POLITECNICO DI TORINO

Master of science program in Energy and Nuclear Engineering

Master thesis

WASTE HEAT VALORIZATION FROM DATA CENTERS: DESIGN OF AN ORC AND INTEGRATION WITH OTHER WASTE HEAT VALORIZATION SYSTEMS

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1 Abstract

Data Centres (DCs) are a critical part of the infrastructure that supports digitalization along with the electricity infrastructure that powers them. The ever-growing digitalization requires an expansion and evolution of data centres to process and store data. In the past years, since many new data centers have been built, the overall electricity consumption of this kind of infrastructure has increased significantly. The share of electricity consumption on total electricity demand in Europe is supposed to pass from less than 4% in 2022 to more than 5% in 2026 [1]. According to IEA data centers in 2022 accounted for 0.6% of total GHG emissions and for 0.9% of energy-related GHG emissions [2]. A potential solution to boost the overall energy efficiency of data centers is Waste Heat Recovery (WHR) and reutilization. The primary drivers of data centres electricity demand are the cooling systems and the servers themselves, with each typically accounting for 40% of the total consumption [1]. Regarding the extraction of heat from ICT equipment, immersion cooling is a liquid cooling method in which the cooling medium is in close and direct contact with ICT equipment. This technique requires lower electricity consumption than traditional computer room air cooling. The heat absorbed by the cooling medium can be used for instance to supply a District Heating (DH) system, or can be released in another fluid for direct exploitation, like for example in the tank of a Domestic Hot Water (DHW) system, in the boiler of an Organic Rankine Cycle (ORC), or in a heat storage system. The scope of this thesis is to design an Organic Rankine Cycle (ORC), fed by waste heat from a data center and to develop a preliminary working logic for the integration of it with other waste heat recovering systems. Regarding the design of the ORC, the selection of the working fluid has been performed firstly by conducing a pre-screening based on fluid properties, environmental aspects and toxicity concerns and then by assessing which fluid results in the highest thermal efficiency. R1234yf has resulted to be one of the best candidates and has been chosen for the further analyses. Regarding the architecture, a basic architecture for the ORC has been chosen, basing on a techno-economic evaluation. The design of the ORC system has resulted in a thermal efficiency varying depending on the operating conditions of the system and having a maximum value of 6.4%, which is a reasoning value compared to the ones that can be found in literature. Being this thesis developed in a company having the aim of building this machine, later the design of the system has been performed, in particular the sizing of the piping and of the various components (pump, boiler, turbine, electric generator, condenser and refrigerant collector tank) has been performed. Furthermore the engineering documentation required for the production of the system has been compiled. In particular this documentation includes the Battery Limits (BL), Fluids List (FL), Block Flow Diagram (BFD), Process Flow Diagram (PFD), Mass and Energy Balance (M&EB), Process Data Sheet (PDS), Mechanical Data Sheet (MDS), Equipment List (EL), Instrument List (IL) and Valve List (VL). The ORC system is part of a larger system which is being developed by the company where this thesis has been developed together with other companies from various parts of the EU. The name of the EU project is MODERATOR. The system that is to be designed is composed by a Space Heating (SH) system, a domestic hot water system, a thermal energy storage (TES), the ORC and other heat recovering systems, such as the heating of greenhouses and the drying of food products, that have not been considered in this work. The first two systems are supposed to supply heating to the data center, while the thermal energy storage is meant to store surplus heat from the data center in order to use it when needed both inside the data center and for other purposes and the ORC produces electricity which can be used inside the data center. These four systems are meant to be integrated in order to maximize the energy recovery from waste heat. The preliminary development of the logic of the integrated system is part of this thesis, together with the production of some engineering documentation for the integrated system comprehending the battery limits, the block flow diagram and the process flow diagram.

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List of Symbols

Symbol	Variable	Unit of measurement
A	area	m^2
c	specific heat	$\frac{kg}{K}$
D	diameter	m
ϵ	roughness	m
F	factor	m
f	maximum allowable stress	Pa
$fric$	friction factor	-
Φ	heat flux of the boiler	kW
H	head	m
h	mass based enthalpy	$\frac{kJ}{kg \cdot K}$
η	efficiency	-
l	length	m
$\dot{m}$	mass flow rate	$\frac{kg}{s}$
μ	dynamic viscosity	$\frac{kg}{m \cdot s}$
p	pressure	Pa
QF	quality factor	-
R	radius	m
Re	Reynolds number	-
ρ	density	$\frac{kg}{m^3}$
σ	mechanical stress	Pa
t	thickness	m
T	temperature	K
v	velocity	$\frac{m}{s}$
$\dot{V}$	volume flow rate	$\frac{m^3}{s}$
V	volume	m^3
W	work of the turbine	kW
y	load losses	m
γ	specific weight	$\frac{kg}{m^2 \cdot s^2}$

List of Subscripts

Subscript	Meaning of the subscript
allow	allowed
B	boiler
corr	corrosion
design	design
in	internal
m	ultimate tensile
min	minimum
P1.0	related to a 1% permanent deformation
p	constant pressure
PB	post-boiler
ther	thermal
turblp	low-pressure turbine
turbhp	high-pressure turbine

2 Introduction

2.1 Current data centers overview and future trends

A Data Center (DC) is a facility designed to house computer systems and IT infrastructure securely and efficiently. It provides a controlled environment for the continuos operation of critical application and data. Their key functions are:

- data storage and management: storing and managing large volumes of data;
- processing: running applications and performing complex computations;
- hosting services: hosting websites, cloud services and virtual machines.

