Figure A.17: Port pressure drop relative to total pressure drop for fluid 1 scenario 3

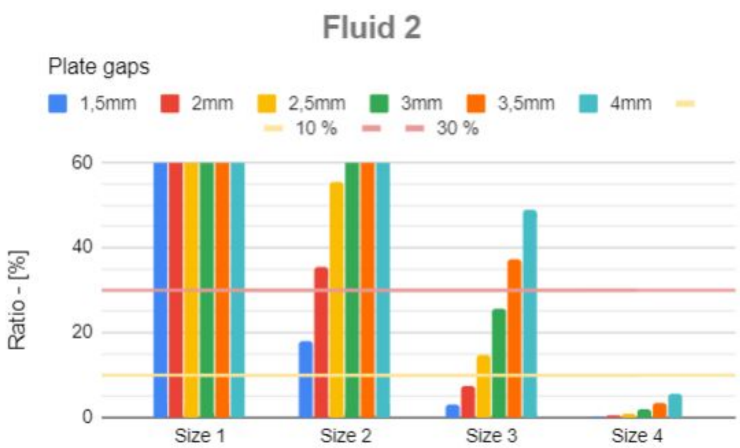

Figure A.18: Port pressure drop relative to total pressure drop for fluid 2 scenario 3

A.3 Scenario 4

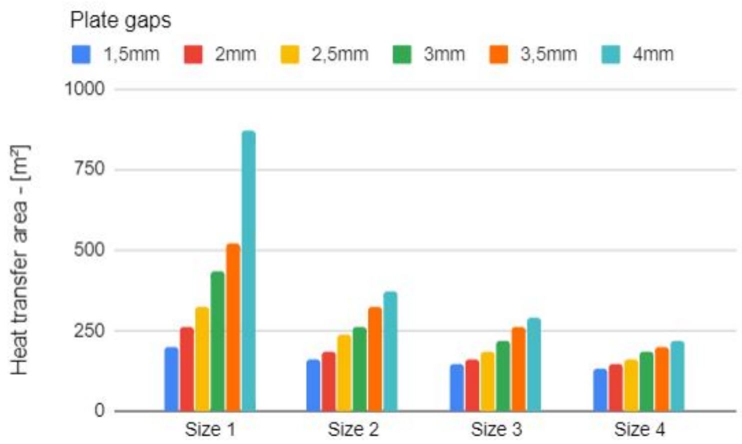

Figure A.19: Required surface area scenario 4

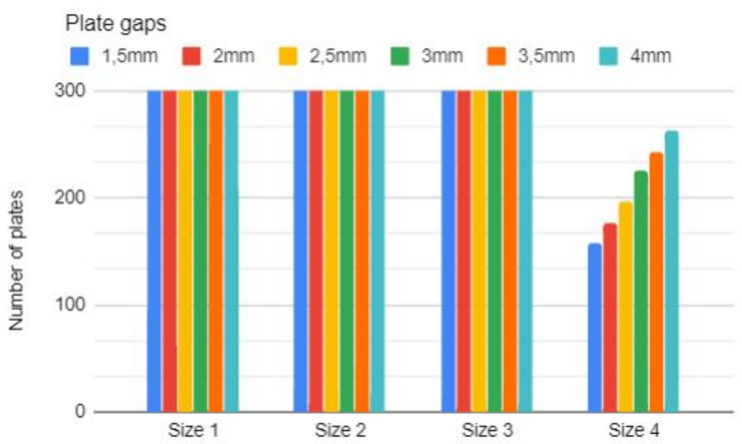

