Master Thesis

Viability of High Temperature Heat Pump with a CO_2 Brayton Cycle for an Industrial Installation

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Abstract: This project consists of studying the utilization of a low-temperature waste heat flow (60-70 °C) from an industrial process to produce steam for the same industry, based on an inverse Brayton Cycle with CO_2.

The steam produced enables the elimination of an extraction cogeneration turbine present in the industry, thereby generating additional electricity and partially offsetting the fuel consumed by the heat pump.

To achieve this, two scenarios are considered, determined by the available waste heat and the proposed steam usage conditions. In addition, the heat pump will be modeled and sized, while assessing the economic viability, LCOS and LCOH, and the benefits gained.

In the best conditions, the COP is 2.03. The dimensions of the heat pump are 21m x 21m x 25 m.

In economic terms, the installation requires a total investment of 44,383,762 €, which corresponds to a nominal heat cost of 611 €/kW. Costs related to heat (LCOH) and steam (LCOS) are also analyzed, based on the fraction of time the steam operates at maximum production (α). It appears that for α values higher than 0.64, the costs normalize between 51 €/MWh and 62 €/MWh (38.8 €/t and 47.2 €/t). On the other hand, the operational cost (Total OPEX) ranges between 43.5 €/MWh and 46 €/MWh (33 €/t and 35 €/t).

Keywords: Heat pump, waste heat, steam.

1. Introduction

a. Project Context

The climate crisis is a current issue affecting all types of existing industries. The increase in CO2eq emissions has been significant in recent decades, as shown in Figure 1 (Hannah Ritchie, 2020).

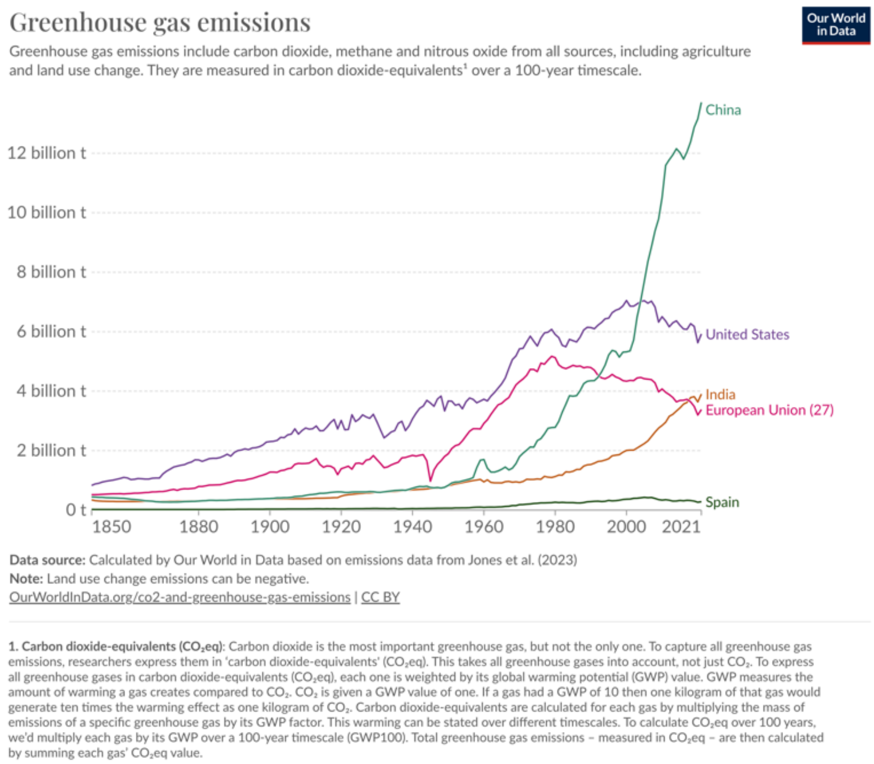


Figure 1 - GHG Emissions (Hannah Ritchie, 2020)

Therefore, measures are needed to slow this rate of emissions. The industry is currently exploring possible solutions, such as reduction, compensation, and CO2 capture. The goal is to focus on those industries that drive climate change, not only to slow their emission intensity but also to optimize the use of natural resources like water.

This project is part of the research by the Repsol Foundation Chair of Energy Transition at Comillas Pontifical University (Comillas, 2023). Its mission is to explore measures to reduce the carbon footprint in various industrial sectors, with this year's focus on revalorizing residual heat from industrial processes to produce steam.

b. The Use of Steam in Industry

Currently, steam plays a crucial role in a wide range of activities with their corresponding physical characteristics. Particularly in the industrial sector, steam is often found as saturated or slightly superheated steam. Additionally, it is a fundamental element in modern industry and is present in a wide range of sectors. Its versatility and efficiency make it an indispensable ally for various processes (HS, 2024):

- <u>Electricity Generation</u>: steam drives turbines, which generate electricity through rotation. This steam is produced by heat absorption and is transformed into mechanical energy that causes the turbine to move. Steam turbines are crucial in cogeneration, a process that involves the simultaneous production of electric-mechanical and thermal energy from the same fuel. The process involves high-pressure steam entering the turbine, causing its rotation, and exiting at low pressure. The final steam is reused for other thermal industrial processes due to its high temperature, such as water heating. Cogeneration is a highly efficient solution for industrial processes that require a large amount of thermal and electrical energy, as it maximizes fuel use, reduces primary energy consumption, emissions, and economic costs. It also provides energy independence, autonomy, and supply security in the plants that contain them (REPSOL, 2024).

- Sterilization: in the form of saturated steam, it is a high-temperature process commonly used as a disinfectant in the healthcare or food sectors.
- Food Processing: Steam is an essential element for the execution of the pre-treatment required for food before consumption. Its high heat capacity allows for quick and efficient energy transfer, ensuring the elimination of microorganisms and food safety. An example of this application is milk pasteurization.
- Climate Control: Both cooling and heating systems use steam for temperature regulation.

c. Cooling Towers

Cooling towers are essential elements and are very present in modern industry, especially in sectors that require high temperatures such as petrochemical, metallurgical, chemical industries, etc. Their main function is to dissipate the heat of the circulating fluid through the different stages of industrial processes, allowing for efficient and safe operation. Their operation consists of a heat exchange between a hot source (steam/gas) and a cold source (water).

As shown in Figure 2 the heat to be dissipated in the heat exchanger HE heats a stream of water, which is sprayed from the top of the tower over an ascending stream of air. Some of the falling water droplets absorb heat from those around them and evaporate, increasing the relative humidity of the ascending air. Thus, the water that reaches the bottom tray is cooler and in smaller quantity than the sprayed water, requiring a makeup supply to maintain the tray level.

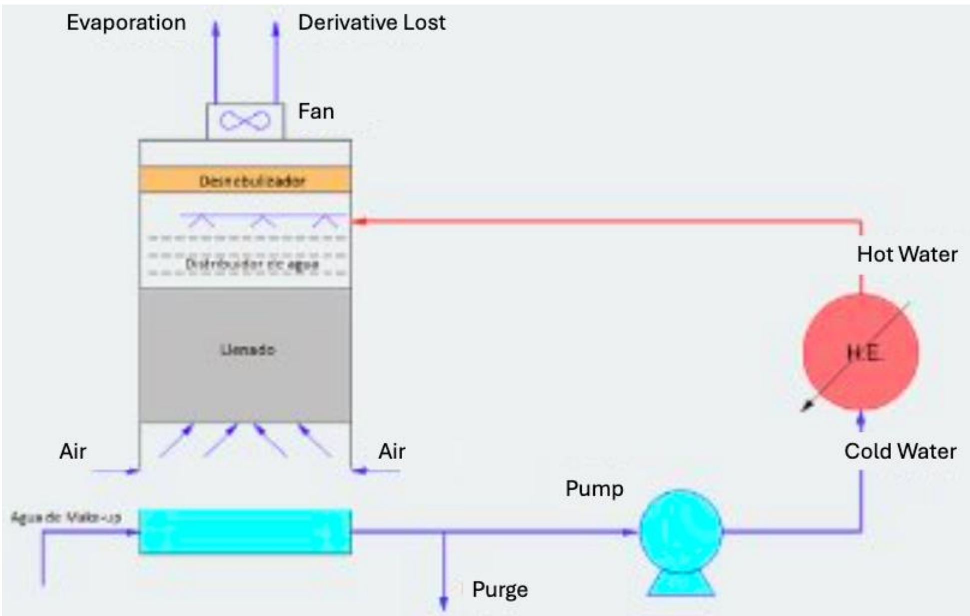


Figure 2 - Cooling Tower (Lenntech, 2024)

In regions with water scarcity or water stress, the use of cooling towers can significantly impact resource availability. To minimize this impact, currently a pressing issue in Spain, various technologies and strategies are needed to save water, such as recirculation and reducing the emission of residual heat to the air.

d. Project Definition and Motivation

Each year, the Chair (Comillas, 2023) conducts a study analyzing potential decarbonization measures for different industrial sectors that are potentially polluting and have high emissions, conducting a comprehensive analysis and reflecting their economic viability. It began in the 2020/21 academic year, analyzing the decarbonization of the automotive industry; in the 2021/22 academic year, the chosen sector was the ceramic industry, and in the 2022/23 academic year, the cement industry was selected.

Finally, in the 2023-24 academic year, it was decided to address a topic transversal to many industries, which is the revalorization of residual heat using a heat pump, making it possible to generate process steam. Commercially available industrial heat pumps typically do not exceed 150°C, with various research projects aiming to reach between 200°C and 250°C. An example is the SUSHEAT project with its Stirling Based High Temperature Heat Pump, whose process includes two isochoric and two isothermal state changes and aims to achieve a COP of 1.9 at 250°C starting from 70°C (Barbero, 2023).

Furthermore, when residual heat is dissipated through a cooling tower, typically 2 to 5 m^3 of water is consumed per MWh transferred. While this is not a large amount, during periods of water stress, it can be a significant restriction. If this heat, instead of being dissipated into the environment, is used to power a heat pump, the need for a cooling tower is eliminated, thus reducing the factory's water footprint. In this project's model factory, cooling towers are used to cool the water, which reaches 70°C and exits at about 40°C to be directed to the treatment plant. Using the heat pump achieves:

- Reducing the temperature of the water directed to the treatment plant to 25°C, which benefits the subsequent biological treatment.
- Eliminating the need for a cooling tower, as greater cooling is produced by the heat pump. This eliminates the need for makeup water for the tower.
- Producing process steam, which allows for a reduction in extractions from the current cogeneration steam turbine, thus increasing electrical production, which can offset part of the heat pump's electrical consumption.

A schematic of the factory before and after the pump installation is presented in Figure 3.