Data centres are a critical part of the infrastructure that supports digitalization along with the electricity infrastructure that powers them. The ever-growing digitalization requires an expansion and evolution of data centres to process and store data. There are currently more than 8.000 data centres globally, with about 33% of these located in the United States, 16% in Europe and close to 10% in China. In the last years the electricity consumption of data centers has been significantly amplified: in 2022 it accounted for 460 TWh [1] with respect to the 25,500 TWh [20] of global electricity consumption in the same year, resulting in a share of 1.57%. The growth of electricity consumption of data center from 2019 on is shown in the following graph, where the continue line indicates the registered data and the dotted line indicates the forecasted data:

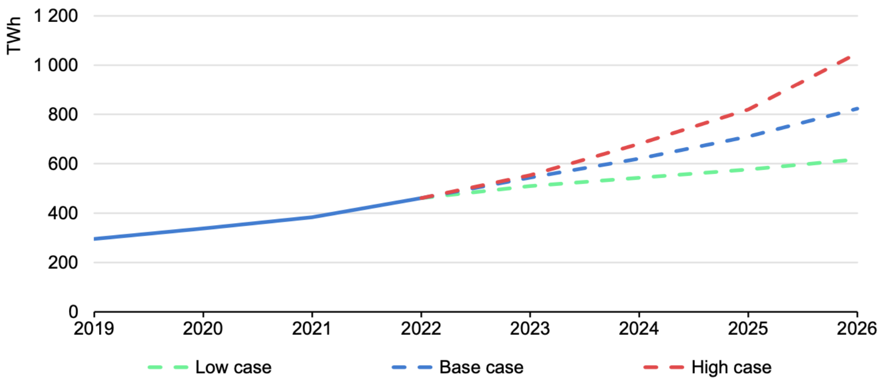


Figure 1: Electricity consumption of data centers from 2019 on [1]

In a report called "Electricity 2024 - Analysis and forecast to 2026", published by International Energy Agency (IEA), the share of electricity consumption by datacenters on total electricity demand in Europe is supposed to pass from less than 4% in 2022 to more than 5% in 2026.

Some data regarding the expected increase in electricity consumption from data centers can be extrapolated from the following graph:

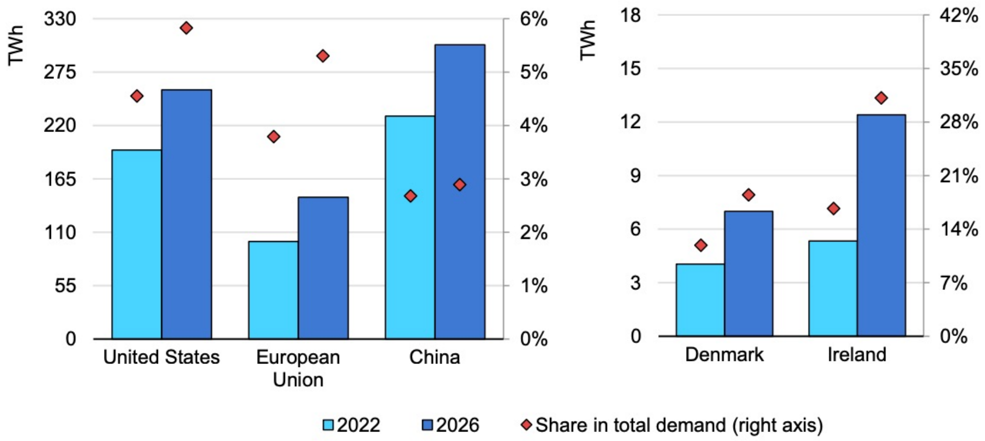


Figure 2: Estimated data centre electricity consumption and its share in total electricity demand in selected regions in 2022 and 2026 [1]

According to IEA data centers in 2022 accounted for 0.6% of total GHG emissions and for 0.9% of energy-related GHG emissions [2]. Therefore there is an urgent need for data centers to assess methods for saving energy and reducing emissions. The primary drivers of data centres electricity demand are the cooling systems and the servers themselves, with each typically accounting for 40% of the total consumption [1]. The remaining 20% is consumed by the power supply system, storage devices and communication equipment. The adoption of high-efficiency cooling systems has the potential to reduce electricity demand in data centres by 10%. Other cooling research shows that a 20% reduction in consumption can be achieved when operating with direct-to-chip water cooling and specific low viscous fluids to cool all other components [1]. A potential solution to boost the overall energy efficiency of data centers is Waste Heat Recovery (WHR) and reutilization.

2.2 Waste heat recovery from data centers: state of the art

Waste heat recovery from data centers is associated to low quality heat, since the temperature at which this energy is available is low, and this limits its possible uses. According to literature survey, the current available and possible WHR methods and waste heat uses in DCs include heating and District Heating (DH) supply, cooling production in absorption or adsorption cooling cycles for power need reduction, electricity production, clean water production through desalination and agricultural processes. In this Section the previously mentioned methods will be discussed, focusing on both how the technology works and on experimental activities found in literature.

