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Department of Engineering and Natural Sciences

Department of Green Engineering

Infraleuna GmbH

Bachelor Thesis

For the award of the degree Bachelor of Engineering (B.Eng.)

"Waste heat recovery from potential heat sinks in the combined cycle power plant in Infraleuna GmbH"

Submitted to:

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Abstract

Waste heat recovery is the most prominent development in businesses aiming to lower their carbon footprint. Instead of allowing the excess heat generated as a by-product to be lost into the atmosphere, businesses in any type of industry are starting the practice of recycling this heat and putting it to better use. Not only does it reduce their overall energy consumption, but it also curtails their contribution to global warming.

Infraleuna GmbH, an exemplar in this endeavour, actively employs WHR technologies. However, heat always manages to escape in some form. In order to do its part in combatting global warming, Infraleuna GmbH is always trying to optimize its efficiency. This research delves into identifying heat loss sources within Infraleuna's operations and devising a system to minimize them.

The study proposes diverse models tailored to specific heat sources, accounting for available thermal potential and varying parameters. To facilitate calculations of enthalpies and construct pressure-enthalpy (p-H) diagrams, the Solkane 8.0.0 application will be employed.

This investigation contributes valuable insights to the field of sustainable energy practices, emphasizing the critical role of WHR in combatting global warming. By bridging theory and practical implementation, we pave the way for more efficient industrial processes and a greener future.

Abbreviations

CCGTP	Combined cycle gas turbine plant
COP	Coefficient of Performance
CO_2	Carbon dioxide
CHP	Combined heat and power
GWP	Global warming potential
HFC	Hydrofluorocarbon
HRSG	Heat recovery steam generator
IHX	Internal heat Exchanger
ODP	Ozone depletion potential
ORC	Organic Rankine Cycle
TXV	ThermoExpansion valve
WHR	Waste heat recovery

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1 Introduction

In this day and age, sustainability has become of paramount importance. With pressing issues piling up due to global warming, mankind has shifted its focus into finding alternative energy resources in order to relinquish the dependency on fossil fuels. Renewable energy has been a forefront contender in recent years, but with the escalating global energy demand, innovative and sustainable solutions are being sought out.

The utilization of waste heat from industrial processes has emerged as a pivotal solution that aligns with the principle of a circular economy. This approach addresses both environmental and economic concerns by minimizing waste and optimizing energy usage.

Infraleuna GmbH is one of the pioneer industries living up to the commitment of responsibly using energy sources in order to maintain sustainable practices. Notably, the company specializes in leveraging waste heat from combustion to generate electricity. This thesis concentrates on a specific facet of Infraleuna's operations—the low-pressure steam system employed for heating water in district heating. Although this system is already part of a waste heat recovery system, there are still potential heat sinks causing energy to dissipate into the environment. The primary objective of this study is to comprehensively assess the potentials and challenges inherent in the efficient utilization of waste heat within this particular heating process.

1.1 Infraleuna GmbH

Infraleuna GmbH and its subsidiaries serve as proprietors and operators of the infrastructural facilities within the Leuna Chemical Complex. Infraleuna is presently dedicated to assisting companies in establishing and developing a robust supporting framework for their production lines. The company delivers cost-efficient services tailored to the specific needs of individual businesses, alleviating them of peripheral activities to the extent desired. Infraleuna provides an appealing array of services, including the provision of utilities such as power and water, comprehensive wastewater treatment through a centralized sewage treatment plant, and analytical services in state-of-the-art laboratories. Through these offerings, Infraleuna empowers its clients to access cost-optimized shared services in a highly efficient manner. (Infraleuna GmbH, 2021)

1.2 Current Situation

Infraleuna GmbH’s heating system relies heavily on the low-pressure steam produced during electricity generation in the boiler house. At the heating water station, this steam is converted into hot water through a condensate heat exchanger and is then transported to the Leuna district heating network for heating residential properties. In addition, Infraleuna also sells this low-pressure steam.

As of today, UPM Biochemicals procures low-pressure steam at 2.2 barg and saturated steam at a maximum temperature of 200 °C from Infraleuna. However, UPM Biochemicals requires a steam volume surpassing what Infraleuna can provide. To meet this demand, Infraleuna resorts to using steam from a higher-pressure stage at 14.5 barg and then reducing it to 2.2 barg—an intricate and costly procedure. Consequently, Infraleuna GmbH is exploring alternative options to replace processes reliant on low-pressure steam.

1.3 Aim and Scope

This thesis investigates the integration of waste heat recovery mechanisms within a Combined Cycle Gas Turbine Plant with the aim of providing an innovative and sustainable approach to district heating. Through the analysis of temperature data, heat transfer mechanisms, and efficiency parameters, this study aims to explore potential strategies for utilizing generated waste heat, including process optimization, heat recovery, and innovative methodologies. The investigation will involve identifying and comparing various waste heat sources to select the most efficient sink. Performance tests will assess several parameters to determine the effectiveness of the chosen solutions. The outcome of this research will contribute not only to the expansion of knowledge in waste heat utilization but also showcase Infraleuna GmbH's commitment to sustainable practices and responsible energy resource management. The insights gained may not only benefit Infraleuna GmbH but also provide a reference for similar industrial facilities seeking to enhance energy efficiency and make positive environmental contributions. The overarching goal of this thesis is to devise a method for operating district heating and other processes using waste heat, thereby conserving low-pressure steam for sale to UPM Biochemicals. The recovered heat will be directed toward heating water for district purposes.

1.4 Structure of Thesis

The following outline will be used as the structure for this thesis and all relevant information can be found in corresponding chapters.

Chapter	Title	Content
1	Introduction	The context of this research will be discussed, followed by the aim and scope of this study.
2	Literature Review	Relevant concepts such as waste heat technologies, district heating and heat pumps will be analysed.
3	Methodology	Different models for the heat pump system will be proposed by varying different parameters
4	Results and Discussion	The calculations made in chapter 3 will be analysed and compared so that the most efficient configuration can be chosen. The limitations to this model will be discussed followed by the environmental impact.
5	Conclusion	A summary of the devised models is discussed.

Table 1: Structure of the thesis

2 Literature Review

The knowledge of waste heat recovery boiler and the processes involved, district heating, heat pumps will be discussed in depth to aid in a better comprehension of which steps need to be taken in developing a better sustainable system in the Combined Cycle Gas Turbine Plant (CCGTP).

2.1 Waste Heat

Conforming to the laws of thermodynamics, diverse processes and activities generate a residual heat that is wasted and dumped into the environment. During industrial processes such as power generation or daily activities like watching the television, heat is used and converted in some way. Some heat is also produced as a by-product, that is lost to the surroundings and is not captured or harnessed for any productive purpose. This excess heat is categorized as **waste heat**.

For years, this type of heat has been contributing to global warming because of its wastage. But with the recent movement towards sustainability, more emphasis is being made on recapturing and reusing. Hence scientists have started tapping into recovery of waste as a potential energy source.

The importance of waste heat lies in the possibility of using this energy source more efficiently in order to achieve various benefits:

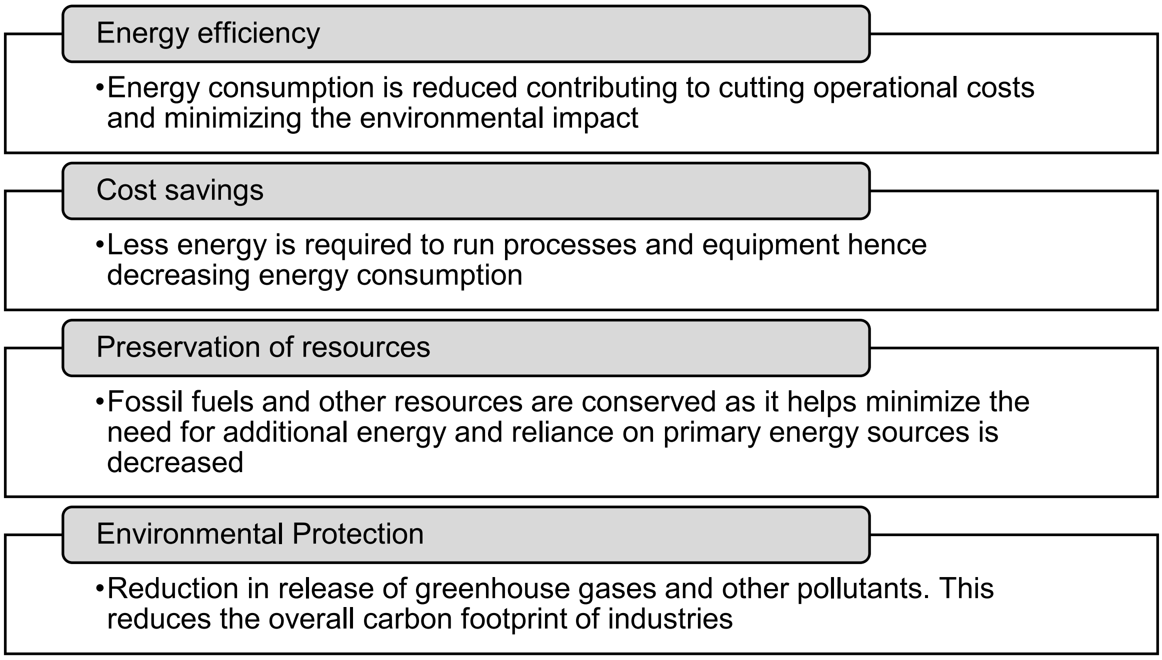

Figure 1: Benefits of waste heat

Waste heat can be released from processes involving combustion, electrical resistance or friction. The release can be either in an **emissive** or a **non-emissive** way.

In the emissive type, heat is lost mainly as a by-product of processes. For example, steam is generated when using water to cool machinery. The non emissive type is the process where heat is lost by conduction, convection and radiation. The CCGTP deals mostly with the non-emissive type of waste for heat tends to escape from the surface of the heat sinks such as the steam drums.

2.2 Waste Heat recovery technologies

Waste heat recovery (WHR) systems capture and convert the excess heat, generated during various industrial processes, into useful energy. The selection of the appropriate waste heat recovery technology depends on various factors, including the temperature and amount of waste heat available, the requirements of the specific application and the cost-effectiveness of the investment.