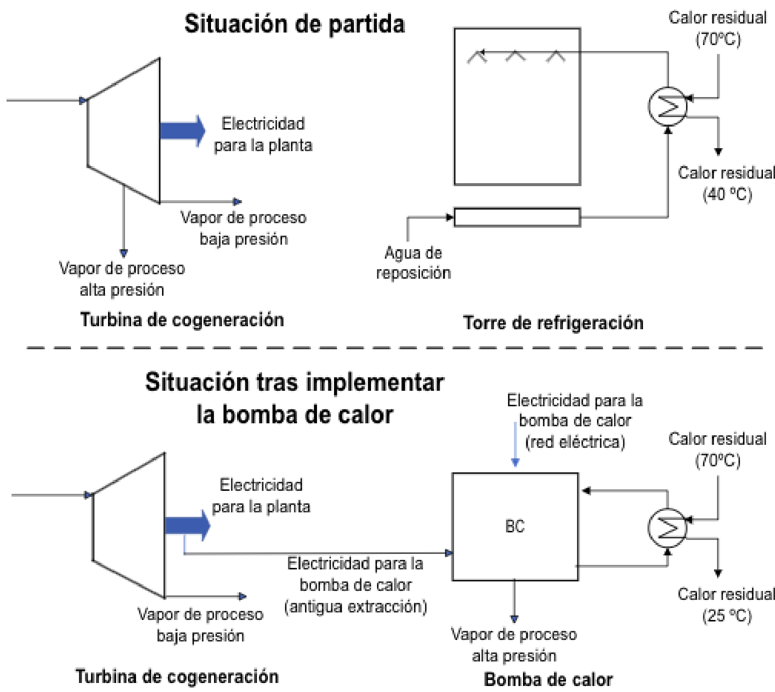


Figure 3 - Before & After (Source: Own Elaboration)

2. State of the Art

Thermal demand currently accounts for 25% of final energy consumption and is divided into three major groups: electricity, transportation, and heat. Thermal energy is an essential resource for modern life, with a significant portion of this energy consumption occurring in industrial processes. Consequently, thermal demand also represents 20% of the total CO_2eq emissions from Scope 1 today, as this resource mainly originates from fossil fuels (Zhisdorf, 2023) (de Boer, 2020).

a. Heat pump technology

To achieve decarbonization in this sector, measures are needed to improve energy efficiency and replace fossil fuels with renewable sources. Among these measures, the installation of heat pumps is notable due to their integration with renewable energy sources. Heat pumps operate on the principle of raising the temperature of a residual heat source to reuse that resource. This measure has the advantage of being installed in existing industrial processes, reducing the residual heat dissipated into the environment while also cutting CO_2eq emissions. If the heat pump were exclusively powered by renewable energy sources, these emissions would be reduced even further.

Industrial heat pumps are a technology that raises the temperature of a waste heat source from a process for the purpose of reusing it. This new use can occur either within or outside the process from which it is recovered. Consequently, a smaller amount of external energy will be required. The advantage of heat pumps is that they are versatile, able to be installed not only in new processes but also in existing ones. To schematically understand the function of a heat pump, the Figure 4 and Figure 5 are presented:

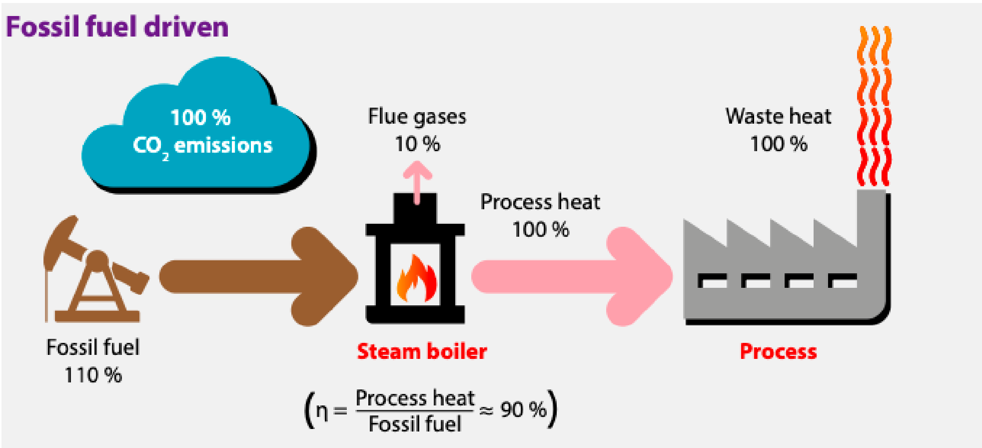


Figure 4 - Fossil Fuel Process (de Boer, 2020)

In the process shown in Figure 4 a total amount of thermal energy of 110% is required, because during the process, up to 10% of heat losses can be emitted to the environment. At the end of the process, the temperature of the waste heat is too low to be reused in other industrial processes, so it is completely dissipated into the environment, resulting in a 100% waste.

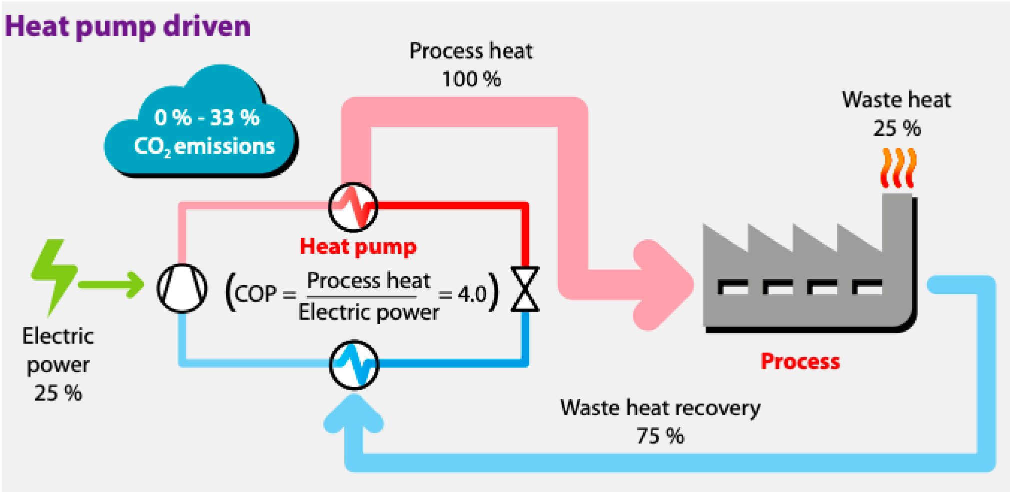


Figure 5 - Heat Pump Process (de Boer, 2020)

In contrast, in the process shown in Figure 5 the waste heat dissipated into the environment is reduced to a quarter (25%), since the rest is reused in the process thanks to the heat pump. The ratio between the thermal energy produced and the electricity used is called COP (Coefficient of Performance) and usually ranges for heat pumps between 2 and 5. The value of this parameter depends on the temperature difference between the process heat and the waste heat, known as the lift of the heat pump. At the same time, CO_2eq emissions are also reduced by about 67% due to the decreased thermal demand thanks to reuse. Moreover, if the electricity comes from renewable sources, this emission reduction could reach up to 100% (de Boer, 2020).

b. Types of Heat Pumps

Heat demand in industrial processes focuses on high-temperature requirements (>100°C); however, this temperature range is not yet fully developed. Initial research on heat pumps operating above 200°C is beginning to emerge. Among these, the inverse Brayton cycle stands out, capable of reaching temperatures up to 600°C with COPs around 2.4 (Linares, 2023).

Every process of installing a high-temperature heat pump requires the selection of an appropriate working fluid. The two fluids considered in this project are N_2 and CO_2, both natural refrigerants with low Global Warming Potential (GWP).

Finally, it is determined that the technology to be used in this project is the Closed Regenerative Inverse Brayton Cycle. To improve the cycle's efficiency, a regenerator will be incorporated, ensuring that the demand temperature (ICU) can be high. A schematic of the heat pump cycle is shown in Figure 6.

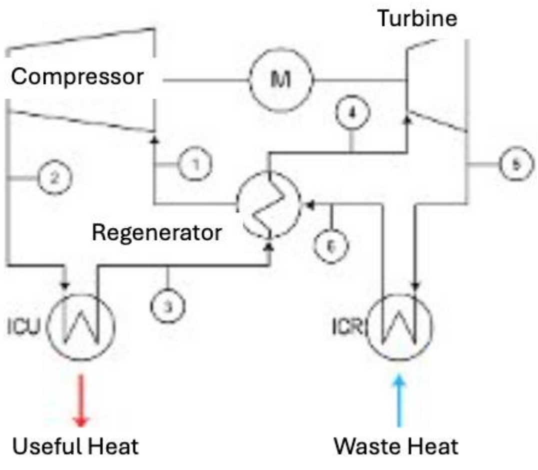


Figure 6 - Conventional Brayton Cycle with Regeneration (Source: Own Elaboration)

3. Objectives

The purpose of this project is to analyze the results of reusing the residual heat dissipated in the cooling towers present in a factory. As a factory prototype, one that uses process steam, produced from cogeneration using back-pressure steam turbines, will be chosen. Thus, the use of the available residual heat will allow:

- Elimination of the existing cooling towers, by using the residual heat to power the heat pump. This removes their water consumption.

- Reduction of steam extractions from the turbine, increasing its electrical production, contributing to the electric consumption of the heat pump.

Based on these results, it will be determined whether this solution is beneficial for the factory and whether it is worth installing. To obtain answers to these statements, provide results, and reach a conclusion, the following will be carried out:

- **Design of the thermodynamic cycle**: It will cool the available residual stream that previously fed the cooling towers, determining electric consumption and steam production. It includes different variants depending on:

- **Architecture:** Depending on the size, shape, and dimensioning of the cycle components (ducts, heat exchangers, and turbomachinery).

- **Fluid:** Selection of the working fluid among the initially proposed variables: $\boldsymbol{N_2}$ and $\boldsymbol{CO_2}$. The COP (Coefficient of Performance) will be the variable determining the type of fluid to be used. Some characteristics of the candidate fluids are:

    - $\boldsymbol{N_2}$: Nitrogen has a critical point (126 K and 34 bar) far from the project's operating range, so it would behave as an ideal gas, suitable for a reverse Brayton cycle. However, its density is much lower than that of CO_2, resulting in bulkier equipment.

    - $\boldsymbol{CO_2}$: It has its critical point at 304 K and 73.77 bar. It is a relatively safe substance compared to other refrigerants. Additionally, it has good heat transfer properties and high density. Although it is not the case in this project, it is a very attractive fluid for transcritical cycles.

- **Operating conditions**: Two steam demand values have been considered to make the study more flexible. Based on this, certain analysis scenarios have been defined. Additionally, the low pressure of the cycle has been established seeking a compromise between specific volume (size) and distance from the critical point (thus avoiding instability problems in turbomachines).

- **Sizing of the main elements of the cycle**: Once the cycle's operating conditions and behavior have been determined, the different elements constituting it have been sized:

- **Ducts**: The pipes through which the selected fluids flow will be sized based on design standards. The most suitable material for each system cycle will also be selected based on their pressure and temperature conditions and the type of fluid passing through them.

- **Heat exchangers**: To capture and reuse the residual heat, it is necessary to size the heat exchangers that will form the heat pump. A total of three heat exchangers will be required: one to capture the residual heat, a regenerator to increase the heat pump cycle efficiency, and a final one to transfer this heat to the complementary oil loop where the steam boiler is located.

- **Turbomachinery:** The cycle contains a turbine and a compressor aligned on the same shaft, so their joint speed, stages, and size will be determined, aiming to maximize efficiency according to usual design criteria.

- **Modeling and preliminary SolidEdge design in the final layout**: Obtaining a more realistic view of the plant and its layout, routing the ducts...

- **Economic feasibility of the proposed solution**: Considering overall costs, establishing an economic balance and an investment cost for the equipment.