- Heating: waste heat from DCs can be used for heating supply to surrounding residential directly or after simple processes like pre-heating and dehumidification. The temperature and quality of waste heat in DCs greatly limit its direct application. Fortunately, heat pumps (HPs) are widely applied to recover waste heat in DCs since they can increase the temperature and grades of waste heat for various heating demands, as for example Domestic Hot Water (DHW), which is considered as the most promising method to achieve WHR in DCs. Apart from HPs, Thermal Energy Storage (TES) is also belived to improve WHR efficiency, which can balance the mismatch between the heat supplies from the waste heat system and the heat demands for the subsidiary buildings. Wang et al. [21] adopted a CO_2 ground source HP (GSHP) system to establish a prosumer DC energy system to achieve WHR for the heating supply in the surrounding buildings and stored the waste heat during the non-heating seasons by a CO_2 direct-expansion GSHP. Through 3 E analysis (Energy, Exergy and Economic), they found the heat-power ratio can achieve 5.7 in the prosumer DC WHR system. Compared with conventional cooling system (air-source heat pumps), the proposed system accounted for almost half (52.8%) of the total power consumption, and significantly improved the matching thermodynamic performance with a 3.87 improvement on exergy efficiency. In addition, the annual net profit in the proposed system is 190.34% compared to the conventional system;

- DH: apart from utilizing waste heat for heating supply in the nearby buildings directly or after pre-heating, waste heat from DCs can also be recovered for supplementing heat loads in DH networks. Differently from WHR for nearby building heating with different temperature levels, excess heat for DH networks need to meet certain supply water temperature requirements (45-55$°C$). Thus, preheating is compulsory for waste heat utilization in DH networks, which requires additional heating devices and power input, and the technical, energy and economic analysis are necessary to study the waste heat recovery feasibility. In the following figure the rough sketch of how the waste heat from DCs links to the DH networks is shown:

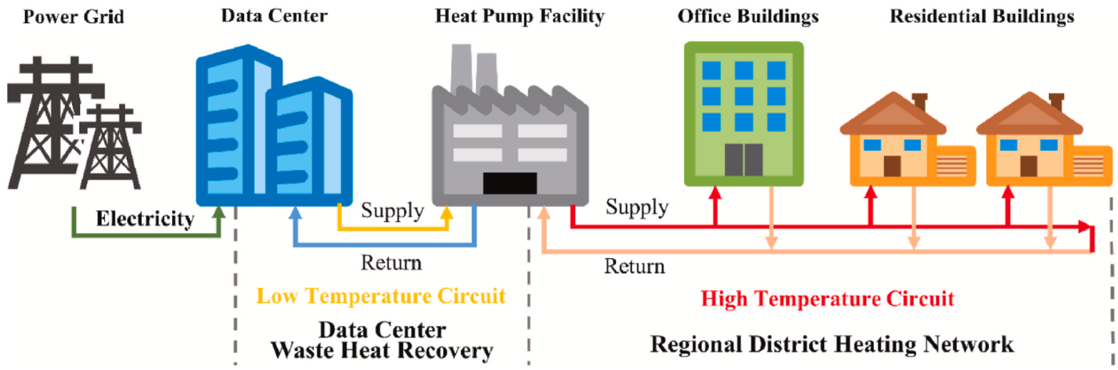


Figure 3: Linking between waste heat from DCs and DH networks [3]

HPs and TES can be integrated with this kind of system in order to obtain the same advantages as described previously for the heating case. Li et al. [22] claimed that although utilizing waste heat from DCs for DH systems is feasible, its utilization is limited by the mismatch between heat supply and heat demands from and to DCs and DH systems, respectively. In addition, the operation cost of the DH systems can be greatly affected by the high peak loads. Thus, they recommended using both short-term (e.g., water tank) and long-term (e.g., borehole) TES to balance the mismatch. They also evaluate the proposed system integrating TES into DH networks based on WHR from DCs though 3 E analysis. They found the peak load can be shaved by the water tank by 31%, which also saved 5% energy cost yearly, and the payback period was below 15 years under the circumstance of higher storage efficiency than 80%. In addition, the borehole TES can achieve WHR rate up to 96%, bringing an annual CO_2 emission reduction of 8%. However, the payback period for the long-term TES was beyond 17 years;

- cooling production: WHR can be used to supply both absorption cooling, adsorption cooling and evaporative cooling.
  - Absorption cooling: using the absorption units for low-grade waste heat utilization will not contribute to greenhouse gases emissions, and the absorption cooling system suits for the systems with all sizes (small, medium and large), which is also simpler than other conventional cooling system. Han et al. [23] proposed using waste heat in DC to drive the $LiBr - H_2O$ absorption cooling system for further cooling to servers, and adopted an enhanced geothermal system with obsolete oil wells for TES. They also analysed the effects of some parameters (e.g., well depth, porosity, injection rate, permeability, and reservoir width) on the proposed cooling system performance. They concluded that the enhanced geothermal system could cause better $LiBr - H_2O$ cooling system performance, and the system cooling capacity can reach 9 MW with a COP of beyond 1.0;
  - adsorption cooling: adsorption refrigeration systems with $LiBr - H_2O$ or $NH_3 - H_2O$ perform well in DCs when the waste heat temperatures ranges between 65$°C$ and 100$°C$. However, the system will shut down and have much low COP under the circumstances of lower temperatures (below 65$°C$). In addition, absorption chillers have the characteristics of low power to weight ratio, which occupy a large area unsuitable for most of DCs with constrained space, and the used absorbents are environmentally unsustainable and corrosive. Thus, restrictions exist in the application of absorption refrigeration in WHR DCs. In addition to absorption refrigeration, adsorption refrigeration is also applicable in DC WHR as its driving temperature can be as low as 60$°C$. In Germany in 2017,