High temperature WHR consists of recovering waste heat at temperatures greater than 400 °C, the medium temperature range is 100–400 °C and the low temperature range is for temperatures less than 100 °C. Usually most of the waste heat in the high temperature range comes from direct combustion processes, in the medium range from the exhaust of combustion units and in the low temperature range from parts, products and the equipment of process units. (S. Bruckner, 2015)

The commonly used waste heat recovery technologies are discussed below:

Combined heat and power (CHP)

CHP systems are cogeneration systems that use a single fuel source to generate electricity while simultaneously producing thermal energy in the form of heat or steam as a by-product. The heat that is generated as a by-product of power generation is used to supply thermal energy to public and private consumers. Compared with other power generation facilities, CHP plants require less fuel and are therefore more efficient. (BWMK, 2024)

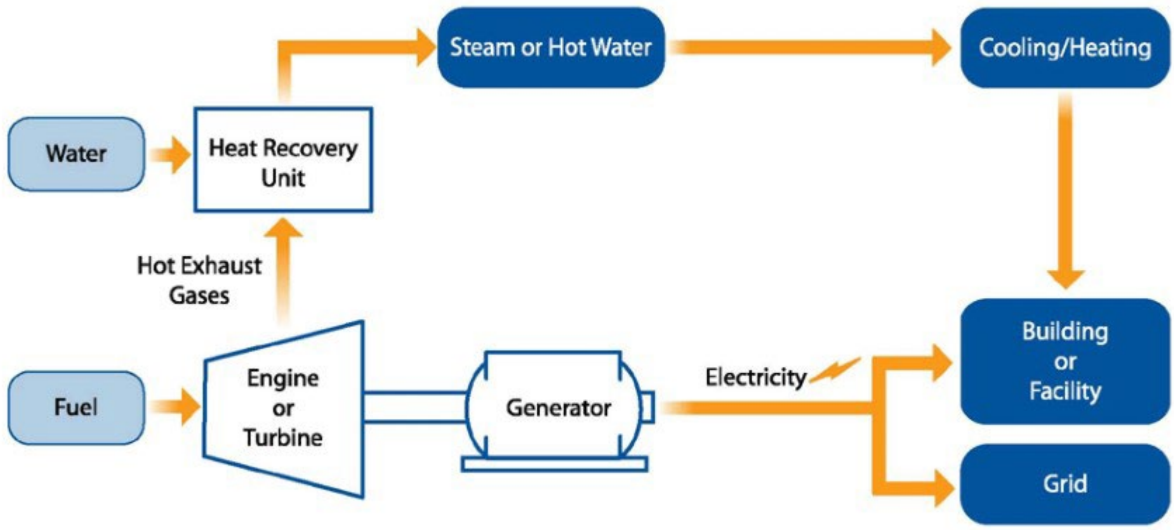

Figure 2: Combustion Turbine with Heat Recovery unit (EPA, 2024)

Heat Recovery Steam Generator (HRSG)

An HRSG is an integral component of the CHP systems, and its main application is in combined cycle power plants. The working principle revolves around the capture of waste heat inherent in exhaust gases, that would have otherwise been expelled through the exhaust pipes to the atmosphere. The waste heat is used to heat up water and generate steam that is used in a steam turbine to produce additional electricity. This avoids the need for separate boilers.

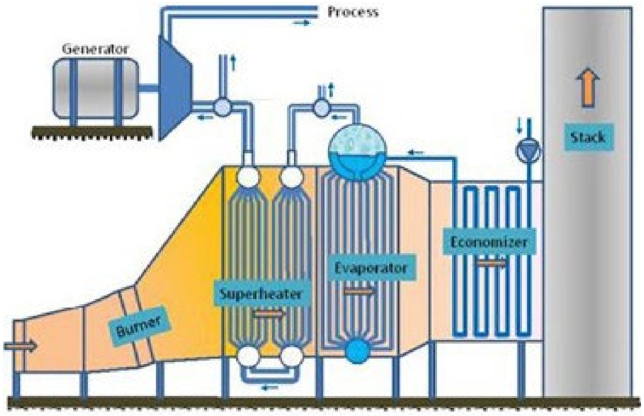

Figure 3: Heat Recovery Steam Generator (HRSG) (Haug, 2016)

Organic Rankine Cycle (ORC)

ORC systems aim to improve efficiency at low waste heat temperature levels. Instead of using water as a working medium, an organic fluid is utilized. It is vaporized in a turbine causing expansion to occur, this converts the thermal energy into mechanical work that drives the turbine to produce electricity.

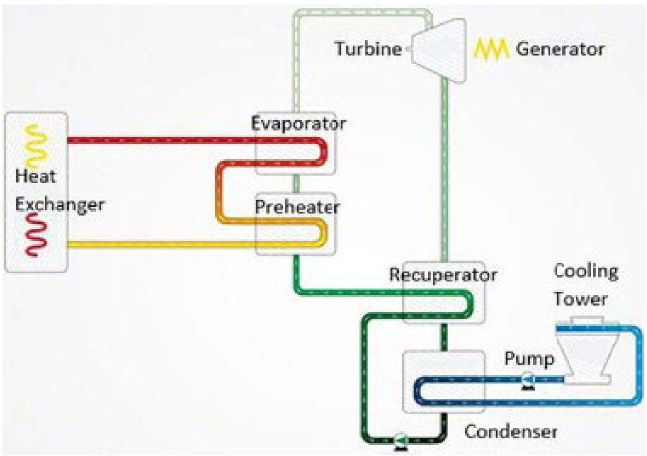

Figure 4: Schematic of a typical Organic Rankine Cycle (coepro, 2017)

In summary, CHP systems incorporate WHR principles by capturing waste heat via the HRSG. (Mo. D, 2022)

2.3 Components used for heat recovery

2.3.1 Heat Exchangers

The capture of waste heat in HRSG involves the use of a heat exchanger typically in the form of tubes or pipes. There are different types of heat exchangers, namely: plate heat exchangers, shell and tube heat exchangers and air-to-water heat exchangers. They allow the transfer of heat from one medium to another without them getting into contact with each other. The mediums can be same such as fluid-fluid or different like gas to water. In the HRSG system, the heat exchanger is strategically positioned in the exhaust system where it intercepts and transfers thermal heat from the hot exhaust gases to water, hence generating steam.

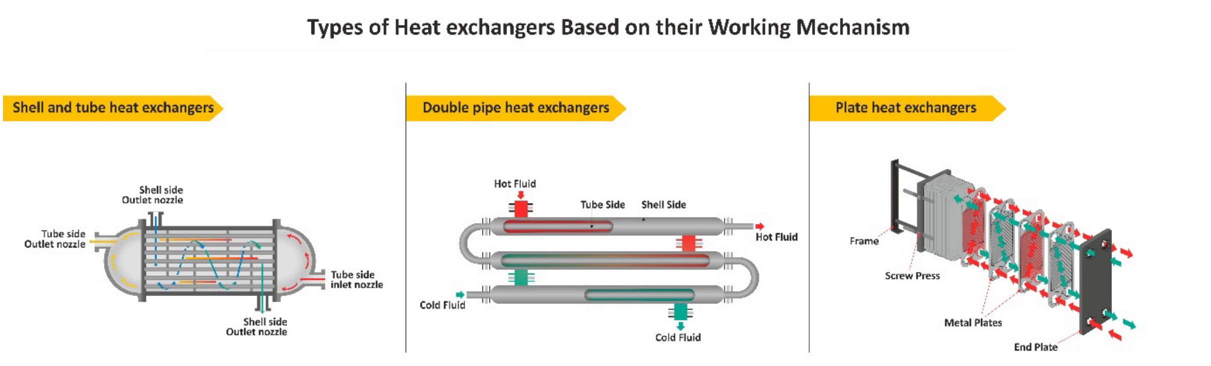

Figure 5: Types of heat exchangers (Stratview Research, 2023)

2.3.2 Steam Drums

The function of the boiler steam drum is to collect the generated steam and to provide a space in which steam can separate from the boiler water. Steam is generated in the water pipes of a boiler and at the saturation point, steam bubbles begin to form, and the two-phase mixture returns to the steam drum to separate steam from the liquid water. Due to the difference in density, steam moves upwards through the tubes that are attached to the connected to the upper part of the steam drum. Steam drums ensure a steady supply of steam to downstream processes.

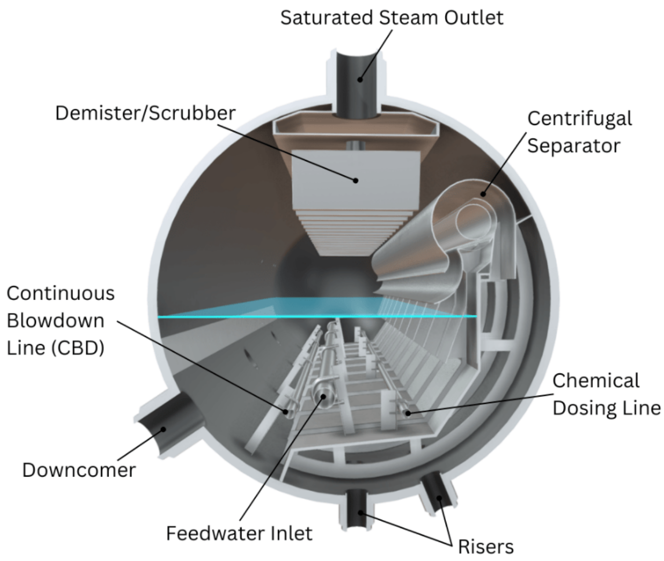

Figure 6: Steam drum (Savree, 2024)

2.3.3 Economizers

An economizer is a water-to-air heat exchanger that improves the overall efficiency of a HRSG. This component utilizes waste heat from the flue gas leaving the gas turbine to preheat the feedwater before it enters the evaporator.

2.3.4 Superheaters

A superheater optimizes the steam temperature and pressure, by further increasing the temperature of the steam beyond its saturation point. The produced superheated steam is crucial for an efficient steam turbine operation for it contains more energy per unit mass and prevents condensation during expansion. This dry steam also minimizes the risk of the blade eroding over time.

2.3.5 Internal heat exchanger (IHX)

The IHX is a simple “tube within a tube” device which transfers heat between the low pressure and the high pressure flow circuits. It has no moving parts uses counter current movement. It simply works by the movement of the hot liquid refrigerant from the high side of the system flowing through the outer tube at the same time the vaporized refrigerant from the low side of the system flows through the inner tube. This process causes a heat transfer in which the high-temperature liquid refrigerant from the high side is subcooled prior to entering the H-Block (also called a ThermoExpansion Valve or TXV) and the cold vapor refrigerant from the low side is superheated before heading to the compressor. (FJC, 2021)

2.4 Heat pumps

As compared to traditional heating systems, heat pumps are eco-friendly and energy efficient devices that are not dependent on fuels, they don't generate heat through processes like combustion. They absorb and transfer heat from a lower temperature source to a higher temperature reservoir through a process called the refrigeration cycle. They work just like the refrigerator except in a reversible way. The biggest selling point of a heat pump is its ability to lower the overall operation cost.

Heat pumps can be sorted into two main categories based on the prime mover in use, they are compression and absorption heat pump. The former uses a compressor operated by mechanical power to increase the pressure and temperature of the refrigerant. It is suitable for small scale usage such as domestic heating or cooling systems. In absorption heat pumps an absorber and generator is installed instead of a compressor, also in addition to the refrigerant, absorbent fluid is also added which helps to increase the pressure and temperature of the refrigerant by changing the concentration of the solution. Heat pumps can be classified according to their heat sources it is illustrated in the table below. (AMPERAYANI, 2021)

There are different types of heat pumps, namely:

- Air source
- Water source
- Ground source - regardless of weather conditions, absorbs heat from the ground.

Depending on the type of heat pump being used, the possible sources the evaporator can absorb heat from can be water, the outdoor air, or the ground.

2.4.1 Air-water heat pump

As the name suggests, air-to water heat pumps use air as the heat source and a water-based system as the heated medium. Owing to the fact that outdoor air is available in unlimited quantities, it can be tapped at low cost, which makes this type of heat pump have the largest market value. Helping to reduce heating bills by up to 60 % and cutting carbon dioxide (CO_2) emissions by 50 %, air to water heat pumps are leading the way in the implementation of more sustainable heating solutions. (Pipelife Eco, 2022)

Additionally, air-to-water heat pumps can also work in reverse during warm weather to provide cooling by extracting heat from indoors and releasing it outside.