4. Materials and Methods

a. High-Temperature Heat Pump Model

This section explains the procedure followed to model the heat pump, as well as the hypotheses used. The scenarios considered will depend on the amount of residual heat available from the factory. Based on this quantity, results will be sought in line with maximum equipment efficiency, measured through the Coefficient of Performance (COP). Parameters such as the thermodynamic properties of the fluid at each point in the cycle (temperatures, enthalpies, entropies, etc.), efficiencies, heat flows, the amount of steam generated, and more will be analyzed to achieve the highest possible system efficiency. Once the model is developed, its parameters will be adjusted to maximize the amount of steam generated from the plant's residual heat flow.

b. Definition of Boundary Conditions, Initial Data, and Model Parameters

The first step is to establish the initial data of the model factory. Based on this data, the heat pump will be dimensioned to obtain equipment that maximizes the target parameters. The plant layout is given in Figure 7, shows the design of the heat pump installation to implement in the model factory.

An oil loop cycle has been added between the heat pump and the steam generation part to improve the cycle performance, considering the large distance between the heat pump (located at the current refrigeration tower site) and the boiler (located next to the steam turbine). This way, heat won't need to be transported over a long distance.

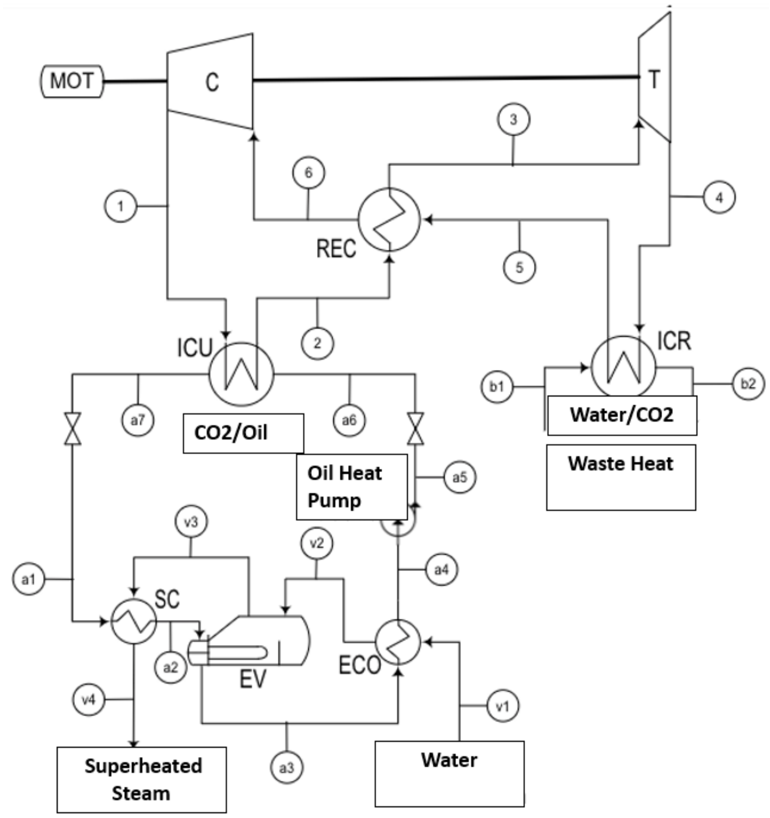


Figure 7 - Final Cycle Design (Source: Own Elaboration)

<u>Characterization of Available Residual Heat</u>

The auxiliary heat source (source) is process cooling water that reaches the heat pump at 70°C and exits at 25°C. Two working flow rates are considered, a small one of 390 m³/h and a large one of 660 m³/h.

Operating Conditions of the Steam Boiler

The water is taken from an external source at 10°C and must produce saturated steam at 12 bar for the smaller cooling water flow rate and 10 bar for the larger cooling water flow rate. This will correspond to steam flows of 45 t/h and 95.6 t/h, respectively. To provide more flexibility to the design, a thermal oil cycle has been interspersed to carry the heat produced by the heat pump to the boiler. The distance between the boiler and the pump has been taken as 400 m.

Pressure Drop in Ducts and Heat Exchangers

For the purposes of solving the thermodynamic cycle, pressure losses in the ducts are neglected (which have been considered for their sizing). A 2% pressure drop will be applied in the heat exchangers (Δp) except for the heat exchangers in the recovery boiler, leaving two pressure levels in the cycle: a high level (between 90 and 100 bar) and a low level (between 40 and 45 bar). This hypothesis is typical when making a preliminary model of a Brayton cycle since the pressure losses are very small compared to the operating pressure. Additionally, except for the connections with the recovery boiler, the distances between the elements of the cycle are short.

Modeling of Turbomachinery

Turbomachinery will be modeled using isentropic efficiency. For the turbine, the method consists of calculating the ideal (isentropic) expansion process, obtaining the fluid properties at the outlet. With these known, they are adjusted with the efficiency to calculate the properties of the real process. For the compressor, the same is done but the process is compression instead of expansion. To obtain efficiencies, the turbomachinery must be sized, which will be explained in section.

Modeling of Heat Exchangers

The size of the heat exchangers is determined by the minimum temperature approach (PP) and the maximum allowable pressure losses. Printed Circuit Heat Exchangers (PCHE) will be assumed to result in a compact installation.

Choice of Working Fluid

Different performance indicators of the cycle (COP, power, mass flows, etc.) will be evaluated depending on the working fluid introduced (CO_2 and nitrogen) in the heat pump. All candidates have advantages and disadvantages, but both share a low Global Warming Potential (GWP) and, consequently, low CO_2eq emissions.

c. *Scenarios to Analyze*

The thermodynamic cycle will be modeled under different scenarios based on the available residual heat and the conditions of the water vapor output from the recovery boiler.

- **Scenario 1:** Characterized by the smaller cooling water flow rate (390 m³/h) and higher pressure steam generation (45 t/h at 12 bar).
- **Scenario 2:** Characterized by the larger cooling water flow rate (660 m³/h) and lower pressure steam generation (95.6 t/h at 10 bar).

Based on the results obtained in both scenarios, a common sizing of the final results will be carried out, always considering the most unfavorable results. This way, the system will always be sized for the worst-case scenario.

Another parameter that has also influenced the definition of the different scenarios is the compressor inlet pressure, which determines the gas density at the inlet and, consequently, the size of the plant. Additionally, the pressure must be chosen so that the cycle always operates in the superheated vapor region, without entering the two-phase dome. Two values have been tested: 15 bar and 40 bar. The first is chosen to facilitate filling, as 20 bar is a common pressure for supplying the working fluid. The 40 bar pressure, on the other hand, significantly reduces the specific volume at the compressor suction.

Finally, a compressor inlet pressure of 40 bar has been decided upon as it is the point at which the fluid density is no longer low and, in addition, it is far from the critical point. Lastly, simulations have been conducted with different working fluids (Nitrogen and CO_2) under the same pressure and temperature conditions, allowing for a solid comparison between them. After verifying that the results with nitrogen lead to excessive sizes, CO_2 is chosen as the working fluid.

c. Performance Equations

To solve the system and obtain the parameters in the most optimal way, thermodynamic equations have been formulated based on the fully dimensioned cycle to solve the system. First, the plant layout and its components are outlined (see Figure 7). Three cycles are distinguished: water (residual heat), CO_2, and oil (Therminol VP1).

Residual heat is transferred to the heat pump cycle through the ICR heat exchanger. The cooling water enters through b1 and leaves the pump through b2. Next, in the CO_2 cycle, this residual heat is upgraded to high temperature in the ICU. Although the turbine produces work, it is not sufficient to cover the compressor's consumption, requiring an additional motor. A regenerator (REC) has also been installed to maximize the heat pump cycle's efficiency and achieve the desired temperatures for steam production. The ICU heat exchanger transfers the heat produced by the pump to the oil cycle. At the end of the oil cycle, a boiler generates steam from incoming feed water. The oil cycle includes a pump to ensure circulation.

<u>Boundary Conditions</u>

Table 1 reflects the established boundary conditions necessary to establish an order and direction among the unknowns.

Data	Value (Scale 1/Scale 2)	Units
Compressor inlet pressure (P[6])	40	bar
Compressor inlet temperature (T[6])	169/158	°C
Compressor efficiency (η_C)	0,88	p.u.
Turbine efficiency (η_T)	0,92	p.u.
Oil pump efficiency (η_{Ba})	0,75	p.u.
Pitch Point (PP) of the heat exchangers	5	°C
Steam cycle outlet temperature (T_v[4])	193/186	°C
Steam cycle outlet pressure (P_v[4])	12/10	bar
Oil pressure at boiler outlet (P_a[4])	5	bar
EA (Difference in steam T in the boiler)	5	°C
Distance between heat pump and boiler (L_a)	400	m
Cooling water outlet temperature (T_b[2])	25	°C
Cooling water outlet temperature (T_b[1])	70	°C
Steam outlet quality (x_v[4])	100	%

Table 1 - Boundary Conditions (Source: Own Elaboration)

Thermodynamic Equations and System Modeling

To achieve the system's optimal operation, thermodynamic equations governing each cycle's behavior are applied. Here are the key equations and principles used:

Heat Transfer in the ICR and ICU Heat Exchangers:

$$Q_{ICR} = \dot{m}_{water} \cdot Cp \cdot (T_{b1} - T_{b2})$$

$$Q_{ICU} = \dot{m}_{CO_2} \cdot (h_{out} - h_{in})$$

Where:

- Q_{ICR} and Q_{ICU} are the heat transferred in the ICR and ICU heat exchangers, respectively.
- $\dot{m}_{water}$ and $\dot{m}_{CO_2}$ are the mass flow rates of water and CO_2, respectively.
- Cp is the specific heat capacity of water.
- T_{b1} and T_{b2} are the inlet and outlet temperatures of the cooling water.
- h_{out} and h_{in} are the enthalpies of CO_2 at the outlet and inlet of the ICU.

Compressor Work and Turbine Work:

$$W_{compressor} = \dot{m}_{CO_2} \cdot (h_{c,out} - h_{c\,,in})$$
$$W_{turbine} = \dot{m}_{CO_2} \cdot (h_{t,in} - h_{t\,,out})$$

where:

- $W_{compressor}$ and $W_{turbine}$ are the work done by the compressor and turbine, respectively.
- $h_{c,out}$ and $h_{c\,,in}$ are the enthalpies of CO_2 at the outlet and inlet of the compressor.
- $h_{t,in}$ and $h_{t\,,out}$ are the enthalpies of CO_2 at the inlet and outlet of the turbine.

Energy Balance for the Regenerator:

$$\mathrm{Q_{REC}} = \dot{m}_{CO_2} * (\mathrm{h_{hot,out}} - \mathrm{h_{hot,in}}) = \dot{m}_{CO_2} * (\mathrm{h_{cold,in}} - \mathrm{h_{cold,out}})$$

where:

- Q_{REC} is the heat exchanged in the regenerator.
- $h_{hot,out}$ and $h_{hot,in}$ are the enthalpies of the hot CO_2 stream at the outlet and inlet.
- $h_{cold,in}$ and $h_{cold,out}$ are the enthalpies of the cold CO_2 stream at the inlet and outlet.

<u>Overall Energy Balance:</u>

The overall energy balance ensures that the total energy input equals the total energy output, considering all heat exchanges, work done, and losses.

These equations are solved iteratively, adjusting parameters to maximize system efficiency and optimize the production of process steam using the available residual heat. Also the efficiency of the compressor, turbine and heat exchanger has been taken into account. The comprehensive model will inform the design and operation of the heat pump system, providing detailed insights into its performance under various conditions and scenarios.

In Table 2 all target parameters to be analyzed in the simulations are summarized.