Wilde et al. [24] firstly built a WHR project with adsorption cooling system in DC and proposed recovering waste heat in the high-performance computing station adopting high-temperature direct-liquid cooling system (namely CooLMUC-2 system), and used the waste heat to drive an adsorption chiller to generate cold water for other IT components cooling. Based on a case study, they found the integrated CooLMUC-2 and adsorption chiller could achieve both 95 kW heat removal from the super-computer and over 50 kW cold water production at the cost of only around 6 kW electricity consumption after optimizing the system operation parameters;

  - evaporative cooling: the waste heat can also be used for achieving evaporative cooling in DCs. In an evaporative cooling system, water evaporation process can achieve fresh-air humidified, and cooling of fresh-air by water evaporative latent heat. Endo et al. [25] firstly proposed a combined direct fresh-air cooling system with both evaporative cooling and WHR from IT equipment in a container DC. After one-year examination, they found that the proposed system could maintain the server environmental conditions with acceptable ranges and could annually reduce the total energy consumption by around 21% compared with container DC with Computer Room Air Conditioning (CRAC);

- WHR for electricity management and production: in addition to being used directly or indirectly for low-quality energy use, like cooling and heating, waste heat from DCs can also be used for the high-quality energy, like mechanical and electrical after conversion. Using waste heat for electricity production can be achieved by heat recovery cycles. ORC is considered as the first choice for its flexible use of electricity production based on waste heat, which has the possibility to achieve WHR in DCs and electricity production. Ebrahimi et al. [26] proposed utilizing ultra-low temperature waste heat from a dual loop DC to drive ORC system for electricity production and validated its feasibility through both thermodynamic and economic analysis. They claimed that this work firstly fully analysed the matching between ORC system and DC operating conditions. Super-heaters are always utilized in ultra-low (less than $85°C$) heat sources to boost the heat recovery temperatures, which were adopted by Ebrahimi et al. [26] as the temperatures of waste heat from DCs were below $85°C$. Through analysis, they found the super-heaters negatively impacted the overall performance due to their extra power input. In addition, the system performance is also significantly affected by increasing the waste heat temperatures (e.g., temperature differential decreasing between the chip and the micro-evaporator and increasing of chip operating temperature). They found R245fa and R134a were the optimum mediums for the ORC working fluid and the server coolant, respectively. The repayment period was 4–8 years considering the variation of regional commercial electricity cost;

- desalination process: Kanbur et al. [27] proposed an improved cooling system, which adopted two-phase liquid-immersion and used waste heat from DCs for desalination, and they assessed its cooling performance by thermos-economical and thermos-dynamic multi-criteria (e.g., thermal and exergy efficiencies, capital and operation costs, the levelized product cost);

- agricultural process: Ljungqvist et al. [28] came up with the idea of combining the DC industry with agricultural production and investigated the feasibility of waste heat from DCs as heat sources for food greenhouse in Northern Sweden. They used IDA ICE 4.8 to build up the energy model and computed the relationship between energy supply from DC waste heat and energy demand of greenhouse production. They concluded using waste heat from DCs for self-efficiency of greenhouse food production is feasible and achievable and could achieve energy cost savings.

2.3 Immersion cooling technology

There are two main obstacles related to the reuse of waste heat from data centers:

- fist of all, the heat that is typically produced in an air-cooled data centers is of poor quality (low temperature and diffuse), so that even for immediate use it is often not economically viable to design exploitation measures;

- secondly, data centers produce more waste heat in summer, when it is hot outside, but such heat is typically needed and could be better utilized in winter.

The two obstacles currently hinder the commercial application of conventional WHR systems in data centers. Immersion cooling is a liquid cooling method in which the cooling medium is in close and direct contact with ICT equipment. This techniques requires lower electricity consumption than traditional air cooling. In immersion cooling method, the ICT equipment is immersed in a dielectric fluid, capable of conducting heat and acting as an electrical insulator, and the heat generated by it is absorbed by the cooling fluid. Finally, the heat absorbed by the cooling medium can be used for instance to supply a SH system or can be released in another fluid for direct exploitation, like for example in a DHW system, or in the boiler of an ORC, or in a TES, thus solving the two problems described in the previous bulleted list. Immersion cooling systems prove to be more efficient than traditional data center cooling methods (like CRAC) due to the increased thermal conductivity of most liquids compared to air. Examples of dielectric fluids include mineral oil hydrocarbons, synthetic fluorocarbons, and silicone fluids. The value of thermal conductivity for the main used dielectric fluids in data centers with respect to air is shown in the following graph:

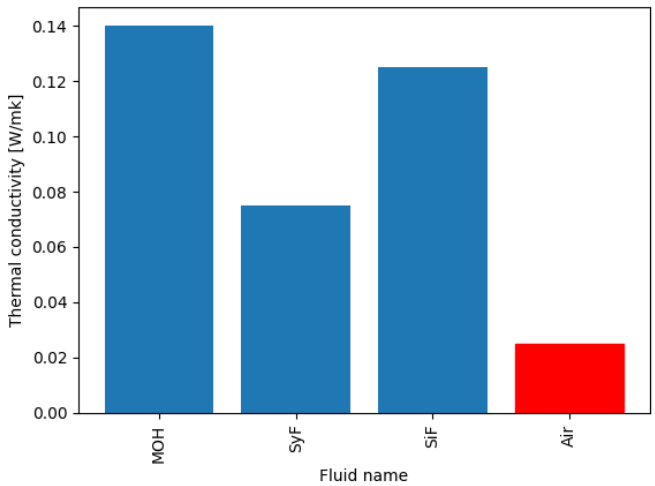


Figure 4: Thermal conductivity of Mineral Oil Hydrocarbons (MOH), Synthetic Fluorocarbons (SyF) and Silicon Fluids (SiF) vs thermal conductivity of air [4]

From a fundamental level, liquid immersion cooling data center components involves dunking them in a dielectric fluid. From there, the process can look drastically different depending on whether a two-phase vs. single-phase immersion cooling system is being used.