Figure A.20: Number of plates for scenario 4

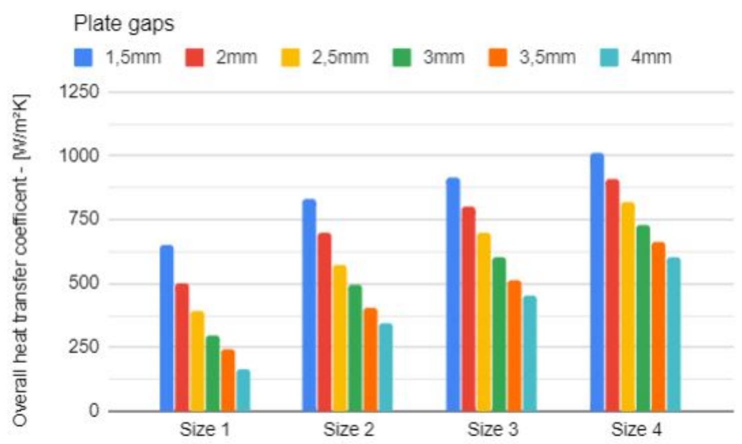

Figure A.21: Overall heat transfer coefficient for scenario 4

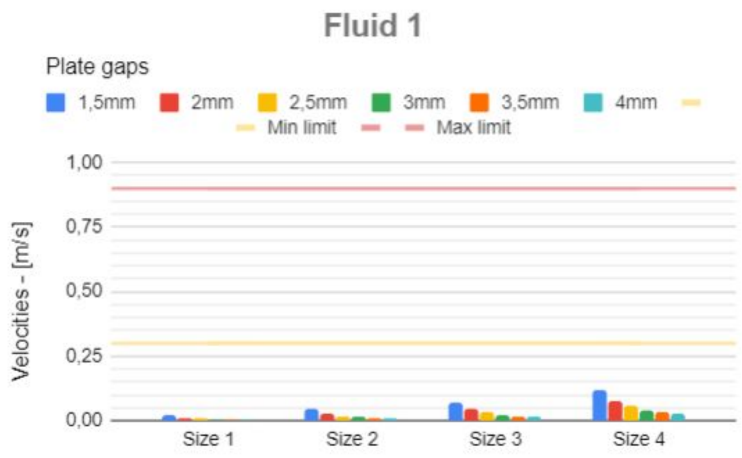

Figure A.22: Velocities fluid 1 scenario 4

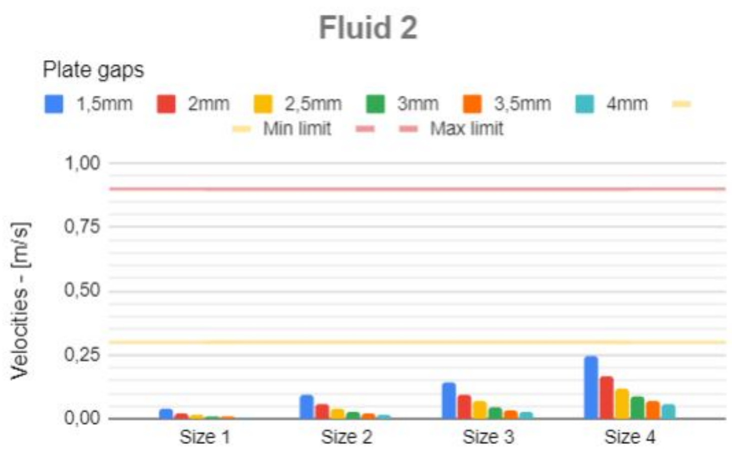

Figure A.23: Velocities fluid 2 scenario 4

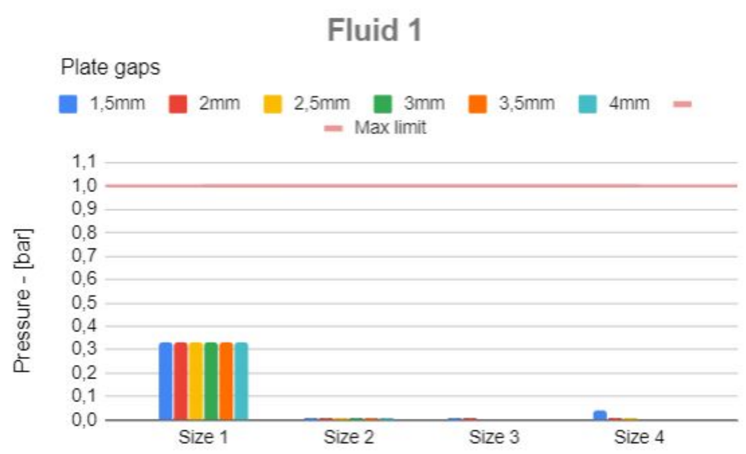

Figure A.24: Total pressure drop fluid 1 scenario 4