Parameter	Units
COP	p.u.
Mass flow rate of CO_2 ($\dot{m}_{CO_2}$)	kg/s
Mass flow rate of Therminol VP1 ($\dot{m}_{oil}$)	kg/s
Mass flow rate of generated steam ($\dot{m}_v$)	t/h
Compressor power consumption ($\dot{W}_{compressor}$)	kW
Motor power consumption ($\dot{W}_{motor}$)	kW
Turbine power ($\dot{W}_{turbine}$)	kW

Table 2 - Parameters to Optimize

These parameters are crucial for determining the system's overall performance and efficiency. The analysis will provide insights into how different operating conditions affect the heat pump cycle, enabling the optimization of the system to achieve the best possible performance.

d. Preliminary Design of Cycle Components

In this section, the design methodology for the heat pump components will be detailed to better characterize the most interesting scenarios.

<u>Sizing of Heat Exchangers</u>

Heat exchangers are the elements that recover and transfer heat between the different components of the cycle. In the case of the cycle sized in this project, the present heat exchangers are:

- **REC**: Heat recuperator in the CO_2 cycle
- **ICR**: Heat exchanger between cooling water and CO_2
- **ICU**: Heat exchanger between CO_2 and oil/Therminol VP1

All three heat exchangers are chosen to be Printed Circuit Heat Exchangers (PCHE). This type of heat exchanger is noted for its high compactness and good design for withstanding high differential pressures. The final goal of this sizing is to determine the exchanger's volume (height, width, and length) and its thermal conductance (UA).

Based on the equations previously presented, temperatures and pressures of the four points connecting to each exchanger, as well as the corresponding mass flows, are known.

The model can be initiated with the following assumptions explained in (Serrano Remón, 2014), which allow the exchanger to be simplified to a single hot and a single cold duct, subdivided into a certain number of nodes:

- The mass flow of the hot and cold streams is evenly distributed across all channels of the PCHE.
- Both the cold and hot sides have the same geometry, consisting of semicircular channels running straight through the exchanger. Additionally, both sides have the same number of ducts.
- Temperature is constant in each stream of each node, and its distribution is periodic. This implies it repeats every two rows of channels.

Furthermore, the chosen PCHE exchanger is modular, meaning it has a predefined width and height, specifically 60 cm. Consequently, the height of the exchanger is fixed at 0.6 meters, and its width will be a multiple of 0.6 strictly greater than the previously calculated length. A model of the exchanger is shown in Figure 8.

It is worth noting that these exchangers are typically mounted vertically, so the length in Figure 8 (the path traversed by the streams) externally becomes the height; the height in Figure 8 becomes the front (width) and the width is the depth. A "module" has dimensions of 60 cm x 60 cm x 150 cm, with the largest dimension being the length of the channels, and the modules are connected in parallel.

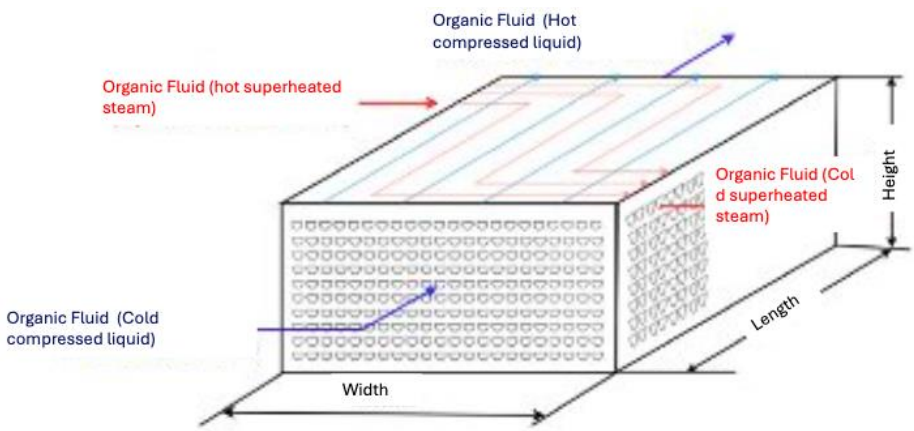


Figure 8 - Heat Exchanger Schema (Albano, 2023)

The solution process follows an iterative approach, where the PCHE is divided into uniform segments called nodes, as shown in Figure 9. From this segmentation and with prior knowledge of the boundary conditions of the heat exchangers and the exchanged heat, the resolution process begins as described by (Serrano Remón, 2014).

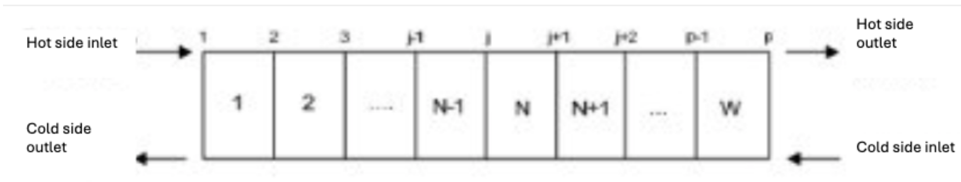


Figure 9 -Heat Exchanger Morphology Schematic (Albano, 2023)

The process to follow is based on the following steps:

1. Determine the number of tubes (n_{tubes}) present in the PCHE and calculate the amount of heat exchanged by each one from the total.
2. The process begins by selecting one side (either hot or cold), considering the pressure and temperature of the fluid at the inlet and outlet. Initial properties are assigned to each node at both ends, which are then averaged to obtain values for density, viscosity, and thermal conductivity.
3. Using the determined properties, the Nusselt number (Nu_N) is calculated in each node using the empirical correlations of Gnielinski (see Table 3). Once the Nusselt number is known, the convection coefficient (h_N) corresponding to each node is determined. It is important to note that there are different values for Nu_N and h_N for the cold side and the hot side. Additionally, since the ducts are semicircular, the characteristic length for calculating the Reynolds number and h_N is the hydraulic diameter (D_h).

$Re_N < 2300$	$2300 < Re_N < 5000$	$Re_N > 5000$
$Nu_N = 4{,}089$	$Nu_N = 4{,}089 + \frac{Nu_{5000} - 4{,}089}{5000 - 2300} \cdot (Re_N - 2300)$	$Nu_N = \frac{f_N \cdot (Re - 1000) \cdot Pr_N}{1 + 12{,}7 \cdot \left(Pr_N^{2/3} - 1\right) \cdot \sqrt{\frac{f_N}{8}}}$

Table 3 - Calculation of the Nusselt number using Gnielinski's correlations. (Serrano Remón, 2014)

$$h_N = \frac{Nu_N \cdot k_N}{D_h}$$

4. Using the heat exchanged at each node and the known convection coefficients, the heat exchange area of the node (A_N) is determined, which in turn will indirectly provide the node length given that the shape of the ducts is known.

$$\dot{Q}_N = U_N \cdot A_N \cdot (T_{cal,N} - T_{frio,N})$$

$T_{cal,N}$ y $T_{frio,N}$ represent the average temperatures of the node, while U_N denotes the overall heat transfer coefficient, which takes into account the present thermal resistances.

$$\frac{1}{U_N} = \frac{1}{h_{cal,N}} + \frac{1}{h_{frio,N}} + \frac{t}{k_{placa}}$$

5. Knowing the length of each node (L_N) the pressure drop in it is calculated.

$$\Delta P_N = f_N \cdot \rho_N \cdot \left(\frac{L_N \cdot v^2}{2 \cdot D_h}\right)$$

After calculating the pressure drop, the pressure at the node outlet is reassessed, and properties at that point are adjusted. If the properties differ by more than 1% from the initial values, the iteration must be repeated with the new properties. If the difference is less than 1%, all pressure drops are accumulated to obtain the total loss, which must be equal to or less than the desired value.

After running the model, the pressure drops on both sides of the exchangers, as well as their length, width, and height, are determined. It is important to highlight that if the resulting number of modules required to size each of the exchangers is greater than 14, two exchangers are imposed so that they do not exceed the dimensions given by the manufacturer.

Once the dimensions of the exchanger are found, its average conductance can be determined according to UA Equation, where r and d are the radius and diameter of the channel and n_{tubes} is the number of tubes of each stream that has been determined from the iterative process. From the average conductance, the logarithmic mean temperature difference (DTLM) can also be determined according to $\dot{Q}$ Equation .

$$UA = [(\pi \cdot r + d) \cdot n_{tubes}] \cdot \sum_{i=1}^{30} U_i \cdot L_i$$

$$\dot{Q} = UA \cdot DTLM$$

Sizing of Conducts

The ducts are the connection that transports the fluid between the different elements that make up the Brayton cycle of this project (heat exchangers and turbomachines). The sizing of the ducts is standardized to facilitate both the manufacturing and the connections of these elements with the rest of the components of the cycle. The standards that will be used to define the sizing of these elements are: Norsok Standard P-001 (Industry, 2006) and ASME B31.1-2007 (Engineers, 2007).

The main criteria for sizing the ducts are:

- The maximum fluid velocity, limiting excess noise and vibrations in the ducts.
- The pressure drop per unit length, a criterion that allows meeting to some extent the hypothesis made in Section 8 of zero pressure drop in the ducts between processes.

The cycle sized in this project, represented in Figure 7 contains three different fluids: CO_2, Therminol VP1, and water carrying the residual heat. Consequently, there will be three types of different ducts: those numbered from 1 to 6, corresponding to CO_2; from a1 to a7, corresponding to Therminol VP1; and b1 and b2, corresponding to water.

For proper sizing, it is first necessary to identify the following properties of all points in the cycle: pressure (P), temperature (T), density (ρ), viscosity (μ), and mass flow ($\dot{m}$).

Secondly, it is necessary to select a suitable material both concerning the fluid passing through it and the pressure and temperature properties present during its journey. For the ducts containing Therminol VP1, the pressure and temperature conditions are not excessively high; thus, following its reference documentation (Eastman, 2019) a carbon steel ASTM A-53B has been selected. The same applies to water, which, being a liquid and due to pressure and temperature conditions, uses the same material. The ducts carrying CO_2 are not subjected to excessively high temperatures and pressures either; however, since CO_2 is highly corrosive, a more resistant material has been chosen: stainless steel ASTM A-213 TP316, as is common in other projects with this fluid (Kruizenga, 2014) (Rochau, 2012) (Konist, 2018).

Sizing begins by defining the maximum velocity in the duct. Its calculation is based on the state of the fluid (gas or liquid). For Therminol VP1 and water (residual heat), liquids, a maximum velocity of 6 m/s is set, taken from Table 4 of the standard (Industry, 2006).

Fluid	Maximum velocities (m/s)			
	CS	SS/Titanium	CuNi [c]	GRP
Liquids	6	[b]	3	6
Liquids with sand [d]	5	7	NA	6
Liquids with large quantities of mud or silt [d]	4	4	NA	NA
Untreated seawater [a]	3	7	3	6
Deoxygenated seawater	6	[b]	3	6

[a] For pipe less than DN 200 (8 in), see BS MA-18 for maximum velocity limitations.
[b] For stainless steels and titanium the maximum velocity is limited by system design (available pressure drop/reaction forces). 7 m/s may be used as a typical starting value for sizing.
[c] Minimum velocity for CuNi is 1,0 m/s.
[d] Minimum velocity for liquids with sand should be in accordance with ISO 13703.