- Single-phase immersion cooling: in single-phase immersion cooling, heat from the immersed server components is transferred directly to the surrounding fluid. However, the dielectric fluid does not undergo a “phase change” from a liquid to a gas. Instead, the fluid is cycled out of the immersion tank by a coolant pump that runs through a heat exchanger and is returned to the immersion tank at a lower temperature where it continues this heat transfer cycle;

- in two-phase immersion cooling, heat from immersed server components causes the special immersion fluid to boil. The resulting steam heats a condenser coil in the top of the sealed chamber. The coolant in the condenser coil is cycled out of the chamber to a heat rejection mechanism (cooling tower, etc.). Then, the coolant is sent back to the sealed chamber at a lower temperature, ready to continue the heat transfer cycle. One of the key elements of this type of immersion cooling tank is the low boiling point of the immersion fluid. However, one of the key complaints of two-phase cooling include the sealing functionality of the immersion chamber. Because of the steam from the phase change in two-phase liquid immersion cooling, the chamber must be sealed during operation. This means that performing maintenance requires a cooling and unsealing process that costs valuable operation time.

Server immersion cooling offers myriad benefits for IT operations:

- immersion cooling uses less energy than air: the average power usage effectiveness ratio (PUE) within a data center can be measured by dividing total energy consumed by energy used by computing equipment. This means that as PUE gets closer to 1, efficiency is improving. According to The Register, PUE for a traditional data center in 2022 was approximately 1.58, while single-phase immersion was able to bring this number down to the 1.05 to 1.10 range [29];

- immersed servers do not require air conditioning: traditional data center infrastructure would indirectly cool servers via raised floors, CRAC units, and strategically placing units on the data center floor. Immersion removes this requirement which accounts for around 35-40% of the power used in a data center. This eliminates the need for hot and cold isles, raised floors, and CRAC units;

- saves space: not only does immersion cooling improve the energy efficiency of data centers, but it can save valuable space as well. According to a 2023 research article, immersion cooling only requires about one-third of the space to that of an air-cooled configuration [30]. One of the main contributors to this efficiency is the improved rack power density from not having to allow for air flow within servers. Keep in mind, an immersion cooling server configuration is a huge departure from a traditional rack-mount server. Space efficiencies will likely not come into play by placing one or two submerged server cooling tanks in your data center. A widescale change in a data center floor plan may be required before enjoying this benefit;

- increases rack density: an average, fully populated data center 42U cabinet (a 42U cabinet in data centers is a standardized rack that provides a specific amount of vertical space to house IT equipment) holds 5-7 kW worth of server compute. Alternatively, fully populated 42U immersion racks can hold over 380 kW of compute. The safe computing density advantages that come from liquid immersion cooling solutions cannot be disputed. Not to mention, these immersion tanks can be installed anywhere with a power and network hookup;

- reduces data center noise: CRAC, one of the main traditional cooling methods for data centers, is reliant on the use of fans. This means that traditional data centers are very loud. Immersion cooling server configurations don't rely on fans and air flow for cooling. Because of their liquid cooling function, immersion cooling has proven to reduce data center noise [31].

2.4 Scope

The scope of this thesis is the design of an Organic Rankine Cycle, fed by waste heat from a single-phase immersion cooled data center and the production of the engineering documentation necessary for the construction of this device. Furthermore a preliminary study regarding the working logic of the Integrated System (IS), comprehending the ORC together with the Space Heating (SH) system, the Domestic Hot Water system and the Thermal Energy Storage is to be performed and a preliminary engineering documentation regarding the IS has to be prepared.
In the section "Methodology"(3), the process for the selection of the working fluid and the architecture will be explained in details, the operation of the ORC in all the working range will be examined, the design of the ORC will be analysed, the reasoning behind the production of engineering documentation regarding the ORC will be assessed, the main ideas behind the development of the working logic of the ORC will be discussed and finally the compilation of some engineering documentation regarding the IS will be inspected. For the ORC the engineering documentation will comprehend the Battery Limits (BL), the Fluids List (FL), the Block Flow Diagram (BFD), the Process Flow Diagram (PFD), the Mass and Energy Balance (M&EB), the Process Data Sheet (PDS), the Mechanical Data Sheet (MDS), the Equipment List (EL), the Instrument List (IL) and the Valve List (VL), while for the IS the engineering documentation will comprehend the Battery Limits, the Block Flow Diagram and the Process Flow Diagram. In the section "Description of the case study"(4), the case study analysed will be presented, in the section "Results"(5) the results from the analysis performed will be presented, dividing them in the ones regarding the ORC and that regarding the integrated system, and will then be discussed in the section "Discussion"(6). In the section "Conclusion" the main achievements obtained will be stated, comparing them to what has already been done and can be found in literature.