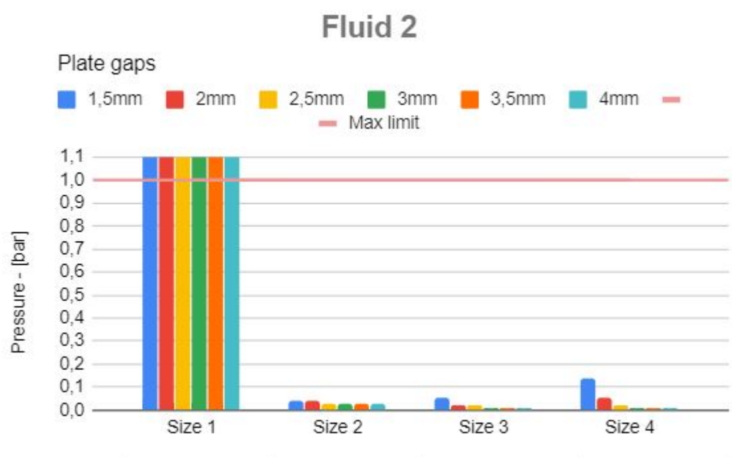

Figure A.25: Total Pressure drop fluid 2 scenario 4

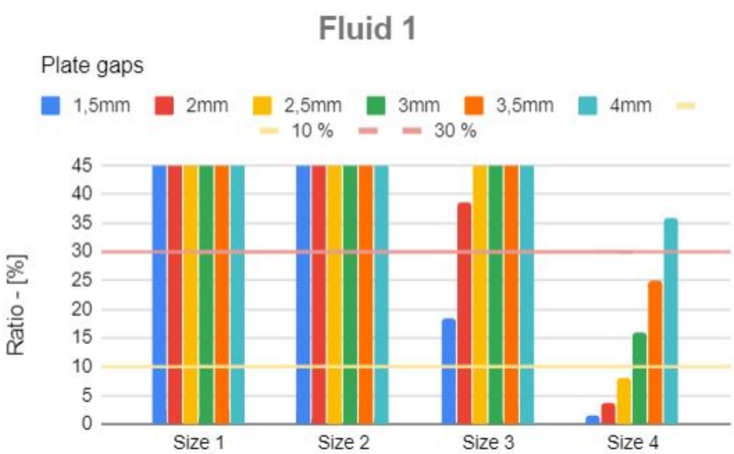

Figure A.26: Port pressure drop relative to total pressure drop for fluid 1 scenario 4

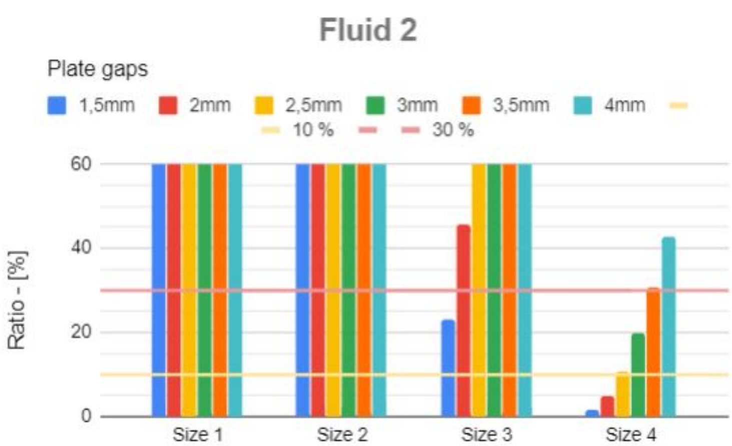

Figure A.27: Port pressure drop relative to total pressure drop for fluid 2 scenario 4