Table 4 - Maximum Velocity in Liquid Carrying Ducts (Industry, 2006)

In the case of CO_2, being a gas, the maximum velocity is determined by the following equation:

$$v_{max} = \min\left(175\left(\frac{1}{\rho}\right)^{0,43};60\right)$$

In the previous equation, ρ is the density. Next, the minimum diameter required due to the conditions of maximum allowable velocity is calculated:

$$d_{min}(mm) = \sqrt{\frac{4 \cdot \dot{m}}{\pi \cdot \rho \cdot v_{max} \cdot n_{tubes}}} \cdot 1000$$

Based on this diameter measurement, the strictly larger standardized diameter than the minimum interior diameter has been chosen, with its corresponding outer diameter defined by the ASME B31.1-2007 standard (Engineers, 2007). This standard also defines the standardized wall thicknesses of the ducts. Additionally, an effort has been made to minimize the number of tubes (n_{tubes}) as much as possible.

The yield strength of the material also needed to be determined, which is necessary for calculating the minimum thickness (t (mm)) defined by equation:

$$t_{min} = \frac{P \cdot D_{ext}}{2 \cdot (\sigma_e + P \cdot y)}$$

Where P is the fluid pressure, D_{ext} is the outer diameter, σ_e is the yield strength of the material, and y is a coefficient that depends on the temperature, material, and diameter as specified in Table 5 (Engineers, 2007).

Table 104.1.2(A) Values of *y*

Temperature, °F	900 and Below	950	1,000	1,050	1,100	1,150	1,200	1,250 and Above
Temperature, °C	**482 and Below**	**510**	**538**	**566**	**593**	**621**	**649**	**677 and Above**
Ferritic steels	0.4	0.5	0.7	0.7	0.7	0.7	0.7	0.7
Austenitic steels	0.4	0.4	0.4	0.4	0.5	0.7	0.7	0.7
Nickel alloys UNS Nos. N06617, N08800, N08810, N08825	0.4	0.4	0.4	0.4	0.4	0.4	0.5	0.7

GENERAL NOTES:

(a) The value of *y* may be interpolated between the 50°F (27.8°C) values shown in the Table. For cast iron and nonferrous materials, *y* equals 0.

(b) For pipe with a D_o/t_m ratio less than 6, the value of *y* for ferritic and austenitic steels designed for temperatures of 900°F (480°C) and below shall be taken as:

$$y = \frac{d}{d + D_o}$$

Table 5 - Values of y (Engineers, 2007)

In this project, it happens that, for all cases, y = 0.4 is applied.

Next, once the standardized thickness (always greater than the minimum thickness calculated using Equation t_{min} has been chosen, the inner diameter of the duct is calculated, as this will be its exact circulation area. The following equation is used:

$$d = D_{ext} - 2 \cdot t$$

Finally, the fluid velocity is calculated and checked to ensure it does not exceed the maximum velocity previously established based on its state (gas/liquid):

$$v_{fluid} = \frac{4 \cdot \dot{m}}{n_{tubes} \cdot \rho \cdot \pi \cdot d^2}$$

Another criterion that must always be met is: $(\frac{D_{ext}}{t} > 6)$. In all cases, this inequality is greatly exceeded.

In the case where the calculated velocity exceeds the maximum velocity, the strictly larger standardized diameter along with its corresponding thickness and outer diameter must be chosen, and the velocity calculation should be re-evaluated through an iterative process until the calculated velocity is less than the maximum.

Once the ducts are sized based on the maximum velocity criterion, the sizing is checked according to the pressure drop per unit length ($\frac{h_f}{L}$). To do this, first, it is verified that the flow is turbulent using the Reynolds number:

$$Re = \frac{\rho \cdot v_{fluido} \cdot d}{\mu}$$

Since in all cases Re > 4000, it is verified that the flow is always turbulent. Consequently, the pressure drop per unit length is calculated using the following equation:

$$\frac{h_f}{L} = f \cdot \frac{v^2}{2 \cdot g \cdot d}$$

The friction factor, f, is calculated using the Colebrook equation, which specifies for turbulent flow. Where $\in$ is the roughness, whose value is specified by the standard (Industry, 2006), and is 0.05 mm for carbon steel and stainless steel pipes.

For liquids, the commonly used pressure drop value is 0.9 bar/100 m.

If the pressure drop exceeds the recommended value, the pipe would be resized with a larger standardized diameter (along with its corresponding outer diameter and thickness). Conversely, if the pressure drop is within the acceptable range, the possibility of reducing the pipe diameter could be considered, provided that the maximum velocity criterion continues to be respected.

Sizing of Turbomachines

For a preliminary sizing of the turbomachines, the following variables will be calculated:

- Rotational speed
- Stages
- Size
- Type (radial, axial, or mixed)
- Maximum efficiency

The procedure used is the Baljé method. This method is based on the premise that the efficiency of turbomachines is defined by 4 dimensionless numbers (Vellini).

$$\eta_{TM} = f(\phi; \psi; Re; Ma)$$

Each of the dimensionless numbers is related to a variable: ϕ is related to the flow rate, ψ to power transfer, Re is the Reynolds number which relates variables such as the viscosity and density of the fluid, and finally, Ma is the Mach number which is related to the compressibility of the fluid (Vellini).

This equation can be simplified in the case where the flow is highly turbulent, which occurs when $Re > 10^6$ and in the case where the fluid is incompressible, indicated by a $Ma < 0.49$. If both conditions are met, the efficiency of the turbomachines can be expressed as follows:

$$\eta_{TM} = f(\phi; \psi)$$

These are two typical assumptions in the sizing of turbomachines. In the case of this project, the flow is turbulent and is considered compressible; therefore, the assumption of $Ma < 0.49$ is not met.

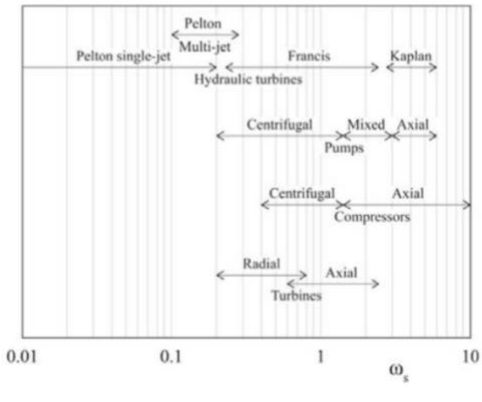
- Specific speed (ω_s): is a parameter indicative of the shape, not the size, of the turbomachine. In our case, (ω_s) turns out to be a function of two parameters: the rotational speed and the number of stages.

  $$\omega_s = f(n; z)$$

  The stages of a turbomachine, also known as cascades, correspond to the insertion of one or more impellers (depending on the number of stages) inside the turbomachine to reduce the large pressure or energy drop between the inlet and outlet. Consequently, a specific speed will be defined for each stage of a turbomachine. This occurs because the fluid is compressible (in the case of an incompressible fluid, the specific speed and flow rate would be the same in all stages). Due to this physical fraction, the isentropic enthalpy drop will be divided by the number of stages when calculating its corresponding specific speed.

  The objective of this project is to choose a design that minimizes, as much as possible, both the speed in rpm and the number of stages.

  The possible geometries are: axial, radial, or mixed. The ranges that optimize performance based on the machine's shape are shown in Figure 10.



Turbomachines	Type	Specific speed
Pumps	Radial	0.5÷1.3
	Mixed flow	1.9÷3.0
	Axial	4.0÷5.0
Hydraulic turbines	Pelton	0.04÷0.1
	Francis	0.5÷2.0
	Kaplan	2.5÷3.5
Compressors	Radial	0.4÷1.0
	Mixed flow	1.0÷2.0
	Axial	1.5÷2.5
Turbines	Radial	0.4÷0.8
	Axial	0.6÷1.2

Figure 10 - Ranges of ws (Vellini)

The ranges and specific speeds for sizing a turbomachine based on its stages (z) and rotational speed (n) can be represented in a graphical model such as Figure 11.

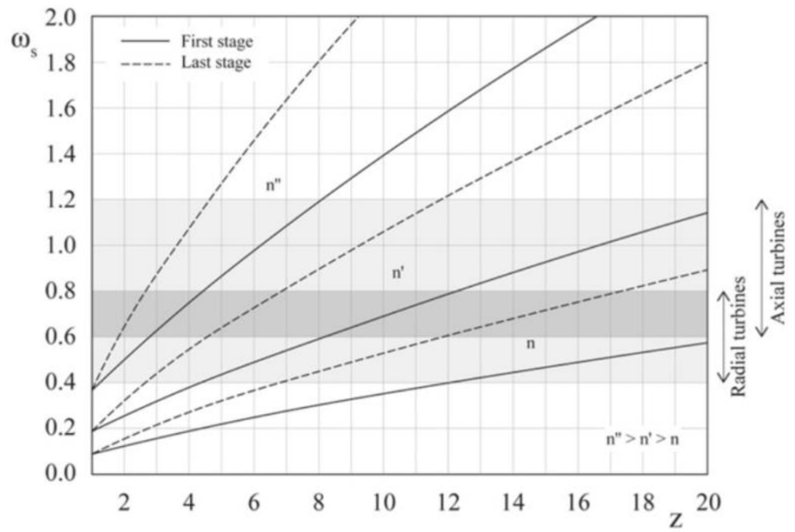


Figure 11 – Basic diagram for selecting the turbine geometry based on n and z (Vellini)

- Specific diameter (D_S): This variable sizes the turbomachine without considering the rotational speed. This variable will be calculated using ω_s and the Baljé diagrams. These diagrams are shown in the following figures:

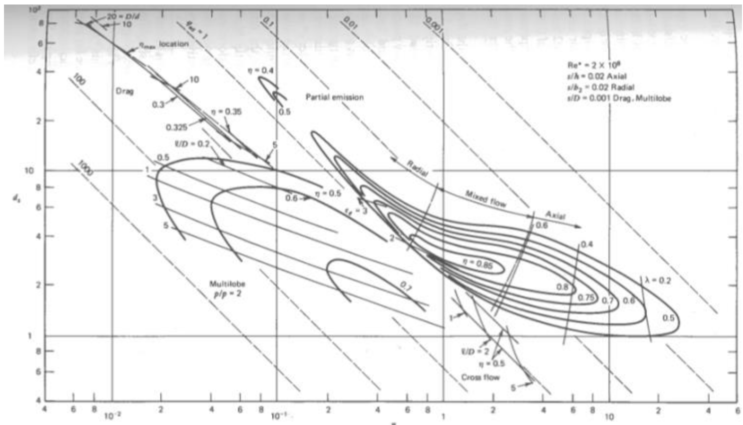


Figure 12 -Baljè diagrams for turbines (Vellini)

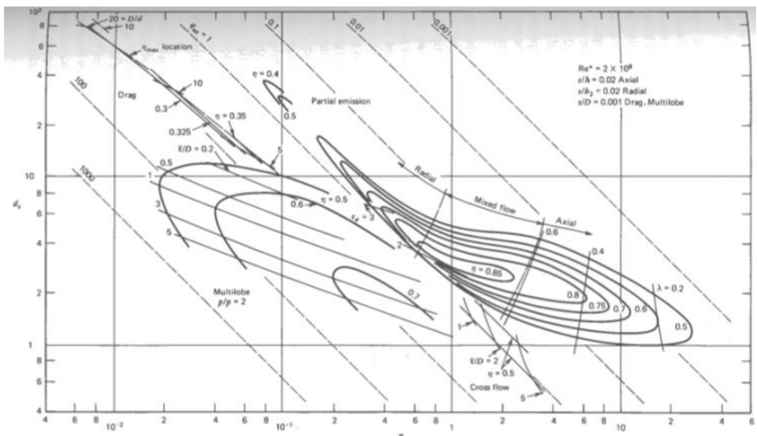


Figure 13 -Baljè diagrams for compressors (Vellini)

e. 3D model of the high-temperature heat pump

The 3D model of the high-temperature heat pump will be created in SolidEdge, based on the previously sized results that define a model capable of supporting all operating conditions. Due to its modeling in SolidEdge, the arrangement of the heat exchangers and the future space they will occupy can be visualized in more detail. It is important to note that the modeling represents the preliminary design of the project and that during the construction phase, this model may be subject to changes.

f. Economic Viability

In this section, the methodology applied for calculating the project costs, the cycle components (investment), considering also the electricity tariff (expenses), and the potential steam savings due to the suppression of one of the turbine extractions, which allows for more electricity production (income), will be explained.