2.5 State of the art for low temperature fed ORCs

In the following some case studies regarding ORC operating in temperature ranges between around 50°C and 20°C will be assessed in order to give an idea of which are the efficiencies achievable with this kind of thermal machines operating at these temperature levels. J.L. Wang et al. have performed a comparative study of pure and zeotropic mixtures in low-temperature solar Rankine cycle [5]. The temperature levels between which this system operated can be assumed equal to the inlet and outlet temperature of the collector, that are represented in the following graph as a function of time during the day.

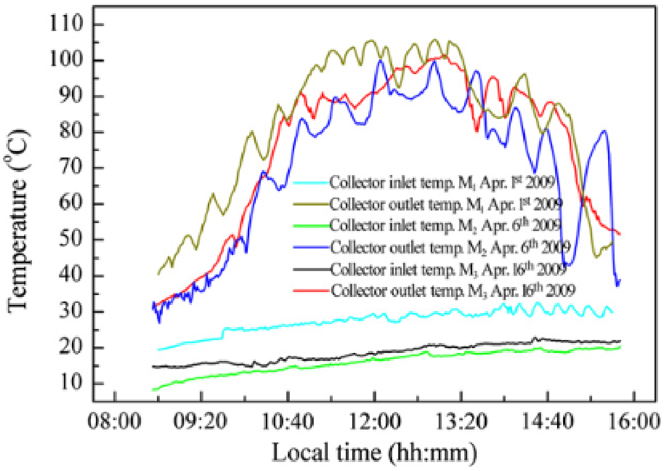


Figure 5: Temperature levels in low-temperature solar Rankine cycle operating with pure and zeotropic mixtures [5]

As it can be seen from Figure 5, considering for the inlet and outlet temperature of the collectors an average evaluated on the time and on the three mixtures, the average collector inlet temperature is approximately 20°C and the average collector outlet temperature 70°C. The efficiencies reached with the three different mixtures are respectively 0.88%, 0.92% and 1.28%. Xia Zhou et al. have performed a comparative study for waste heat recovery in immersion cooling data centers with district heating and Organic Rankine cycle (ORC) [6]. The temperature levels between which the four cases analysed operated are represented in the following histogram:

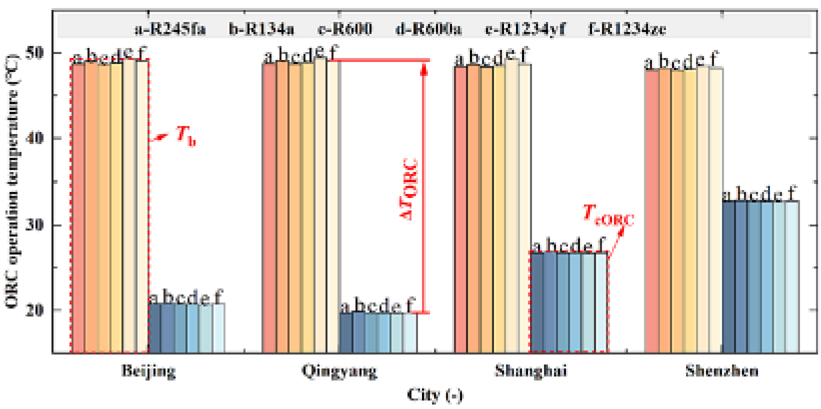


Figure 6: Comparative study for waste heat recovery in immersion cooling data centers with district heating and ORC [6]

As it can be seen from Figure 6 the high temperature level for the four cases is on average 50°C, while the low temperature level 25°C. The efficiencies in decreasing order are 5.7%, 5.5%, 4.3% and 3.8%. M. Antonelli et al.

have performed an analysis of a low concentration solar plant with compound parabolic collectors and a rotary expander for electricity generation [7]. The results are summarized in the graph below:

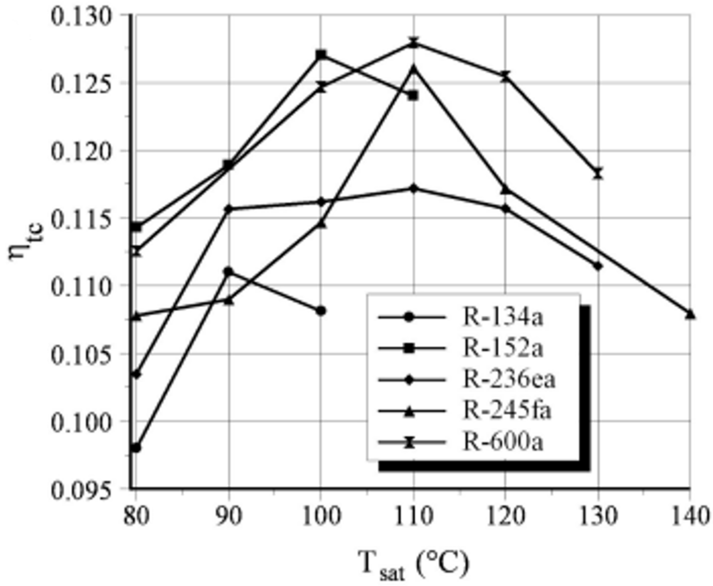


Figure 7: Analysis of a low concentration solar plant with compound parabolic collectors and a rotary expander for electricity generation[7]