2.4.2 Water-water heat pump

Such a device can extract heat from a flowing source of low temperature water and deliver that heat to another, higher temperature water stream. The low temperature source may be ground water, cooling water from an industrial process or even process water that needs to be chilled before use. Almost any situation where heat is available in the form of low temperature water, in combination with a simultaneous load to which heat can be delivered in the form of higher temperature water, is a possible application for a water-to-water heat pump. (P.E., 2000)

2.4.3 Refrigeration cycle

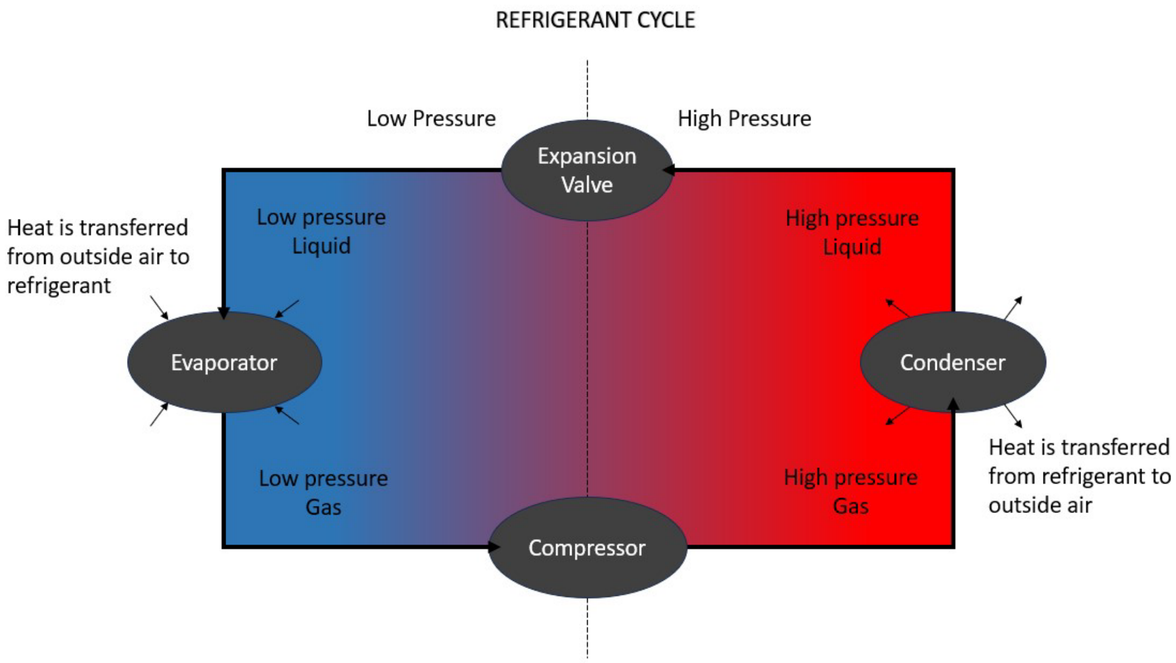

Figure 7: Refrigerant Cycle

As described by the laws of thermodynamics, there is no such thing as cold. Scientifically, a cold object would mean that its heat has been removed. Hence, a refrigeration cycle can be described as a way to redirect heat away from the object that needs to be cooled. The cycle works by varying the pressure of the refrigerant being used through either compression or expansion.

The refrigeration cycle comprises of the following steps:

1. Superheated refrigerant vapor is sucked into a compressor where the pressure is increased. In doing so the temperature and energy of the refrigerant also increases while the specific volume decreases.

2. In the condenser, a large amount of heat energy is released causing a phase change from saturated vapor to saturated liquid to occur.
3. The refrigerant that enters the expansion valve is a high pressure but low temperature liquid. A drastic drop in pressure in the valve causes the refrigerant to quickly boil. This phenomenon is known as flashing.
4. The liquid-vapor mixture moves to the evaporator where heat is transferred to the refrigerant to increase its energy and remove traces of liquid. At this point the air is cooled due to the heat absorption by the evaporator.
5. The saturated refrigerant vapor then flows through the suction pipe and the whole process starts anew.

All heat pumps follow basically the same process. This whole process can be plotted in the p-h as shown in Figure 3.

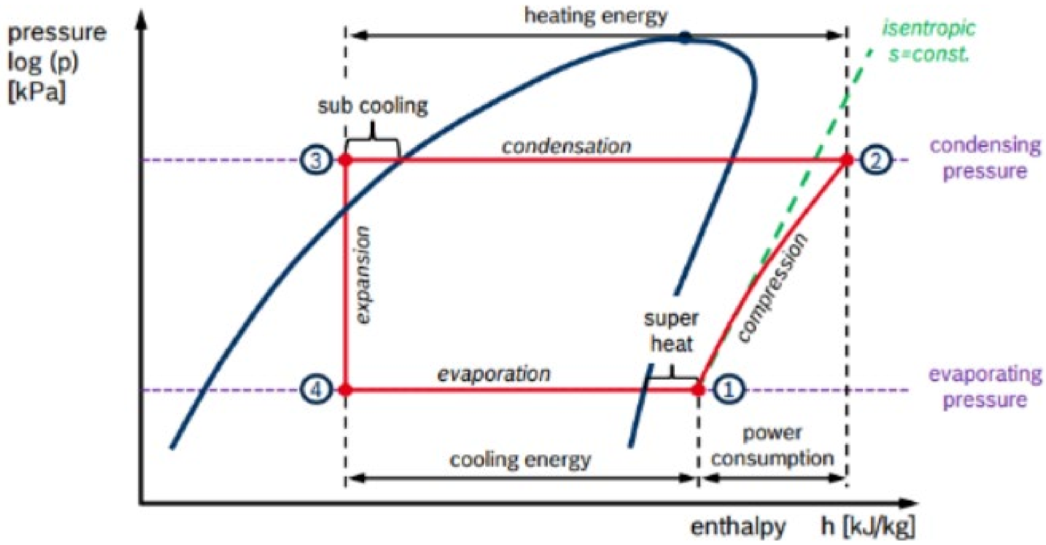

Figure 8: p-H diagram of a basic heat pump circuit (AMPERAYANI, 2021)

2.5 Possible heat loss sources in Boilers

In a waste heat recovery system, heat loss can occur at various stages within the system. Some common sources of heat loss include:

Source	How?
Flue Gas Loss	Incomplete heat exchange in the heat exchanger due to fouling or deposits on the heat exchanger.
Incomplete Heat Recovery	Inadequate design or sizing of the heat recovery equipment
Transmission Losses	Improper insulation or poor thermal conductivity of the materials. System leakages such as in valves or joints
Temperature Differences	Due to an increase in temperature gradients, larger temperature differences between the waste heat source and the heat recovery medium can result in higher losses.

Table 2: Sources of heat loss

Energy losses in a boiler can be studied in 3 main categories:

1. Stack gas loss
    a) Dry stack gas loss: Discharge of heat from the stack through waste gases.
    b) Combustion loss: Loss due to unburned fuel and incomplete combustion.
    c) Enthalpy loss: Convection of heat out of the chimney by hot water vapor, including both latent and sensible heat.
2. Heat loss from the surface and
3. Blow off loss

One of the works that can be done to achieve a high boiler efficiency is to reduce heat loss from the surface. For this, the outer surface temperatures should be at the lowest possible values.

According to EN 12953, the outer wall heat losses of a boiler should be limited to 1 % of the total boiler capacity. These losses are depending on the boiler's outer wall temperatures. Heat losses on the outer surface of the boiler occur in two different ways in the form of radiation and convection. The heat loss caused by convection is directly proportional to the wall temperatures, and the heat loss by radiation is directly proportional to the fourth force of these temperatures. Heat loss by convection from the surfaces is shown in Equation (2.1), where A is the total surface area, α is the convective heat transfer coefficient, T_s is the surface temperature, and T_a is the ambient temperature. (Kocabaş, 2022)

$$\mathbf{Q_{convection} = \alpha * A * (T_s - T_a)} \quad \textbf{(2.1)}$$

To calculate α, the following equation is used:

$$\boldsymbol{\alpha} = \frac{\mathbf{Nu*\lambda}}{\mathbf{L}} \quad \textbf{(2.2)}$$

Where:

- **Nu** : Nusselt number
- **L** : Characteristic length [m]
- $\boldsymbol{\lambda}$: Thermal conductivity of air [W/(mK)]

The Nusselt number, Nu is the dimensionless parameter characterizing heat transfer. (Shires, 2011). In this thesis, Nu can be estimated using the equation (VDI-Wärmeatlas., 2006):

$$\mathbf{Nu = 0.664 * Re^{0.5} * Pr^{0.33} * K} \quad \textbf{(2.3)}$$

Where:

- **Re** : Reynolds number
- **Pr** : Prandtl number
- K : Correction factor (= 1 for gas)

Re is the Reynolds number that determines if a flow is laminar or turbulent and can be determined as follows:

$$\mathbf{Re} = \frac{\mathbf{\nu * l}}{\boldsymbol{\upsilon}} \quad \textbf{(2.4)}$$

Where:

- $\boldsymbol{\nu}$: Flow velocity [m/s]
- **l** : Length of wall [m]
- $\boldsymbol{\upsilon}$: Kinematic velocity [m^2/s]

The Prandtl number, Pr can be calculated as follows:

$$\mathbf{Pr} = \frac{\boldsymbol{\mu} * \mathbf{c_p}}{\boldsymbol{\lambda}} \quad \textbf{(2.5)}$$

Where:

- $\boldsymbol{\mu}$: Dynamic viscosity [kg/(ms)]
- $\mathbf{c_p}$: Specific heat capacity of air [J/(kgK)]

The characteristic length can be calculated using the formula:

$$\mathbf{L} = \mathbf{D} * \frac{\boldsymbol{\pi}}{\mathbf{2}} \qquad \textbf{(2.5)}$$

Where D is the diameter of the pipe in meters.

2.6 District Heating

Heating up buildings such as houses and offices accounts for more than half of the overall energy consumption in Germany. (BMWK Newsletter Energiewende, 2021)

That being the case, it would be beneficial to integrate sustainable alternatives in the system in order to save energy and thus mitigate climate change. Typically, each building tends to have their own heating system, however a greener option would be district heating. This type of system contributes to lower emissions, increased use of renewable energy sources and reduced energy consumption.

District heating involves a central energy source which is used to generate heat using different methods such as burning fossil fuels like natural gas or biomass, or utilizing renewable energy sources like geothermal, solar, or heat recovered from industrial processes. The produced heat is then transferred to water or steam which is then distributed through a network of underground insulated pipes to buildings such as offices, houses, or commercial buildings.

2.6.1 District Heating in Leuna

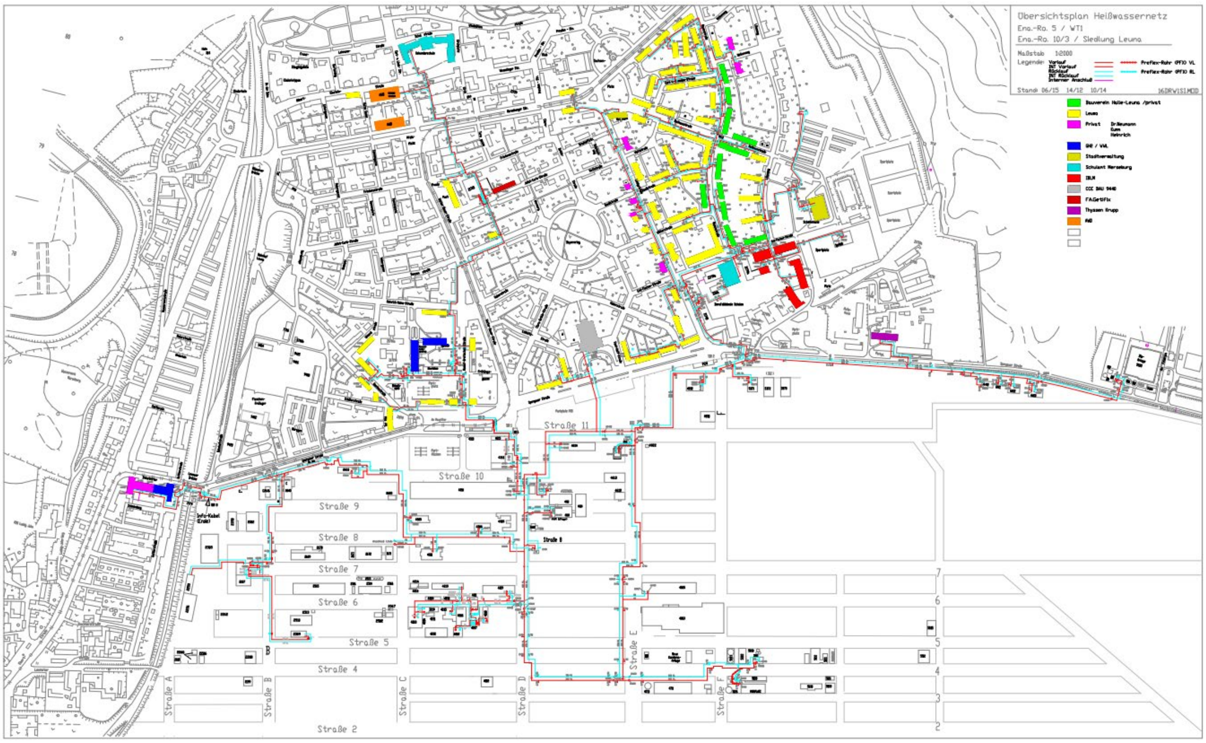

Figure 9: District heating network in Leuna.