Various time distributions between these flows will be proposed in order to graphically represent the behavior of each cost variable based on the amount of residual heat provided.

Depending on this flow rate, the useful heats produced by the heat pump for the generation of 34.37 MWt steam for scenario 1 and 72.7 MWt for scenario 2 are obtained. In all cases, a total annual operating time of 8000 hours is assumed. With all the above, the values of useful heat (MWh) (Scenario 1: 274160 and Scenario 2: 2581600) and the amount of steam produced per year (t) (Scenario 1: 360400 and Scenario 2: 764800) can be calculated.

Inversion Calculation

The investment has been calculated for the scenario with the highest steam flow (scenario 2; 95.6 t/h), since the equipment must be able to satisfy it. Due to the different nature of the cycle equipment, the costs of each are calculated individually (PEC).

Once the PEC (Project Equipment Cost) and ONSC (Operating and Non-Stock Cost) of each cycle component have been determined, the total investment is calculated by summing the direct and indirect costs. The direct costs are equal to the ONSC, as costs associated with land and civil works are not considered, and the indirect costs are assumed to be 25% of the direct costs. Subsequently, the direct and indirect costs are added to obtain the FCI (Fixed Capital Investment) or total investment for each cycle component. Finally, the total investment will be the sum of the FCIs of the components. It is worth noting that dollars ($) and euros (€) will be used interchangeably due to the current conversion rate being nearly 1:1 and the difference being masked by the inherent uncertainty of the equations of (Weiland, 2019).

Final Economic Balance

Based on the total investment values, the economic analysis is conducted by calculating the levelized costs in two versions: LCOH and LCOS. These indicators differ in their units, with LCOH being divided by the useful heat (in €/MWh) and LCOS being divided by the steam (in €/t).

The LCOS (Levelized Cost of Steam) is a measure of the average net cost of steam generation over the lifetime of the facility. It is determined by summing the investments (installation and equipment costs) along with the expenses (variable costs). For this project, the expenses include operational and maintenance costs, the electricity consumed for the process, and a negative cost (savings) due to the elimination of steam turbine extraction. LCOH (Levelized Cost of Heat) is defined as the normalized cost of heat and is a measure of the average net cost of producing useful heat over the system's lifetime.

$$LCOS\ \left(\frac{€}{\mathrm{MWh}}\right) = \mathrm{CAPEX} + \mathrm{OPEX_{OM}} + \mathrm{OPEX_{elec}} + \mathrm{OPEX_{savings}}$$

It is equally applied for the LCOH but with its respective measures.

Each of the costs is calculated as follows:

- **CAPEX (Capital Expenditure):** Also known as capital expenses, these are related to the initial investment costs of the project, such as acquisition or construction. They are intended to be offset over the long term. In this case, they include everything related to the purchase of equipment for installing the heat pump presented. It is calculated using the following equation, where P refers to the useful heat or steam produced in a year.

$$CAPEX\ (\frac{€}{P}) = INV_{total}\ (€) * f_a/P$$

- **OPEX (Operational Expenditure):** These are the costs associated with maintenance and the cost of electricity used.

  - $\boldsymbol{OPEX_{OM}}$**:** Operation and maintenance costs. These are necessary to ensure the heat pump operates correctly throughout its lifespan. They include maintenance operations, all consumables used (oils, screws, tools, etc.), and the salaries of the involved operators. Overall, this represents an estimated cost of 1.5% of the project investment.

$$OPEX_{OM}\ \left(\frac{€}{P}\right) = INV_{total}\ \left(\frac{€}{P}\right) * 0{,}015 * f_a * f_{\Sigma_{OM}}$$

  - $\boldsymbol{OPEX_{elec}}$ The variable cost associated with electricity consumption in the plant. A mean electricity tariff of $Te = 75€\ /\ MWh$ has been considered.

$$OPEX_{elec}\ \left(\frac{€}{P}\right) = \left(Heat\ Pump\ Consumption\ (MWe) + \frac{Oil\ Pump\ Consumption}{1000}\right) * h * T_e * f_a * f_{\Sigma_{elec}}/P$$

  - $\boldsymbol{OPEX_{savings}}$**:** The variable cost associated with the savings from reducing steam consumption by eliminating the turbine extraction (with a maximum flow of 78 t/h), which will be a negative value due to it being a saving. It is important to note that the turbine work value has been estimated based on commercial information (124 kWh/ton). The result is obtained through two operations. First, the savings are calculated:

$$\mathrm{savings}\ (€) = -\mathrm{T_e}\left(\frac{€}{\mathrm{MWh}}\right) \cdot \frac{124}{1000}\left(\frac{\mathrm{kWh}}{\mathrm{t}}\right) \cdot \dot{\mathrm{m}}_{\mathrm{turbine_{out}}}\left(\frac{\mathrm{t}}{\mathrm{h}}\right) \cdot 8000$$

Where: $\dot{\mathrm{m}}_{\mathrm{turbine_{out}}}\left(\frac{\mathrm{t}}{\mathrm{h}}\right) = \min\{\dot{m}_v, 78\}$

Secondly, the savings can be accounted for based on the electricity tariff:

$$OPEX_{savings}\ \left(\frac{€}{MWh}\right) = savings * f_a * f_{\Sigma_{elec}}/P$$

5. Results and Discussion

In this section, the results common to all scenarios obtained from the application of the methodology explained earlier are presented and discussed.

Next, the economic evaluation results for the installation will be presented. Finally, an estimation of the footprint of the heat pump installation and a 3D visualization will be provided.

a. Results Common to All Scenarios

The working fluid (CO_2) has been a common parameter across all scenarios considered.

To determine the final working fluid, two main candidates were initially considered due to their low global warming potential: CO_2 and nitrogen. Ultimately, after obtaining similar results for both in terms of analyzed parameters (COP, work outputs, mass flows, etc.), nitrogen was discarded due to excessively large equipment size. Consequently, CO_2 was chosen as the final working fluid.

Once the working fluid was selected, the remaining parameters required to define the model—such as pressures, temperatures—could be determined, providing a broader understanding of the heat pump's behavior.

The choice of working fluid was crucial in determining the material for the ducts. Without a prior definition of the working fluid, sizing the ducts would not have been feasible.

As shown in Figure 7 depicting the global cycle scheme, three different fluid cycles can be distinguished, each assigned its corresponding material:

- Heat Pump (CO_2): Stainless Steel A-213 316
- Oil (Therminol VP1): Carbon Steel A-53 B
- Waste Heat (cooling water): Carbon Steel A-53 B

These materials are consistent across all considered scenarios.

The definition of turbomachinery is the final parameter common to all scenarios. Both the type and staging number of turbomachinery coincide. To achieve optimal performance, it is important to ensure that the specific speed falls within one of the ranges stablished by the sellers. For both the compressor and the turbine, it is determined that the minimum speed is 6000 rpm and the minimum number of stages is 2. Thus, the compressors would be radial and the turbines axial.

b. Common Sizing

Once the various elements of the cycle previously explained in the Materials and Methods Chapter have been calculated, the final plant layout is now presented (Figure 14). For each element, both scenarios have been considered, always choosing the final result based on the more unfavorable scenario. This means that for ducts, the number of ducts, diameters, etc., have been taken from the scenario with the higher values. The same applies to turbomachinery and heat exchangers.

- Final Ducts and Heat Exchangers: Figure 14 shows the final number of ducts and heat exchangers per element. A more detailed view will be available in the 3D diagrams. However, it is important to note that in the 3D version, certain heat exchangers have been divided for practical reasons related to modular sizing.

- Final Turbomachinery: like the ducts, the most unfavorable value has been chosen for both the compressor and turbine in the cycle. Once the stages, shapes, and speeds of both turbomachinery and both scenarios have been obtained, the optimal specific diameter is calculated based on the identification of maximum efficiency. Since this variable must be calculated using the most restrictive w_s (lower), the chosen one is shown in the following comparison:

- Weakest w_s for the turbine: Scenario 1. w_{s1}= 0,64 and w_{s2}= 0,75
- Weakest $w_s$for the compressor: Scenario 1. w_{s1} = 0,60 and w_{s2}= 0,50

Ultimately, the relationship between efficiency and optimal diameter is obtained using Cordier's graphs. In both cases, a maximum efficiency of 0.8 is achieved, and the corresponding D_s values are identified. Their final diameters are coherent with the total size of the installation and the heat pump:

- Compressor: Diameter 0,78 and 0,74 meters for respective stages 1 and 2
- Turbine: Diameter 0,37 and 0,42 for respective stages 1 and 2

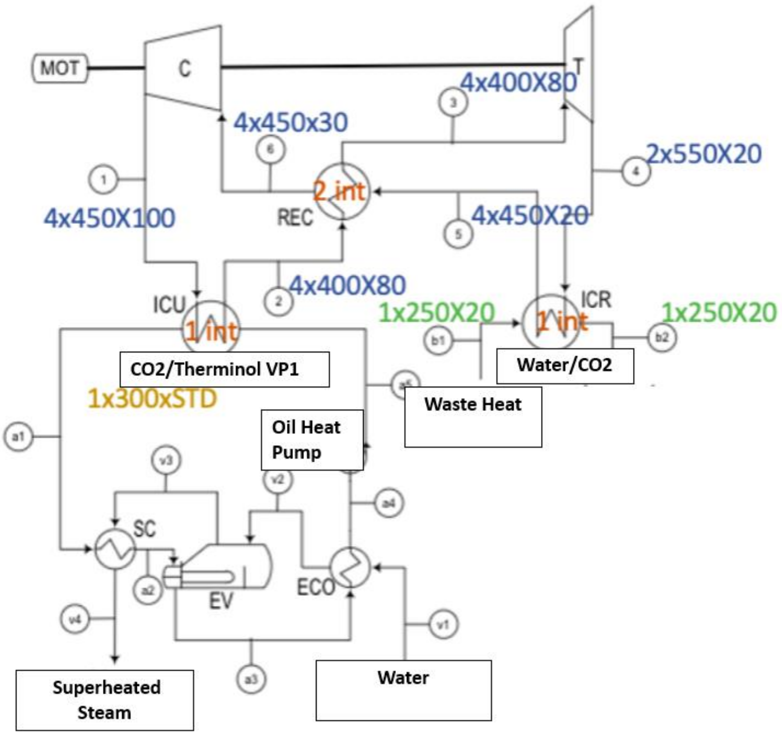


Figure 14 - Final Pipes and Number of Heat Exchangers. The first digit represents the number of pipes, the second the nominal diameter, and the third the schedule. (Source: Own Elaboration)

c. *3D Visualization*

The plan of the sized plant in its extended and compressed version, modeled in SolidEdge, is shown below.