Infraleuna GmbH is currently a fourth-generation district heating system that supplies hot water to all the companies in Leuna and provides for half of Leuna.

2.7 Gas and combined cycle power plant in Infraleuna

In the gas and combined cycle power plant, three gas turbines are operated. Ambient air is drawn in, filtered and fed into a 17-stage compressor. This compressed air is then fed into ten combustion chambers in which the fuel/air mixture, created after the addition of natural gas, is ignited. The combustion gases then flow through a 3-stage turbine in which the temperature and, above all, the resulting pressure of the exhaust gases are reduced. This converts the energy of the combustion gases into rotational energy that drives the shaft.

The shaft of the gas turbine reaches a speed of 5100 min^{-1} in gas turbine 1 and 2. Coupled to the turbine shaft, a gearbox steps down the turbine speed to the designated speed of 3000 rpm of the generator which converts the rotational kinetic energy to electricity. During this process, hot exhaust gases of a temperature of 542 °C are released. Through an exhaust system, the gases are directed to a heat exchanger located strategically within the system. As the hot gases traverse the tubes containing circulating hot water, heat absorption occurs. With the increase in temperature, a phase change occurs and the water in the tubes evaporate and turn into steam.

The steam is then directed to the hot water station where it is passed through additional heat exchangers to transfer the heat present in steam to a secondary water loop. This secondary water loop has a lower temperature than steam, it is circulated through the district heating network. The hot water from the secondary loop is distributed through pipes to various buildings and facilities within a local community. It serves as a source of space heating, domestic hot water, or other heating applications.

After releasing its heat into the water for district heating, the temperature of the steam decreases causing condensation to happen. Hence a phase change occurs, and the steam turns back to water. This water is again returned to a heat exchanger to go through the same process and repeat the cycle.

The steam generator is a forced circulation waste heat boiler consisting of the following:

Low pressure system	High pressure System	Feed water system
• Economizer	• Economizer 1	• Feed pumps
• Drum	• Drum	• Condensate heat exchanger
• Circulation pumps	• Circulation pumps	• Feed water heat exchanger
• Evaporator	• Evaporator	
• Superheater	• Superheater 1	
	• Superheater 2(with injection cooler)	

Figure 10: Steam generator components

3 Methodology

This chapter focuses on identifying the potential heat sources and on the steps to be taken to model a possible solution to optimize the efficiency of the Combined Cycle Gas Turbine Plant in Leuna.

3.1 Hot water System

To assess the viability of the proposed heat sources, first an analysis of the hot water system will be conducted. The objective is to have an estimate of the energy consumption associated with heating water at the hot water station. This will help in evaluating the cost-effectiveness of the system by analysing if the proposed heat source provides enough recoverable energy that would warrant the financial investment required in building the new system. The following diagram represents the readings in winter.

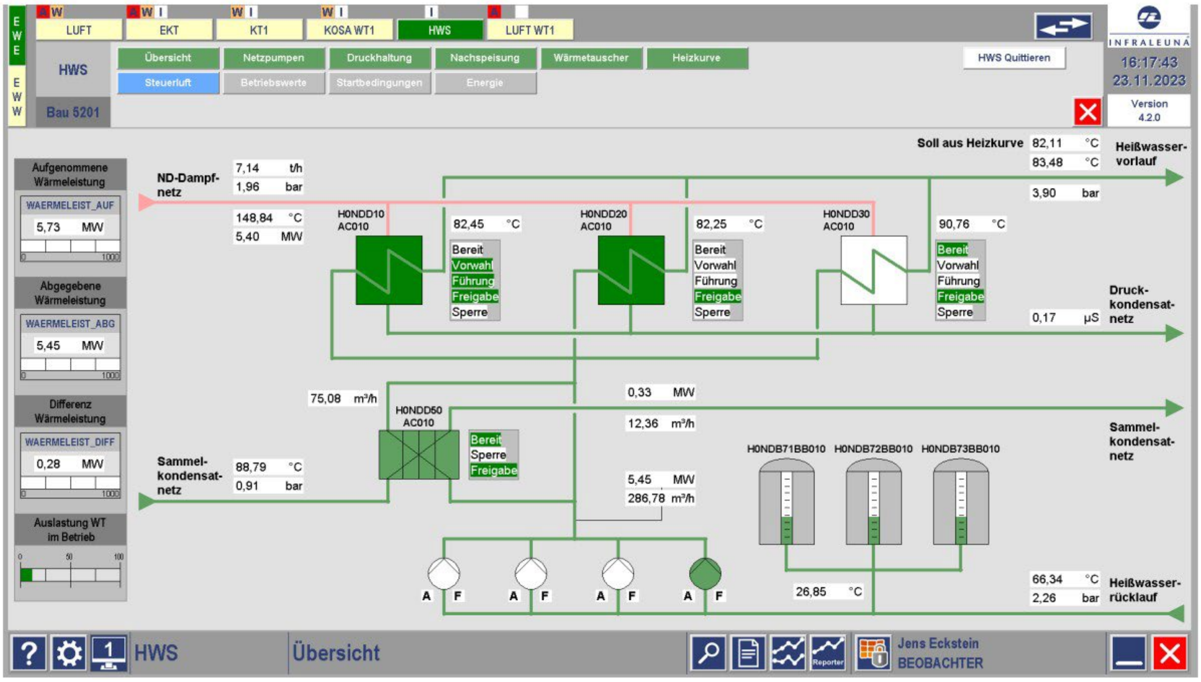

Figure 11: Hot water station overview in Winter

In the hot water station, low steam pressure is used to heat up water to a temperature of 83.48 °C before it is directed to the district heating network. The water returns with a temperature of 66.34 °C and 2.26 barg and is pumped to a flowrate of 286.78 m^3/h. It is then directed to 3 different exchangers. While the temperature remains constant, the flowrate for each exchanger gets divided equally. There the water is again heated up by the low steam pressure and the cycle repeats.

3.1.1 Heat Exchanger calculations

The basic equation for heat transfer in a heat exchanger is:

$$\mathbf{Q = \dot{m} * c_p * \Delta T} \tag{3.1}$$

Where:

Q : heat energy in Joules [J]

$\dot{\mathbf{m}}$: mass flow of water in kilogram per second [kg/s]

$\mathbf{c_p}$: specific heat of the substance in Joule per kelvin per kilogram [J/kgK]

$\Delta\mathbf{T}$: temperature difference in kelvin [K]

The mass flow of the water can be calculated using the formula:

$$\mathbf{\dot{m} = \dot{V} * \rho} \tag{3.2}$$

By replacing the known values of $\dot{V}$ and ρ, the mass flow can be determined. However, equation (3.2) represents the combined mass flow of the water before it is directed into each heat exchanger. Therefore, to calculate the mass flow in each individual heat exchanger, the value obtained needs to be divided by 3.

	Value	SI Unit
Volume flow, $\dot{V}$	268.75	[m^3/h]
Density, ρ (at 74°C)	975	[kg/m^3]
Combined $\dot{m}$ for all 3 heat exchangers	262031	[kg/h]
	72.79	[kg/s]
$\dot{m}$ in each heat exchanger	24.26	[kg/s]

Table 3: Mass flow of water in the heat exchangers

In the first heat exchanger, the water entering has a temperature of 66.34 °C. After the water is heated up by the low-pressure steam, it exits the heat exchanger at a temperature of 82.45 °C. $\Delta\mathbf{T}$ can hence be calculated by replacing the known values in the following equation:

$$\mathbf{\Delta T = \vartheta_{out} - \vartheta_{in}} \tag{3.3}$$

Heat Exchanger	ϑ in [°C]	ϑ out [°C]	Difference in [°C]
1	66.34	82.45	16.11
2	66.34	82.25	15.91
3	66.34	90.75	24.41

Table 4: Temperature difference in each heat exchanger

The Q in each heat exchanger can now be calculated by replacing all known values in equation (3.1). Furthermore, by adding the Q from each heat exchanger, the total heat energy can be calculated. As an example, the values of heat exchanger 1 will be used.

$$\mathbf{Q} = 24.26\ \frac{\mathrm{kg}}{\mathrm{s}} * 4.182\ \frac{\mathrm{kJ}}{\mathrm{kgK}} * 16.11\ \mathrm{K}$$

$$= 1634\ \frac{\mathrm{kJ}}{\mathrm{s}}$$

$$= 1.634\ \mathrm{MW}$$

The same steps are followed to calculate the values for the other heat exchangers.

Heat Exchanger	Q [MW]	Total Q [MW]
1	1.634	5.73
2	1.614	
3	2.477	

Table 5: Heat energy in the heat exchangers

Now that an estimate of the heat energy in each heat exchanger is known, the potential heat sinks can be identified and the potential energy available can be compared.

3.2 Identification of potential heat sinks

The combination of understanding the system design, inspecting physical components and monitoring operational parameters leads to the identification of potential heat sinks. Areas where heat exchange occurs would include components like tubes, fins and other such surfaces that facilitate heat transfer.

The convective heat emitted from the boiler, drums and pipelines are the most common waste heat sources in the boiler. The combined heat increases the temperature in the boiler house to a high of 36 °C and this heat is expelled into the atmosphere via exhaust vents located on the roof at a height of 24 m.

3.3 Steam drum

Even though the boiler house and steam drums are properly insulated, heat loss from the surface still occurs. To calculate the exact position where the highest amount of heat loss occurs, a thermal camera can be used. However, in this thesis, no such experiment has been carried out, and instead an approximate heat loss is calculated using theoretical values.

The steam drums, both low-pressure and high-pressure are made of a heat resistant alloy 15 NiCuMoNb 5. This material is also known as WB36 and belongs to the category of CuNi steels for boiler drums. Its thermal conductivity at a high temperature is expected to be approximately 35 W/(mK). (Steel Grades, n.d.)

The amount of heat lost by convection in the low-pressure steam drum with the following dimensions will be calculated using equation (2.1).

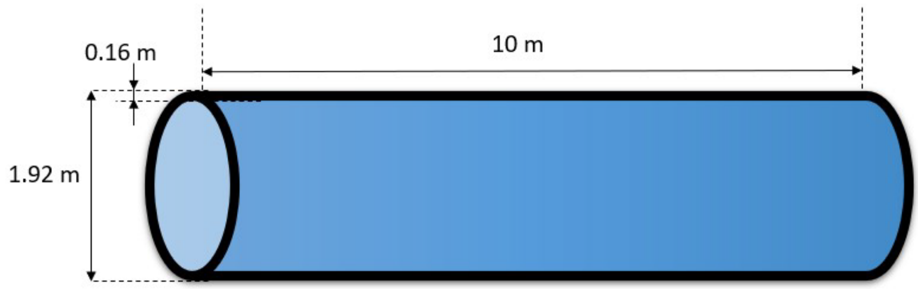

Figure 12: Low-pressure steam drum dimensions

First the surface area is determined as follows:

$$\textbf{Surface Area, A} = (2\pi \times \text{radius}^2) + [(2\pi \times \text{radius}) \times \text{length}] \quad (3.4)$$

$$= (2\pi \times (0.96\ \text{m})^2) + [(2\pi \times 0.96\ \text{m}) \times 10\ \text{m}]$$

$$= 66.2\ \text{m}^2$$

The surface temperature of the sheet Aluminium metal is measured to be 60 °C and the ambient temperature in the boiler house is at a temperature of 36 °C.

Using equations (2.2) to (2.5) the following data is calculated in order to determine the heat loss.

Quantity	Value	SI Unit
α	0.0165	W/(m^2K)
Nu	1.887	-
Re	10	-
Pr	0.7268	-
λ	0.02625	W(mK)
L	3	m
μ	1.895 *10^{-5}	kg/(ms)

Table 6: Values needed to calculate the convective heat loss

The values are read for air at a temperature of 35 °C in the table titles "Properties of air at different temperatures" in the appendix.

By replacing the known values from table 6 in equation (2.1), the heat loss by convection can be calculated:

$$\mathbf{Q_{convection}} = 0.0165\,\frac{W}{m^2 * \mathrm{K}} * 66.2\ \mathrm{m}^2 * (60 - 36)\mathrm{K}$$

$$= 26.2\ \mathrm{W}$$

This amount of heat is barely significant when compared to the power being used in heating up water. Hence, another alternative needs to be sought out.

3.4 Exhaust pipes

The convective heat that is emitted inside the boiler house, mainly from the steam drums, joints and pipes, need to be extracted so that the temperature inside the boiler house does not get too hot. To do so, fans with an efficiency of 80 % are used to extract the heat in the form of hot air. The biggest heat sink is assumed to be the exhaust air vents which expel a considerable amount of convective heat into the atmosphere. This is known as an enthalpy loss. There are 9 exhaust vents located on the roof, out of which only 6 of them are currently in operation. Each exhaust pipe possesses the same properties and are all of the type DRVF 710/30-6 AOSAH 02 AN008.

To assess the potential heat that can be recovered, the temperature, flow rate and composition of exhaust gases need to be defined. These are important in helping select the appropriate heat recovery technology to be used.

	Symbol	Value	
Volume flow	$\dot{V}$	28500	m^3/h
Density	ρ	1.2	kg/m^3
Pressure	Δ_{pt}	30	Pa
Speed of rotation	n	955	Min^{-1}
Power	P_M	5.5	kW
Maximum temperature	t_{max}	40	°C

Table 7: Technical data of each exhaust vent

To understand how much energy can be captured from the exhaust air, the following parameters need to be measured and calculated:

- **T** : the temperature of the exhaust air at point of exit
- $\mathbf{\dot{V}_{air}}$: the airflow rate of the exhaust gas
- $\mathbf{\dot{m}_{air}}$: the rate at which air is being expelled from the exhaust vents
- $\mathbf{\Delta T_{air}}$: the temperature difference of the exhaust gases before and after heat recovery

The specific heat capacity of the exhaust gases. $\mathbf{c_{p,air}}$ varies according to what the composition of the gases are. This value can be read from the thermodynamics table and assumed to be 1.005 kJ/kg. Using equation (3.2), the mass flow of the air of each exhaust vent can be calculated as follows:

$$\mathbf{\dot{m}_{air}} = \mathbf{\dot{V}_{air}} * \boldsymbol{\rho_{air}}$$

$$= 28500 \frac{m^3}{h} * 1.2 \frac{kg}{m^3}$$

$$= 34\ 200 \text{ kg/h}$$

$$= 9.5 \text{ kg/s}$$

To capture the heat from this mass flow, the use of a heat pump would be required. At the point of exit, the air was measured to have a temperature of around 39 °C. Assuming that a temperature of 32 °C would be the exit temperature of the gas from the evaporator, the possible heat recovery from the exhaust air of one exhaust vent can be calculated using equation (3.1) to be:

$$\mathbf{Q} = \dot{\mathbf{m}}_{\mathbf{air}} * \mathbf{c}_{\mathbf{p,\,air}} * \Delta\mathbf{T}$$

$$= 9.5\ \frac{\mathrm{kg}}{\mathrm{s}} * 1.005\ \frac{\mathrm{kJ}}{\mathrm{kgK}} * (39 - 32)\ \mathrm{K}$$

$$= 66.83\ \mathrm{kJ/s}$$

This shows that from one exhaust vent, an approximate of 66.83 kW can be recuperated. When compared with the heat being used in the hot water station, this source seems to be of more significance. Hence more will be elaborated on finding a way to recuperate this heat.

The technology to be used for this recovery will be an air to water heat pump. Parameters such as refrigerant, exit temperature, number of exhaust vents to be used will be varied to choose the most efficient way to install a heat pump.

3.5 Modelling an appropriate design

The heat from the exhaust vents can be captured using an air to water heat pump. However, to design the model that will most efficiently provide significant heat for district heating, different parameters such as mass flow, position of heat pump components must be taken into consideration. Different models will be proposed and at the end the most suitable option will be chosen.

The assumptions that will be made are:

1. The difference in temperature of the exhaust air at point of entry and point of exit from the evaporator will be a constant of **7 K**.
2. The heat pump will be operating in heating mode only
3. The temperature difference over the internal heat exchanger will be in the limits of subcooling and superheating, which in this case will be **5 K**.
4. Since these are not ideal conditions, superheating will occur at the exit point from the evaporator and superheating will occur at the exit point of the condenser.
5. The water entering the condenser from district heating will be **70 °C** and the water returning will be **80 °C**.
6. No heat loss occurs due to the insulation.

3.5.1 Model 1: Basic Heat Pump Circuit using R134a

This basic model will comprise of the widely used hydrofluorocarbon refrigerant R134a, also known as 1,1,1,2-Tetrafluoroethane (CH_3CH_2F). R134a is an affordable and non-flammable gas extensively used in the domestic refrigeration, most notably in automobile air conditioners.

Critical temperature	101.06 °C
Ozone Depletion potential (ODP)	0
Global warming potential (GWP)	1430

Table 8: Properties of R134a

Only one exhaust vent is taken into consideration and the temperature at point of entry in the evaporator is assumed to be 39 °C. The heat pump is proposed to be installed on the roof right next to the exhaust vent to maximize the capture of the air flow. The following schema depicts the proposed model for the heat pump system.

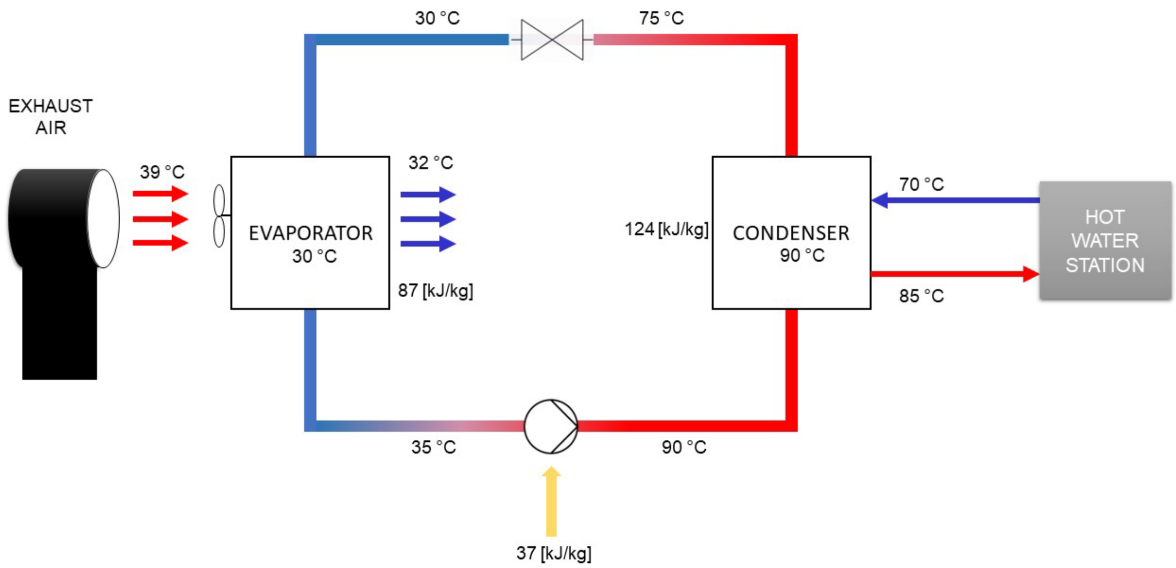

Figure 13: Heat pump system using R134a

The Solkane 8.0.0 app is used to sketch the log p-H diagram and derive the output parameters including enthalpy(h), entropy(s) of the suggested heat pump model.

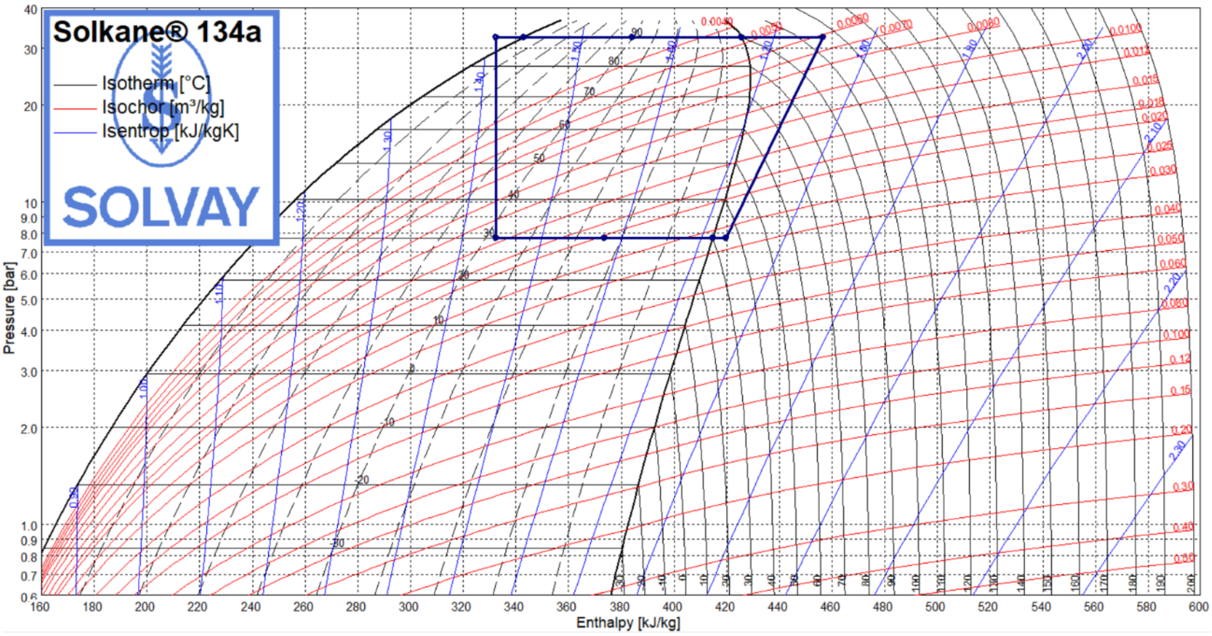

Figure 14: log p-H diagram of R134a

From the p-H diagram, the following points can be read. Using these values, we can calculate the energy being absorbed in the evaporator, the energy released in the condenser and the energy require to operate the compressor.