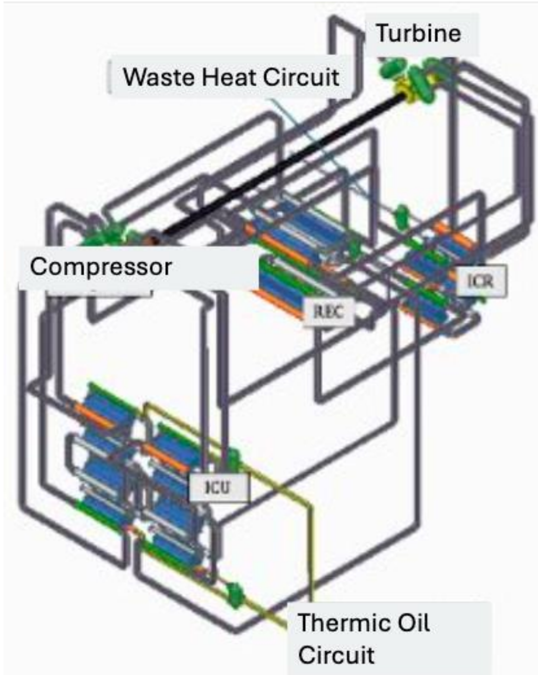


Figure 15 - Extended 3D view (Source: Own Elaboration)

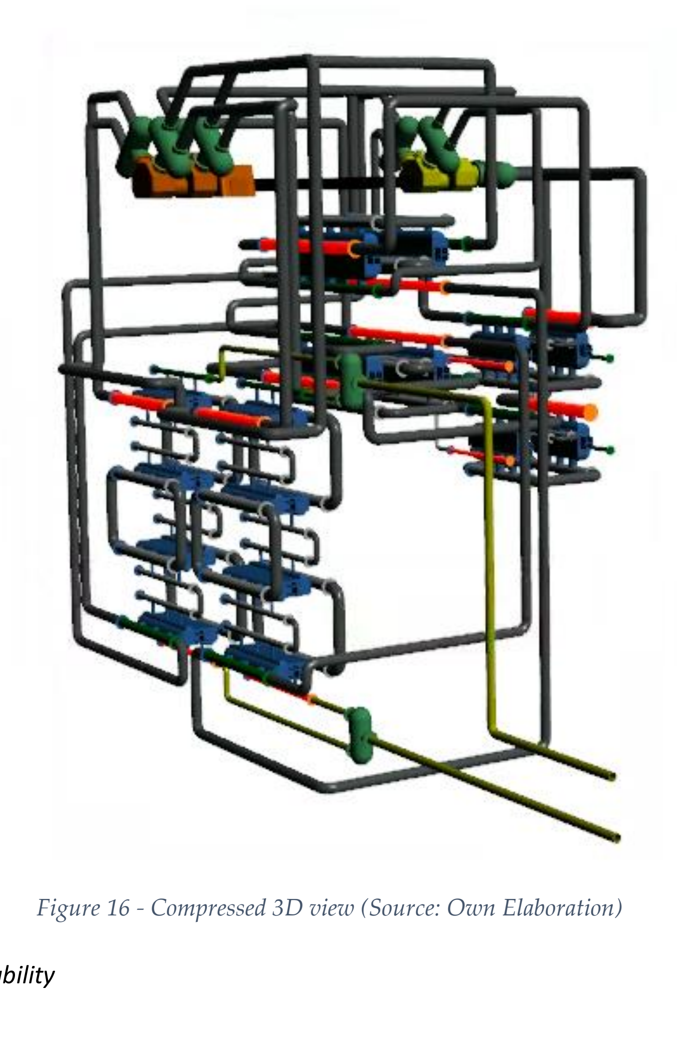


Figure 16 - Compressed 3D view (Source: Own Elaboration)

d. Economic Viability

Total Inversion

By performing the individual calculations for each piece of equipment, the following results are obtained for the equipment investment and the total investment:

Exchanger	UA [W/(m^{2}K)]	CAPEX [€]	OPEX [€]	FCI [€]
ICU	4,341,593	5,030,473	6,539,614	8,174,518
ICR	2,207,637	3,020,128	3,926,166	4,907,708
REC	5,087,890	5,669,939	7,370,920	9,213,651
Total HXs		13,720,539	17,836,701	22,295,876

Table 6 - Exchanger Inversion

On the other hand, Table shows the investment in rotating equipment. 58% of the investment is concentrated in the compressor, 32.5% in the motor, and only 9.5% in the turbine.

	Power [MW]	CAPEX [€]	OPEX [€]	FCI [€]
Compressor	57.851	6,214,692	8,079,100	10,098,875
Turbine	22.067	1,020,367	1,326,477	1,658,096
Motor	35.784	3,493,448	4,541,482	5,676,853
Total		10,728,507	13,947,059	17,433,824

Table 7 - Investment Required for Rotating Equipment

Regarding the steam generator, the values are:

- PEC: 1,861,625€
- ONSC: 3,723,250€
- FCI: 4,654,062€

Summing these items results in a total investment (FCI) of €44,383,762, which, in relation to the nominal useful heat, amounts to €611/kW. This is slightly higher compared to the range (€300 to €500/kW) provided by the IEA (Zhisdorf, 2023) for a comparable MAN heat pump.

LCOS and LCOH results

As represented in Figure 17 and Figure 18. It is important to highlight that the economic balance and the results for LCOH and LCOS, which evaluate the real operational cost of the factory, show a normalized cost between 51 €/MWh and 62 €/MWh (38.8 €/t and 47.2 €/t). The operational cost (total OPEX) ranges between 43.5 €/MWh and 46 € /MWh (33 €/t and 35 €/t). Additionally, it has been proven that the most significant costs are those related to $OPEX_{elec}$.

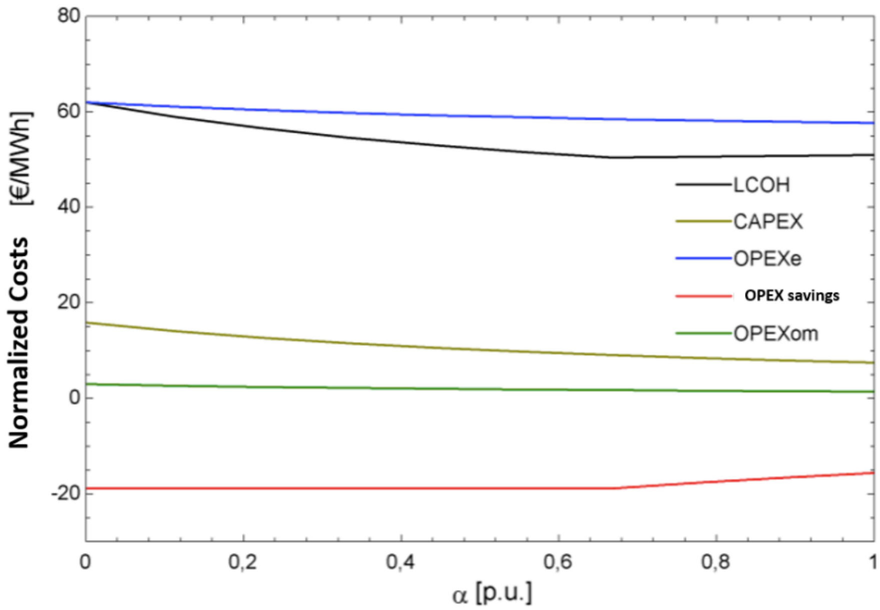


Figure 17 - Normalized Cost (LCOH) according to the fraction of time operating at maximum steam flow. (Source: Own Elaboration)

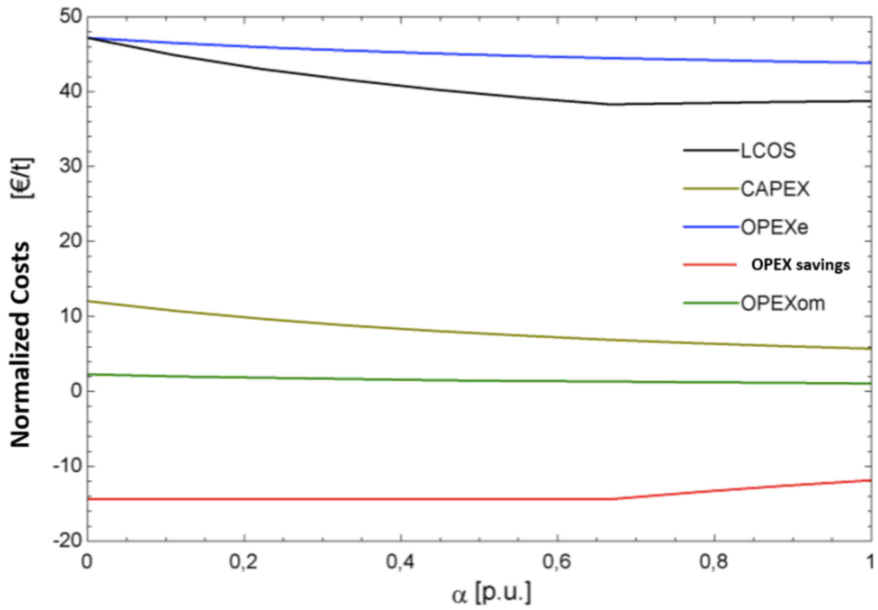


Figure 18 -Normalized Cost (LCOH) according to the fraction of time operating at maximum steam flow. (Source: Own Elaboration)

Figure 17 and Figure 18 show how the saturation affects the savings, with verification that for values of α greater than 0.64, the heat pump produces more vapor than that extracted from the turbine, with the excess needing to be allocated to other uses not considered in the cost. The normalized cost ranges from €62/MWh to €51/MWh (€47.2/t to €38.8/t), while the operational cost (total OPEX) ranges from €46/MWh to €43.5/MWh (€35/t to €33/t). These costs are competitive compared to ETES (Electro-Thermal Energy Storage) systems, which are established for Spain at €75/MWh in 2023, with a target of €63/MWh in 2030 (including CAPEX and OPEX) (Systemiq, 2023).

Additionally, it is noted that the most significant costs are those related to $OPEX_{elec}$, so its sensitivity was also studied by varying the electricity tariff price, as shown in Figure 19 and Figure 20.