	p	t	v	h	s	x
Point	bar	°C	dm³/kg	kJ/kg	kJ/kgK	--
1	7.70	35.00	27.48	419.88	1.7311	
2s	32.44	101.23	5.82	449.33	1.7311	
2	32.44	105.69	6.16	456.69	1.7506	
3	32.44	105.69	6.16	456.69	1.7506	
3'	32.44	90.00	4.62	425.63	1.6667	
3"4'm	32.44	90.00	2.91	384.36	1.5529	
4'	32.44	90.00	1.19	343.08	1.4391	
4	32.44	85.00	1.13	332.51	1.4116	
5	7.70	30.00	14.38	332.51	1.4425	0.525
56"m	7.70	30.00	20.51	373.60	1.5783	
6"	7.70	30.00	26.65	414.69	1.7141	
6	7.70	35.00	27.48	419.88	1.7311	

Table 9: Output parameters using R134a

- Energy absorbed (h_6-h_5) = 87.37 kJ/kg
- Energy released (h_2-h_4) = 124.18 kJ/kg
- Energy for compressor (h_2-h_1) = 36.81 kJ/kg

In the evaporator, the energy given out from the exhaust air, Q_{out}

$$\mathbf{Q_{out} = \dot{m}_{air} * c_{p,\,air} * \Delta T}$$

is equal to the energy absorbed by the refrigerator, Q_{in}.

$$\mathbf{Q_{in} = \dot{m}_R * \Delta h(6-5)} \qquad \textbf{(3.5)}$$

Hence, the mass flow of the refrigerant, $\dot{m}_R$ can be calculated using the following equation:

$$\mathbf{\dot{m}_R = \frac{\dot{m}_{air} \cdot c_{p,air} \cdot \Delta T}{\Delta h(6-5)}} \qquad \textbf{(3.6)}$$

By replacing the known values in the equation (3.6) , $\mathbf{\dot{m}_R}$ is equal to:

$$\mathbf{\dot{m}_R} = \frac{9.5\,\frac{kg}{s} * 1.005\,\frac{kJ}{kgK} * 7\ K}{87.37\,\frac{kJ}{kg}}$$

$$= 0.765\ kg/s$$

The energy, Q at each stage can be calculated by simply multiplying the enthalpy at each stage with $\dot{m}_R$.

- Q absorbed = $87.37\,\frac{kJ}{kg} * 0.765\,\frac{kg}{s}$ = 66.83 kW
- Q released = $124.18\,\frac{kJ}{kg} * 0.765\,\frac{kg}{s}$ = 94.99 kW
- Q compressor = $36.81\,\frac{kJ}{kg} * 0.765\,\frac{kg}{s}$ = 36.81 kW

The coefficient of performance (COP) of the heating system is a measure to evaluate its efficiency. It is defined as the ratio of the heat released by the condenser to the amount of energy required to operate the compressor. The higher the COP, the better the efficiency. Using the equation (3.7), the COP can be calculated:

$$\mathbf{COP = \frac{Condenser\ heat\ output}{Compressor\ power}} \qquad \textbf{(3.7)}$$

$$= \frac{124.18 \frac{kJ}{kg}}{36.81 \frac{kJ}{kg}}$$

$$= 3.37$$

The theoretical COP depicts the ideal efficiency the heat pump system under perfect conditions. However, in real life this cannot be achieved due to factors such as friction and heat losses within the system. Hence it serves as a benchmark to evaluate the performance of the heat pump system and can be calculated as follows:

$$\mathbf{COP_{theoretical}} = \frac{\mathbf{T_{condensation}}}{\mathbf{T_{condensation}} - \mathbf{T_{evaporation}}} \qquad (\mathbf{3.8})$$

$$= \frac{(90 + 273)\mathrm{K}}{(90 - 30)\ \mathrm{K}}$$

$$= 6.05$$

The considerable difference between the theoretical COP and the actual COP indicates that the system would operate at a significantly lower efficiency compared to its theoretical potential. To improve the system's efficiency, alternative refrigerants with better performance characteristics can be considered.

3.5.2 Model 2: Heat Pump Circuit using an IHX with R365mfc

Critical temperature	186.85 °C
Ozone Depletion potential	0
Global warming potential	825

Table 10: Properties of R365mfc

The refrigerant R365mfc also known as 1,1,1,3,3-pentafluorobutane is suitable for operating high temperatures, though it is flammable. It is mainly used as a foam blowing agent. In this case, the typical refrigeration cycle cannot be used for there would be an isentropic discharge point in the two-phase region. Hence an internal heat exchanger needs to be installed before the compressor to aid in increasing the temperature without any additional equipment. The following schema is a proposed model to design the heat pump to be installed on the roof.

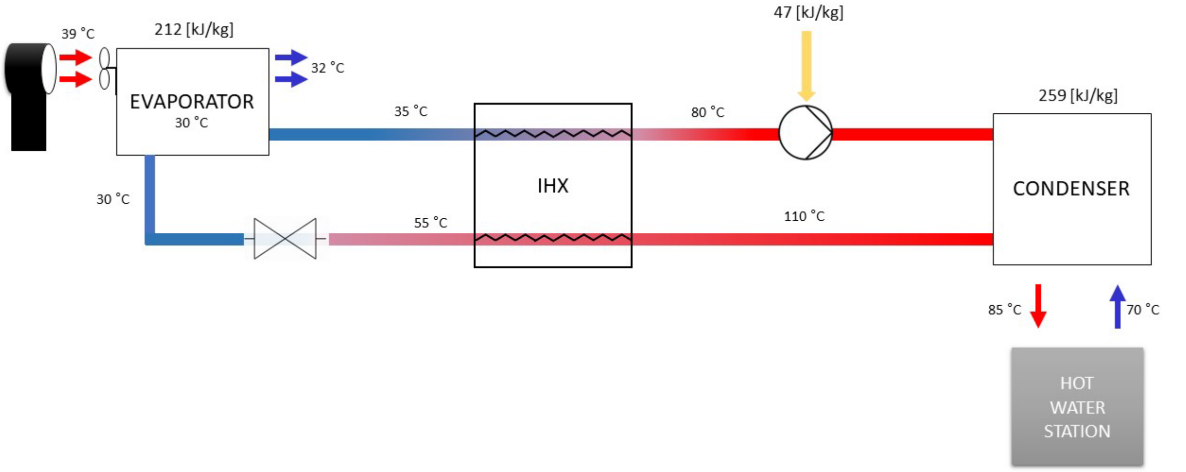

Figure 15: Heat pump system with R365mfc.

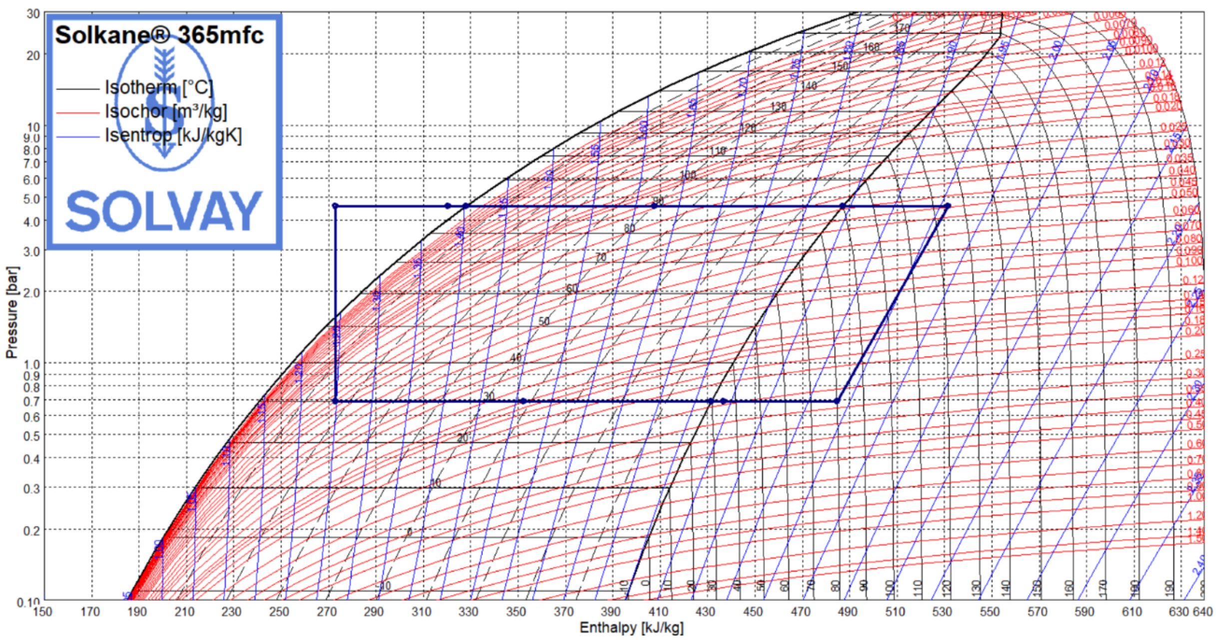

Figure 16: log p-H diagram of R365mfc

From the p-H diagram, the following points can be read and the enthalpy at the different stages can be calculated.

	p	t	v	h	s	x
Point	bar	°C	dm³/kg	kJ/kg	kJ/kgK	--
1	0.69	80.00	280.98	484.60	1.9344	
2s	4.58	117.52	43.30	522.12	1.9344	
2	4.58	124.74	44.43	531.50	1.9581	
3	4.58	124.74	44.43	531.50	1.9581	
3'	4.58	90.00	38.70	487.30	1.8419	
3"4'm	4.58	90.00	19.80	407.80	1.6224	
4'	4.58	90.00	0.91	328.31	1.4028	
4	4.58	85.00	0.90	320.67	1.3819	
5	4.58	52.74	0.84	272.84	1.2438	
6	0.69	30.00	40.63	272.84	1.2483	0.168
67''m	0.69	30.00	138.92	352.29	1.5108	
7''	0.69	30.00	237.21	431.75	1.7732	
7	0.69	35.00	241.69	436.77	1.7896	
8	0.69	35.00	241.69	436.77	1.7896	

Table 11: Temperature, enthalpy and entropy of R365mfc

- Energy absorbed (h_7-h_6) = 163.93 kJ/kg
- Energy released (h_3-h_4) = 210.83 kJ/kg
- Required energy for pump (h_2-h_1) = 46.9 kJ/kg
- Energy in IHX (h_4-h_5) = 47.83 kJ/kg

The calculations follow the same steps as the ones illustrated in Model 1.

$\dot{m}_R$	0.407	kg/s
Q absorbed	66.83	kW
Q released	85.95	kW
Q compressor	19.12	kW
Q IHX	19.50	kW
COP	4.5	
COP theoretical	6.05	

Table 12: Energy values for the model 2

This refrigerant shows a better COP comparison. This model would require the installation of 6 or 9 such heat pump systems on the roof, depending on how many exhaust fans to be operated. However, the roof might not be able to withhold such a heavy weight. A different location for the heat pump needs to be explored.

3.5.3 Model 3: Heat Pump System on the ground

The roof of the boiler house was originally not designed to support heavy weights which makes installing 6 to 9 heat pumps not realistically possible. The vibrations of the compressors would additionally cause problems with the stability of the roof. An alternative would be to locate the heat pump on the ground and use ducts to direct the exhaust air to the evaporator. The following schema could be an idea on how to go on with the modelling.

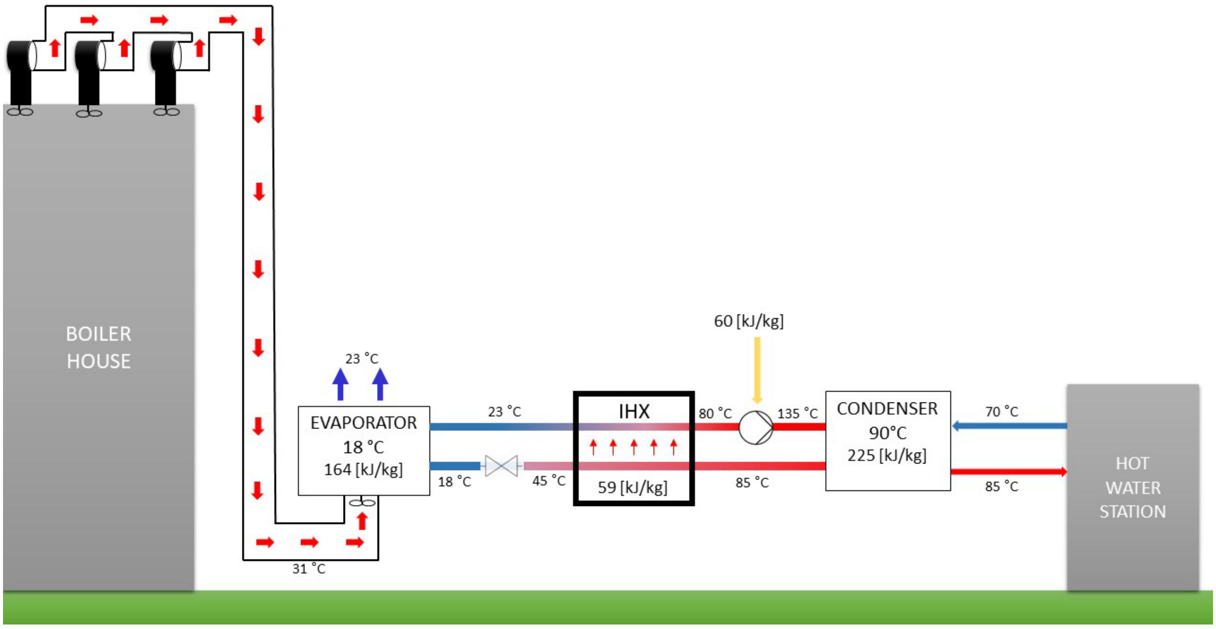

Figure 17: Heat pump system on the ground

This design utilizes a single main duct to merge the airflow from all the vents into one unit, altering the mass flow and temperature at the point of entry in the evaporator. In order to calculate these new values, the first step was to measure the temperature of the air of each exhaust vent at the point of exit using sensors. However, the temperature for exhaust vent number 9 was not found.

Exhaust vent		1	2	3	4	5	6	7	8	9
Temperature, T	°C	37	36	40	34	21	17	17	20	-
	K	310	309	313	307	294	290	290	293	-

Table 13: Air temperature of each exhaust vent at point of exit

For this model, only the first 6 vents will be considered. To determine the temperature at which the air enters the evaporator, first we need to calculate the combined mass flow of air, $\dot{m}_1$

$$\dot{\mathbf{m}}_{\mathbf{1}} = \dot{\mathbf{m}}_{\mathbf{air}} * \mathbf{Number\ of\ vents} \tag{3.9}$$

$$= 9.5\ \frac{\mathrm{kg}}{\mathrm{s}} * 6$$

$$= 57\ \mathrm{kg/s}$$

With $\dot{m}_1$ we can now determine the new combined temperature, T_1:

$$\mathbf{T_1} = \frac{(\mathbf{T\ exhaust\ vent\ 1} * \dot{\mathbf{m}}_{\mathbf{air}}) + (\mathbf{T\ exhaust\ vent\ 2} * \dot{\mathbf{m}}_{\mathbf{air}}) + \cdots + (\mathbf{T\ exhaust\ vent\ 6} * \dot{\mathbf{m}}_{\mathbf{air}})}{\dot{\mathbf{m}}_{\mathbf{1}}}$$

$$= \frac{\left((37 * 9.5) + (36 * 9.5) + (40 * 9.5) + (34 * 9.5) + (27 * 9.5) + (17 * 9.5)\right) {}^\circ\mathrm{C} * \frac{\mathrm{kg}}{\mathrm{s}}}{57\ \frac{\mathrm{kg}}{\mathrm{s}}}$$

$$= 31\ {}^\circ\mathrm{C}$$

With the known value of the new mass flow and temperature at point of entry in the evaporator, we can proceed with the same steps as in Model 1 to calculate the energy values.

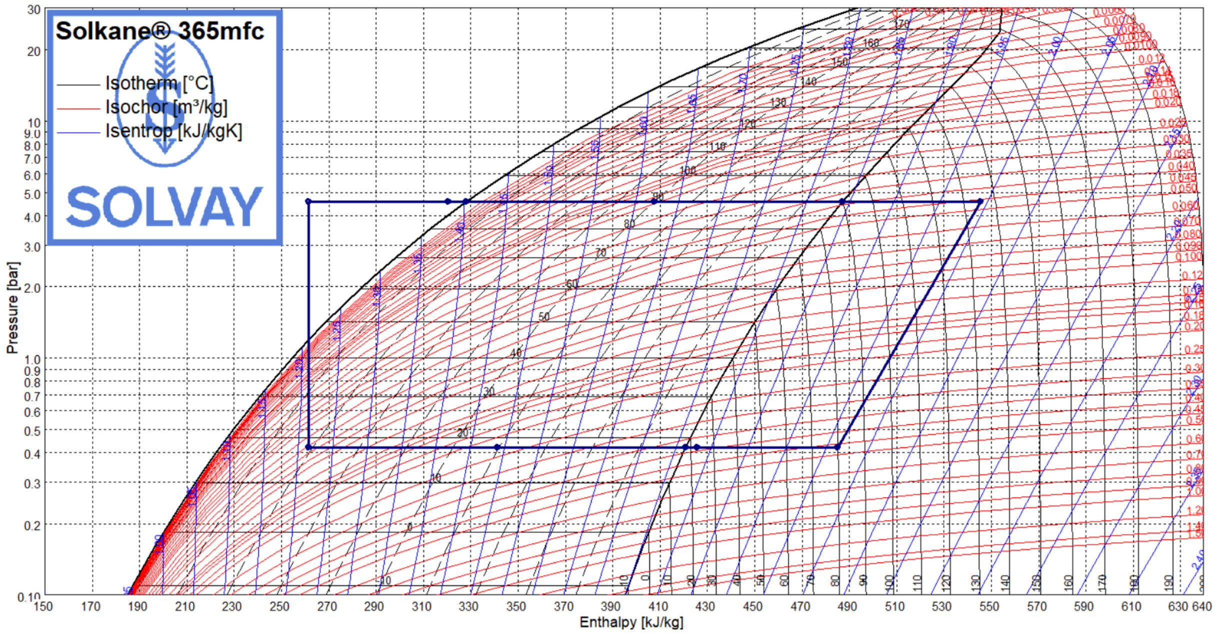

Figure 18: log p-H diagram of R365mfc with combined temperature

	p	t	v	h	s	x
Point	bar	°C	dm³/kg	kJ/kg	kJ/kgK	--
1	0.42	80.00	462.08	485.17	1.9630	
2s	4.58	126.23	44.66	533.45	1.9630	
2	4.58	135.37	46.05	545.52	1.9929	
3	4.58	135.37	46.05	545.52	1.9929	
3'	4.58	90.00	38.70	487.30	1.8419	
3"4'm	4.58	90.00	19.80	407.80	1.6224	
4'	4.58	90.00	0.91	328.31	1.4028	
4	4.58	85.00	0.90	320.67	1.3819	
5	4.58	44.73	0.82	261.35	1.2085	
6	0.42	18.00	71.60	261.35	1.2142	0.189
67´´m	0.42	18.00	223.67	341.18	1.4890	
7´´	0.42	18.00	375.75	421.02	1.7638	
7	0.42	23.00	382.86	425.85	1.7802	
8	0.42	23.00	382.86	425.85	1.7802	

Table 14: Temperature, enthalpy and entropy of R365mfc with new T_1

- Energy absorbed (h_7-h_6) = 164.5 kJ/kg
- Energy released (h_3-h_4) = 224.85 kJ/kg
- Required energy for pump (h_2-h_1) = 60.35 kJ/kg
- Energy in IHX (h_4-h_5) = 59.32 kJ/kg

$\dot{m}_R$	2.71	kg/s
Q absorbed	445.9	kW
Q released	609.4	kW
Q compressor	163.6	kW
Q IHX	160.8	kW
COP	3.73	
COP theoretical	5.04	

Table 15: Energy values for the model 3

By altering the number of vents used in this model, the heat output and the COP can be varied.

Number of exhaust vents	**8**	**6**	**4**	**SI Unit**
	1 to 8	1 to 6	1 to 4	
Combined temperature	28	31	37	°C
Combined mass flow	76	57	38	kg/s
Q absorbed	590	446	292	kW
Q released	821	609	387	kW
Q compressor	230	164	95	kW
COP	3.57	3.73	4.07	-
COP theoretical	4.84	5.04	5.5	-

Table 16: Comparison of values by varying the number of exhaust vents used

4 Results and Discussion

This chapter presents the results obtained from the analysis of potential heat sources and the modelling of the solutions aimed at optimizing the efficiency of the CCGTP in Leuna. The focus is on assessing the viability of utilizing heat sources to enhance energy efficiency, particularly in the hot water system and waste heat recovery from the boiler house.

4.1 Waste heat source

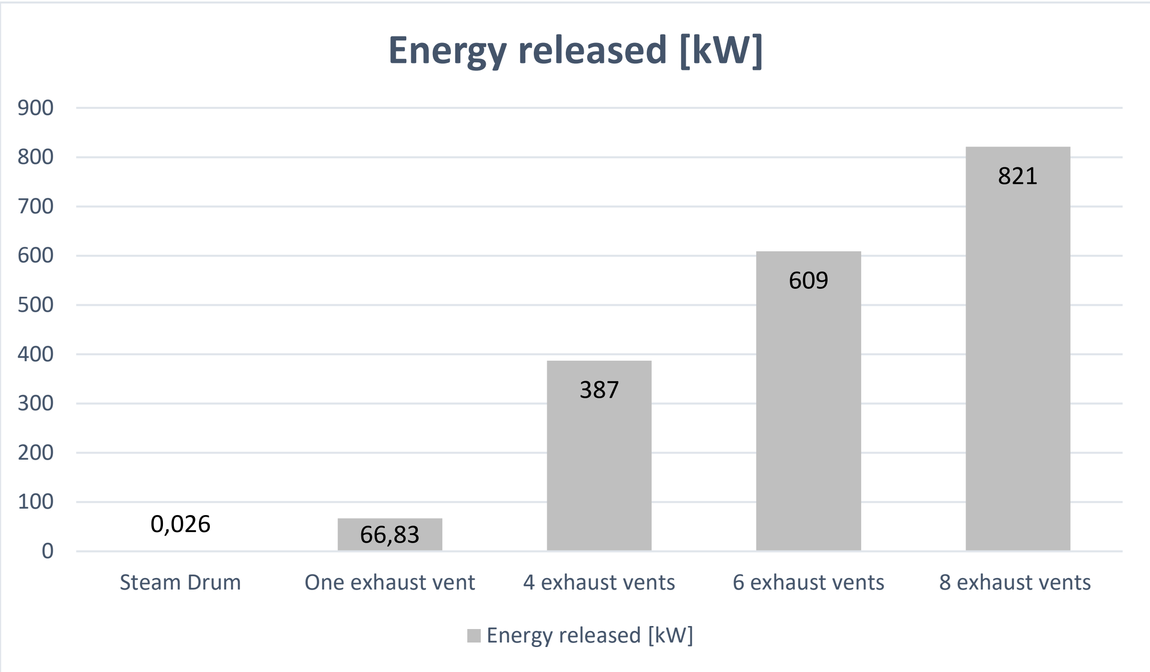

Chart 1: Energy released by potential heat sinks

By comparing the amount of energy we can obtain from each potential heat sinks, it can be deduced that the exhaust vents provide the highest potential in waste heat recovery. Whether it is used in a combination of 4,6 or 8, it still surpasses the energy available to be recovered from the system.