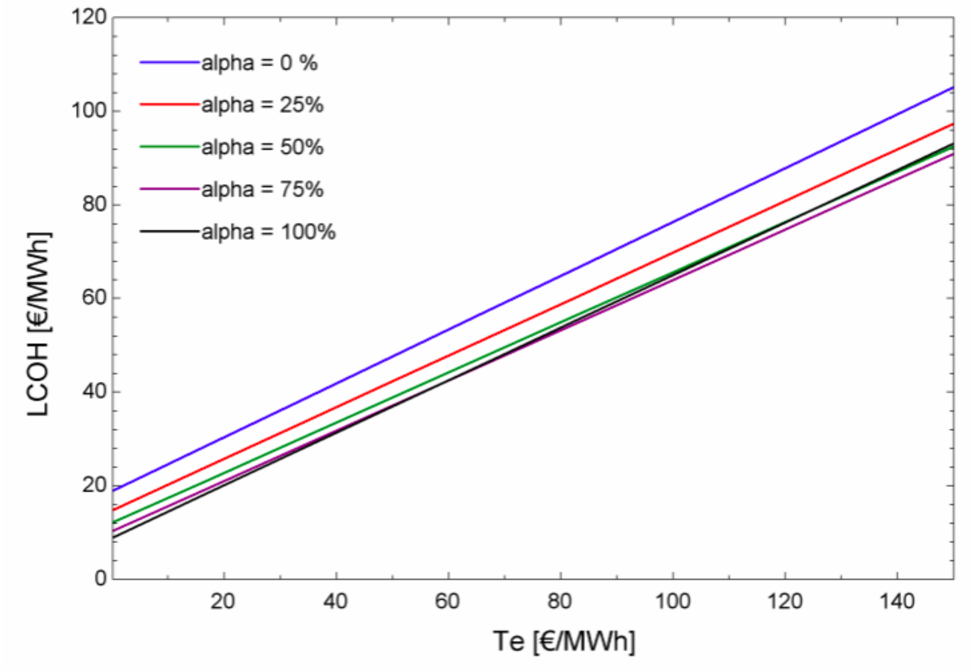


Figure 19 - *LCOH according to the variation in the electricity tariff (Source: Own Elaboration)*

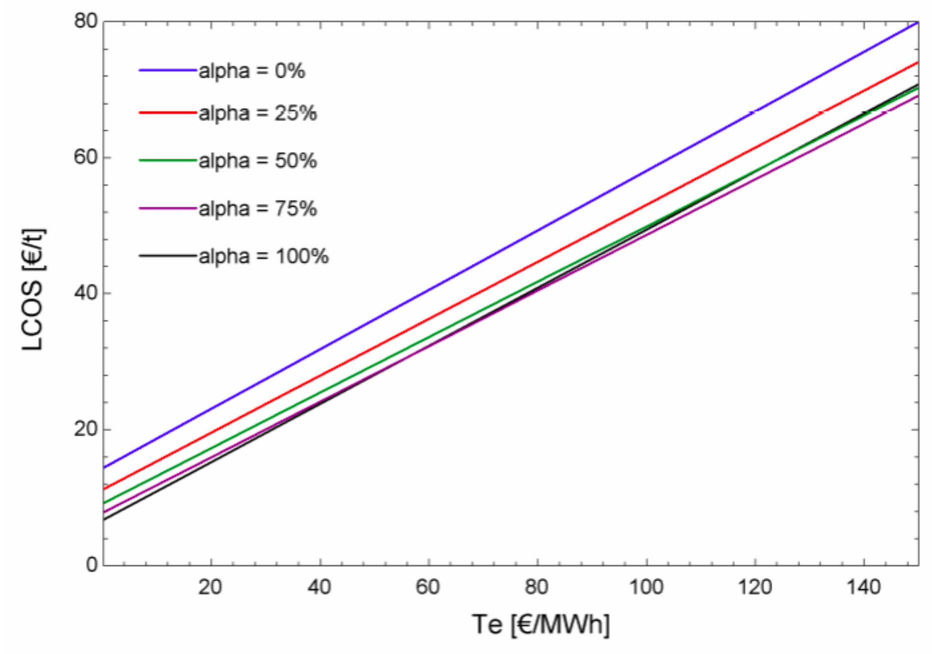


Figure 20 - *LCOH according to the variation in the electricity tariff (Source: Own Elaboration)*

It is observed that the lines for α=75% and α=100% converge at high tariff values. This is because, as previously mentioned, for α values above 64%, the usable steam flow in the turbine becomes saturated, causing the normalized cost to plateau. The operational expenditure (OPEX) then becomes predominant, with its impact increasing as the electricity tariff rises.

6. Conclusions

In this project, the utilization of residual heat from the cooling towers of an industrial process was studied. To fulfill this study, a heat pump was sized using a reverse Brayton cycle. This solution allows:

- Cooling the hot water stream that reached the towers, eliminating them, resulting in water replenishment savings.
- Achieving a lower hot water outlet temperature than with the towers, facilitating the subsequent treatment of this water in a treatment plant before discharging it externally.
- Using the heat recovered from the hot water to produce process steam, eliminating an extraction from the cogeneration turbine. This allows the turbine to produce more electricity, partially compensating for the heat pump's consumption.

The project analyzed two main scenarios based on the following variables:

- Available residual heat.
- Conditions of the generated steam.

These variables result in two final scenarios. Scenario 1 is characterized by a lower cooling water flow rate (390 m³/h) and a goal of generating higher pressure steam (45 t/h at 12 bar). Scenario 2 is characterized by a higher cooling water flow rate (660 m³/h) and generating steam at a lower pressure (95,6 t/h at 10 bar).

In Scenario 1, a COP of 1,89 and a mass flow of CO_2 of 333,7 kg/s are highlighted. Although the COP is not very high, it is noted that the industry of high-temperature heat generation through heat pumps is not yet well developed.

Conversely, in Scenario 2, the COP is higher (2,03) though still relatively low, while the mass flow of CO_2 is 590,8 kg/s.

Regarding the heat exchangers, the aim is to choose the dimensions of those that are larger between the two scenarios. For the three exchangers presented in the cycle: ICU, ICR, and REC, the results show that in all cases, the exchangers in Scenario 2 are larger. Therefore, these exchangers have been selected for the final sizing of the common cycle to cover the most unfavorable case.

The same applies to the duct sizing; since Scenario 2 requires a higher CO_2 flow, the cycle is sized with the number of ducts and the minimum diameter required by this scenario. The final heat exchangers and duct measurements can be seen again in Figure 14.

For the turbomachinery, the most unfavorable case belongs to Scenario 1, as it presents the w_s that best fits the range. Based on this value, the dimensions of both the compressor and the turbine are determined. The dimensions of both are consistent compared to other heat pump projects.

The total estimated volume of the plant according to the 3D design is approximately 10.367 m³ (21 m x 21 m x 25 m). The plant has been minimized considering a vertical arrangement of the heat exchangers.

Finally, regarding the economic feasibility, a total investment (FCI) of €44.383.762 is required for the installation of the heat pump, which, relative to the nominal useful heat, amounts to €611/kW, slightly higher compared to the range (€300 to €500/kW) given by the IEA for a comparable MAN heat pump.

The values of LCOH and LCOS have been studied based on the fraction of time that operates at maximum steam production (α). For α values greater than 0.64, the heat pump produces more steam than the turbine extraction, normalizing the cost between €51/MWh and €62/MWh (€38,8/t and €47,2/t), while the operating cost (total OPEX) ranges between €43,5/MWh and €46/MWh (€33/t and €35/t). These costs are competitive against ETES systems, established for Spain at €75/MWh in 2023, with a target of €63/MWh by 2030 (including CAPEX and OPEX).

During the study, it is demonstrated that both costs are mainly determined by the OPEXe variable, referring to the variable cost of electricity. As this variable is the most determining, several studies have been conducted by varying the electricity rate.

a. Future Developments

The proposed heat pump allows replacing cooling towers to cool the process cooling water, even below the temperature achieved by the towers. Additionally, this removed heat is reused to produce process steam, enabling the closure of the cogeneration steam turbine extraction, so the additional electricity produced reduces the heat pump's electric consumption.

Another possible application of the pump would be to be part of the electrical grid services that the industry can provide as a flexible electricity demand. For this, the installation of a thermal storage system in the thermal oil loop and an ammonia/water absorption machine would be necessary. The operation would be as follows:

- Increased electricity demand: When there are excesses in the grid, the pump can increase its demand by maintaining the turbine extraction to produce steam, so the heat transferred by the ICU to the oil is stored in the thermal tank.

- Reduced electricity demand: During peak moments in the electricity market, the heat pump can stop, opening the turbine extraction to produce the steam required by the plant. The electricity produced by the turbine is self-consumed (allowing modulation). The cold demand for the effluent is met by operating the absorption machine with the stored heat. Being an ammonia/water machine, it does not require a cooling tower.

8. Patents

I would like to thank my directors, José Ignacio Linares and Eva Arenas Pinilla, for dedicating all the necessary time to the completion of the project and for teaching me with great detail and dedication in the areas of their expertise. Special thanks to José Ignacio for always being willing to support my learning throughout my academic journey at the university.

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References

1. Albano, L. S. (2023). *Recuperación de calores residuales en una cementera mediante ciclo de Rankine orgánico (ORC).* Madrid: Trabajo fin de máster, Universidad de Madrid.
2. Barbero, R. a. (2023). HIGH-TEMPERATURE HEAT PUMPS FOR DECARBONIZATION IN INDUSTRY 4.0. *UNED*.
3. Comillas, U. P. (Noviembre de 2023). *Cátedra Fundación Repsol de Transición Energética*. Obtenido de Cátedra Fundación Repsol de Transición Energética: https://www.comillas.edu/catedras-de-investigacion/catedra-fundacion-repsol-de-transicion-energetica/
4. de Boer, R. a. (2020). *Strengthening Industrial Heat Pump Innovation: Decarbonizing Industrial Heat Category Report.* Norway: SINTEF Energi AS.
5. Eastman. (2019). *Systems Design Data Guide, Therminol Heat Transfer Fluids .* Eastman Corporate Headquarters.
6. Engineers, T. A. (2007). *ASME B31.1-2007.* America: ASME Code for Pressure Piping, B31.
7. Hannah Ritchie, P. R. (June de 2020). *Greenhouse gas emissions*. Obtenido de Greenhouse gas emissions: https://ourworldindata.org/greenhouse-gas-emissions
8. HS, M. (February de 2024). *Descubriendo el Poder del Vapor: Tipos y Aplicaciones*. Obtenido de https://medicalhs.com/blogs/articulos/descubriendo-el-poder-del-vapor-tipos-y-aplicaciones
9. Industry, N. S. (2006). *NORSOK STANDARD P-001.* Edition 5, Sep. 2006.
10. Konist, E. L. (2018). *International Scientific Conference "Environmental and Climate Technologies", CONECT 2018.* Tallinn University of Technology, Ehitajate tee 5, Tallinn 19086, Estonia.
11. Kruizenga, D. D. (2014). *Corrosion and Erosion Behavior in Supercritical {CO}$_2$ Power Cycles.* Sandia National Laboratories.
12. Lenntech. (2024). *Consumo de Agua de Torres de Enfriamiento*. Obtenido de https://www.lenntech.es/applications/process/cooling-towers-water-consumption.htm
13. Linares, J. I.-C.-D. (2023). *Carnot Battery Based on Brayton Supercritical {CO2} Thermal Machines Using Concentrated Solar Thermal Energy as a Low-Temperature Source.* Multidisciplinary Digital Publishing Institute.
14. REPSOL. (2024). *Aprovechando al máximo la energía generada*. Obtenido de https://www.repsol.com/es/energia-futuro/futuro-planeta/cogeneracion/index.cshtml
15. Rochau, J. P. (2012). *Supercritical CO2 Recompression Brayton Cycle: Completed Assembly Description.* Sandia National Laboratories.
16. Serrano Remón, I. P. (2014). *Análisis de sistemas de conversión de potencia en reactores nucleares de fusión con envolturas regeneradoras de doble refrigerante.* Madrid: Universidad Pontificia de Comillas.
17. Systemiq. (2023). *The Future of Energy Storage in Europe*. Obtenido de The Future of Energy Storage in Europe: https://www.systemiq.earth/wp-content/uploads/2024/02/240226-Country-pull-out-DK-vf.pdf
18. Vellini, M. G. (s.f.). *urbomachinery: Fundamentals, Selection and Preliminary Design.*
19. Weiland, N. T. (2019). *SCO2 power cycle component cost correlations from DOE data spanning multiple scales and applications.* Phoenix, Arizona, USA.
20. Zhisdorf, B. (2023). *Annex 58. High-Temperature Heat Pumps. Task 1 – Technologies. Technology Collaboration Programme in Heat Pumping Technologies (HPTTCP).* HPT-AN58-2.