4.2 Most efficient design

Model		Energy released [kW]	Energy for compressor [kW]	COP	COP theoretical
1		95	28	3.37	6.05
2		86	19	4.50	6.05
3	8 vents	821	230	3.57	4.84
	6 vents	609	164	3.72	5.04
	4 vents	292	95	4.07	5.5

Table 17: Comparison of values for the different models proposed

4.2.1 Choosing the refrigerant

By comparing the COP, refrigerant R365mfc offers improved efficiency. It is compatible with metals commonly used in refrigeration systems and has a lower GWP. However, it is flammable and its application as a refrigerant for high temperature heat pumps is still in development.

However, a more suitable replacement for R365mfc is refrigerant R600a, also known as isobutane. It is a hydrocarbon based refrigerant with a critical temperature of 134.98 °C, making it suitable for operating high temperatures. It significantly outperforms the most common refrigerants such as R134a in terms of environmental impact. With a GWP of only 3 (99.78% lower than R134a) and 0 ODP, adopting R600a contributes to ozone layer preservation, minimizes climate change, reduces energy consumption and hence greenhouse gas emissions. Due to its lower toxicity level, it poses minimal risk to human health and the environment in case of accidental leaks. (United Refrigerants, 2023)

4.2.2 Choosing the location

The roof of the boiler house was initially not engineered to withstand substantial loads making the installation of more than one heat pump impractical. Furthermore, the vibrations from the compressors would also compromise the roof's stability. The installation of a main single duct is way lighter. However, the system does not need to be situated directly on the ground. It could be positioned one or two levels lower, utilizing the steel frameworks on the lower decks for installation. This approach would be more practical, as placing it directly on the ground would necessitate considerable effort to connect the district heat piping to the condenser, which can be quite heavy.

4.2.3 Choosing the number of exhaust vents

When comparing the different configurations based on the number of vents, distinct trade-offs in energy release, compressor energy requirements, and resulting COP values can be observed. Operating 8 vents would generate a higher energy output but with the lowest COP while operating 4 vents would offer the highest COP but with lowest energy release. Considering the trade-offs, the 6 vents configuration strikes a balance between energy release and compressor energy requirements, offering a reasonable energy output while achieving a relatively higher COP.

It is not common to run all 3 blocks in the plant. So, when all the heating ducts will be installed, there could be cases where there will be some fans that will not be operating due to the low exhaust heat that rises in the boiler house. 6 fans running is the most common mode of operation for heat extraction in the boiler house.

4.2.4 Alternative proposals

An alternative to the vents would be to install a single huge fan and have the air extracted only from there. It is a possible solution which would minimize the number of ducts to be installed on the roof, however it needs to be considered that there are 3 blocks emitting heat in the boiler house. If the hot air is retracted from one spot, we need to consider the fact that all places between the blocks need to be cooled in a rather similar way. Therefore, some ducting would have to be installed inside of the boiler house so as to direct the hot air to the fan.

4.3 Environmental Impact

Greenhouse Gas Emissions Reduction

By efficiently recovering waste heat from exhaust vents and other sources, the need for additional energy generation—often reliant on fossil fuels—can be minimized. This leads to a decrease in greenhouse gas emissions associated with conventional energy production methods, contributing to climate change mitigation.

Resource Conservation

Repurposing waste heat that would otherwise dissipate into the atmosphere conserves valuable resources. Capturing and utilizing this heat reduces overall demand for energy resources like natural gas or coal, promoting sustainable energy consumption practices.

Enhanced Energy Efficiency

Implementing heat pump systems and efficient heat recovery technologies improves the plant's overall energy efficiency. This translates to reduced energy consumption per unit of output, resulting in cost savings and a smaller environmental footprint per unit of energy produced.

Environmentally Friendly Refrigerants

Considering refrigerants with lower Global Warming Potential and Ozone Depletion Potential, such as R365mfc and R600a, demonstrates commitment to environmental stewardship. These refrigerants offer better performance while minimizing their impact on global warming and ozone layer depletion.

Holistic Design and Operation

Thoughtfully locating heat pump systems, optimizing exhaust vent usage, and exploring alternative proposals (like installing a single large fan) contribute to minimizing environmental impact. By evaluating various options and selecting efficient configurations, the plant can maximize energy recovery while minimizing resource use and emissions.

In summary, the proposed solutions pave the way for enhancing the environmental sustainability of the Combined Cycle Gas Turbine Plant in Leuna.

4.4 Parameters to take into consideration.

4.4.1 Water temperature

A mandatory health related requirement of the district hot water supply system is that the water temperature should always be above 60 °C to ensure that the legionella bacteria does not survive and reproduce.

4.4.2 Installation process

- Dampers or louvers need to be installed in the duct system to control the flow of air. This allows regulation of airflow based on heating demands and system requirements.
- Filters need to be included in the ductwork to remove any potential contaminants from the air before it reaches the evaporator.
- The evaporator of the heat pump should be installed where it can efficiently absorb the most heat possible from the exhaust air.

- In order to prevent overheating and enhance the heat exchange process, there should be a proper ventilation and airflow around the heat pump units.
- Plumbing lines need to be installed for proper flow of the refrigerant.
- An electrical connection needs to be included for the power supply.
- The piping and ductwork need to be insulated to minimize the heat loss; this will help to maintain the heat captured by the heat pump.

5 Conclusion

The results indicate promising opportunities for waste heat recovery in the boiler house from the exhaust vents. By harnessing wasted heat, significant energy savings can be achieved, contributing to the overall efficiency of the Combined Cycle Gas Turbine Plant. The comparison of different waste heat recovery models underscores the importance of selecting appropriate refrigerants and system configurations to maximize efficiency. While all models exhibit potential for heat recovery, factors such as COP and theoretical efficiency vary, necessitating careful consideration during system design and implementation.

By using the amount of heat used in the hot water station as a comparison, the amount of heat that can be recovered from the exhaust vent is deemed to be significant enough to invest in a new WHR system.

In conclusion, the results presented in this chapter provide valuable insights into potential heat sources and waste heat recovery strategies for optimizing energy efficiency in the Combined Cycle Gas Turbine Plant. Further research and experimentation will be necessary to refine modelling approaches and validate the feasibility of proposed solutions in real-world scenarios.

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9 Appendix

GWP of HFR refrigerants (Australian Government, 2021)

Code	Substance Name	Chemical Name	Chemical Formula	GWP (AR4)
R-125	HFC-125	Pentafluoroethane	CHF_2CF_3	**3,500**
R-134A	HFC-134a	Tetrafluoroethane	CH_2FCF_3	**1,430**
R-143A	HFC-143a	Trifluoroethane	CF_3CH_3	**4,470**
R-152A	HFC-152a	Difluoroethane	CH_3CHF_2	**124**
R-227EA	HFC-227EA	Heptafluoropropane	CF_3CHFCF_3	**3,220**
R-23	HFC-23	Trifluoromethane / Fluoroform	CHF_3	**14,800**
R-236CB	HFC-236CB		$CH_2FCF_2CF_3$	**1,340**
R-236EA	HFC-236EA		CHF_2CHFCF_3	**1,370**
R-236FA	HFC-236FA	Hexafluoropropane	$CF_3CH_2CF_3$	**9,810**
R-245CA	HFC-245CA		$CH_2FCF_2CHF_2$	**693**
R-245FA	HFC-245FA	Pentafluoropropane	$CHF_2CH_2CF_3$	**1,030**
R-32	HFC-32	Difluoromethane	CH_2F_2	**675**
R-365MFC	HFC-365MFC	Pentafluorobutane	$CF_3CH_2CF_2CH_3$	**794**
R-41	HFC-41	Fluoromethane (or Methyl fluoride)	CH_3F	**92**
R-43-10MEE	HFC-43-10MEE	Decafluoropentane	$CF_3CHFCHFCF_2CF_3$	**1,640**

Properties of air at different temperatures (Engineers Edge, 2024)

Temp T, °C	Density Air ρ, kg/m^3	Specific Heat of Air c_p, J/kg-K	Thermal Conductivity Air k, W/m-K	Thermal Diffusivity Air α, m^2/s	Dynamic Viscosity Air μ, kg/m-s	Kinematic Viscosity Air ν, m^2/s	Prandtl Number Air Pr
- 150	2.866	983	0.01171	4.158×10^{-6}	8.636×10^{-6}	3.013×10^{-6}	0.7246
- 100	2.038	966	0.01582	8.036×10^{-6}	1.189×10^{-5}	5.837×10^{-6}	0.7263
- 50	1.582	999	0.01979	1.252×10^{-5}	1.474×10^{-5}	9.319×10^{-6}	0.7440
- 40	1.514	1002	0.02057	1.356×10^{-5}	1.527×10^{-5}	1.008×10^{-5}	0.7436
- 30	1.451	1004	0.02134	1.465×10^{-5}	1.579×10^{-5}	1.087×10^{-5}	0.7425
- 20	1.394	1005	0.02211	1.578×10^{-5}	1.630×10^{-5}	1.169×10^{-5}	0.7408
- 10	1.341	1006	0.02288	1.696×10^{-5}	1.680×10^{-5}	1.252×10^{-5}	0.7387
0	1.292	1006	0.02364	1.818×10^{-5}	1.729×10^{-5}	1.338×10^{-5}	0.7362
5	1.269	1006	0.02401	1.880×10^{-5}	1.754×10^{-5}	1.382×10^{-5}	0.7350
10	1.246	1006	0.02439	1.944×10^{-5}	1.778×10^{-5}	1.426×10^{-5}	0.7336
15	1.225	1007	0.02476	2.009×10^{-5}	1.802×10^{-5}	1.470×10^{-5}	0.7323
20	1.204	1007	0.02514	2.074×10^{-5}	1.825×10^{-5}	1.516×10^{-5}	0.7309
25	1.184	1007	0.02551	2.141×10^{-5}	1.849×10^{-5}	1.562×10^{-5}	0.7296
30	1.164	1007	0.02588	2.208×10^{-5}	1.872×10^{-5}	1.608×10^{-5}	0.7282
35	1.145	1007	0.02625	2.277×10^{-5}	1.895×10^{-5}	1.655×10^{-5}	0.7268
40	1.127	1007	0.02662	2.346×10^{-5}	1.918×10^{-5}	1.702×10^{-5}	0.7255
45	1.109	1007	0.02699	2.416×10^{-5}	1.941×10^{-5}	1.750×10^{-5}	0.7241

10 Affidavit

I hereby certify that I have written the thesis independently and have not used any sources or aids other than those specified, that all statements taken verbatim or analogously from other writings have been identified and that the thesis has not yet been part of a coursework or examination in the same or a similar version.

Signature of the author