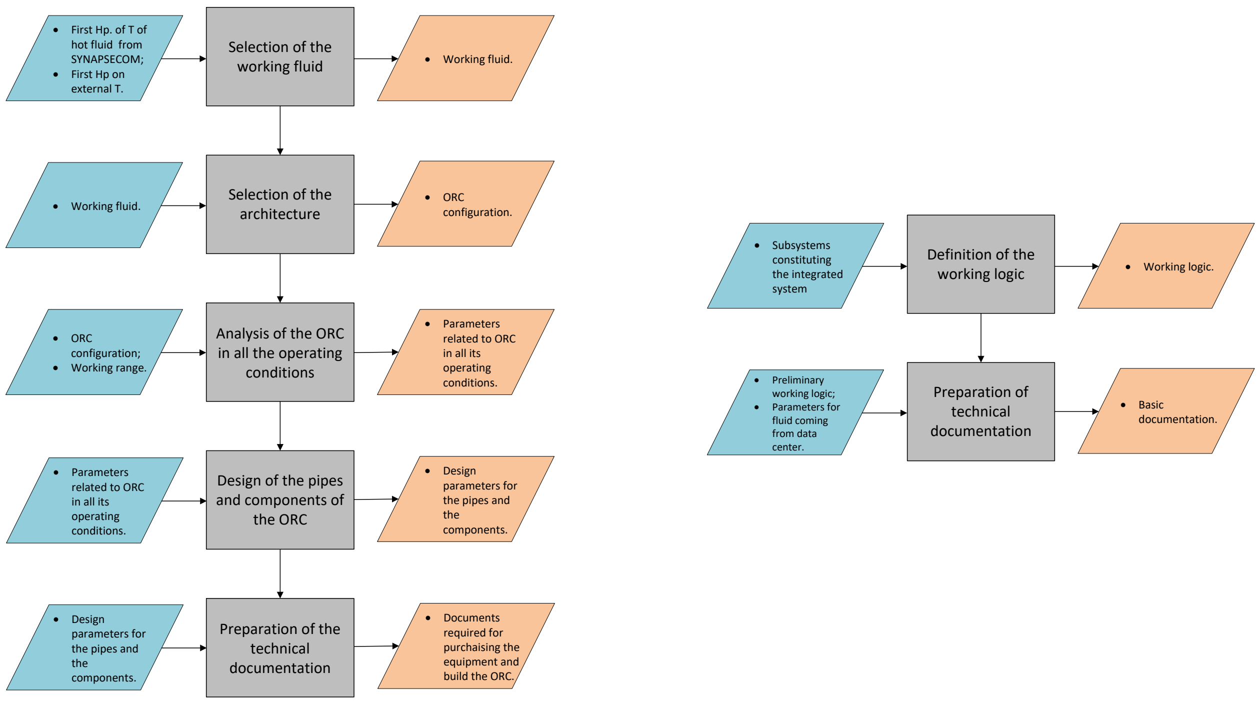
As it can be seen from Figure 7, considering a saturation temperature of $80^{\circ}C$ the efficiencies varies for the different fluid analysed and ranges between around 9.8% and 11.4%.

3 Methodology

In this section the structure of the work is represented. This is divided in two parts: one concerning the assessment of the ORC and another one regarding the integrated system. The work development is briefly described in the following bulleted list:

- analysis of the ORC: at first the working fluid is selected, then the architecture is selected, then an analysis of the ORC in all its operating conditions is performed, then the pipes and the components of the ORC are designed and finally the technical documentation is assessed;
- analysis of the integrated system: first a preliminary logic of the integrated system is developed and then the technical documentation is assessed.

What has just been said is summarized in the following schemes, where the inputs and the outputs of all the processes are indicated:



(a) Methodology scheme for the ORC

(b) Methodology scheme for the IS

Figure 8: Methodology scheme. Water green: input, grey: process, orange: output

It is to be noted that in the Figure 8a and 8b the parts highlighted in water green indicates the input of the process and the ones highlighted in orange indicates the output of the process.

3.1 Analysis of the ORC

3.1.1 Selection of the working fluid

The first consideration that has been done while selecting the working fluid to be used for the cycle is that it had to be an organic fluid, in particular a refrigerant. In fact this type of organic substance works well when exploiting heat from low temperature heat sources, mainly because of the low boiling temperatures. In order to carry out the analysis, since thermodynamic properties in the main working points of the cycle where needed, the translated-consistent Peng-Robinson equation has been taken into account while modelling the behaviour of the fluids. In particular an open source thermodynamic package implemented in Python, called "ThermoPack", which implements this equation for a defined list of fluids for which the needed parameters are contained in the database included in the own package has been used. The preliminary list of fluids that has been considered is indeed composed of refrigerants available on the "ThermoPack" database and it is show below.

Fluid name
R134a
R143a
R152a
R1132a
R114
R124a
R142b
R1234yf
R124
R14
R22
R115
R13
R12
R21
R32
R116
R41
R218
R125
R1114
R1234ze
R11
R23
R290

Table 3.1: Refrigerants' list from ThermoPack

Once selected the preliminary list of refrigerants, the analysis has been developed following the next steps: first a pre-screening of the fluids has been done and than an analysis aiming at finding the refrigerant that maximizes the thermal efficiency of the cycle has been conduced. It is to be noted that, while carrying out this preliminary analysis, the temperature at the outlet of the boiler has been considered equal to $40°C$ and the condenser temperature equal to $30°C$. These temperatures are just first estimates of the working range of the ORC. Regarding the first step the following criteria has been used:

- critical points: since the aim was to work in the biphase region of the phase diagram during evaporation, criterias has been set both on critical temperature and critical pressure. Regarding the critical temperature this had to be at least $10°C$ higher than the maximum temperature of the cycle, namely, being the maximum temperature of the cycle $40°$, the critical temperature had to be at least higher than $50°$. Regarding the pressure criteria, the limit imposed was that the maximum pressure had to be 10 *bar* lower than the critical pressure. Indeed, near the critical point, the fluid properties undergo strong variations: for low temperature variations, high variations of pressure are obtained making the whole system unstable. The maximum pressure that can be reached in the cycle corresponds to the one in the evaporator, which is then compared with the limit pressure, namely the critical pressure minus 10 *bar* as said previously, in order to assess the pressure criteria. The two criteria just described have been analysed by implementing a Code on python (1). As it can be seen from the previous code, for the fluids satisfying temperature requirement, the evaporator pressure and the limit pressure, evaluated as previously shown in the section, are stored in the vectors p_{ev} and p_{lim} respectively, while for fluids not satisfying temperature criteria the value 0 is assigned for the corresponding elements in the two vectors: this way while plotting the two vectors to perform the check of the pressure constraint, only fluids satisfying temperature requirement are assessed;

- Ozone Depleting Potential (ODP): the ODP of a chemical compound is the relative amount of degradation to the ozone layer it can cause, with trichlorofluoromethane (or R-11), being fixed at an ODP of 1.0. ODP of a compound is used as a measure of how environmentally detrimental it can be. The substance with the smallest ODP, which is classified as an ozone-depleting substance in the EU regulation No 1005/2009, has an ODP of 0.01 [32]. Tacking this into account, in this analysis the threshold for the upper value of the ODP has been set to 10^{-3}. Refrigerants having ODP satisfying this requirements have been considered acceptable. In order not to exclude potential candidates as working fluids this threshold has been relaxed by two order of magnitudes and the refrigerants having ODP value ranging between the previous and the relaxed threshold has been considered "acceptable with warning";

- Global Warming Potential (GWP): Global Warming Potential is an index to measure how much infrared thermal radiation a greenhouse gas would absorb over a given time frame after it has been added to the atmosphere. The GWP makes different greenhouse gasses comparable with regards to their effectiveness in causing radiative forcing, namely the change in the energy balance of the Earth's atmosphere. GWP of a compound is used as a measure of how environmentally detrimental it can be. In this analysis the threshold has been set to 150, according to the new EU F-gas regulation 2024/573 [33] and then released by two order of magnitudes in order to define the categories "acceptable" and "acceptable with warning", as done with the ODP;

- toxicity: ASHREAE safety classification for refrigerants divides these compounds in two classes (A and B) corresponding to low and high toxicity respectively and four classes (1,2,2L,3) corresponding to increasing levels of flammability. A figure exhibiting the classification is shown below:

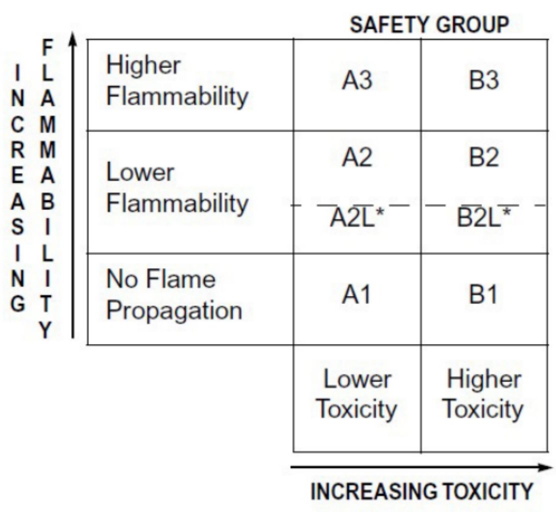


Figure 9: ASHRAE refrigerants' safety classification [8]

In this analysis the flammability is not taken into account in order not to exclude potential candidates as working fluid, and only toxicity is assessed setting as a criteria for the selection the classification of the fluids in class "A", namely "low toxicity". The symbol "−" in the Table A.3 means that for the specific refrigerant there is no classification available by ASHRAE. For those fluids the pictograms from Global Harmonized System of Labelling and Classification of Chemicals present in their data sheets are taken into account and only the fluids referred to by pictograms not involving toxicity hazard are assumed acceptable for the analysis.

In the following figure the pictograms involving toxicity are represented.

Figure 10: Pictograms involving toxicity hazard [9]

Once the pre-screening has been done, in order to assess which fluid was the best one for the cycle, the one resulting in the highest thermal efficiency has been considered. The thermal efficiency of the cycle for each fluid has been analysed by using Aspen Plus, as described in the following.
First of all, while designing the cycle, the constrains which come from the interface with the data center and the condenser type used have to be taken into account. The data center is cooled by using the "immersion cooling" technology and the dielectric fluid exchanges heat with a solution of water and propylene glycol that in the case of interest is heated up to 50°C and delivers 40kW of heat power to the working fluid. Considering a minimum temperature approach of 10°C in the boiler, which consists of a counter current heat exchanger between the working fluid and the dielectric fluid, the maximum temperature reached by the operating fluid in the boiler is of 40°C. The condenser is a fin heat exchanger. Considering the worst operating conditions, which happens in summer, and which is the period of the year in which the ORC works most of the hours, for the reason said in section 2.4 (in the other seasons the heat demand from SH and DHW will be predominant), by this device the working fluid exchanges heat with air at 30°C. Regarding the architecture, as a first approximation, when designing the cycle, a basic architecture, composed of a pump, a boiler, a turbine and a condenser will be considered. After the optimal fluid will be chosen also more complex architectures will be assessed. The cycle has been modeled as represented in the next figure:

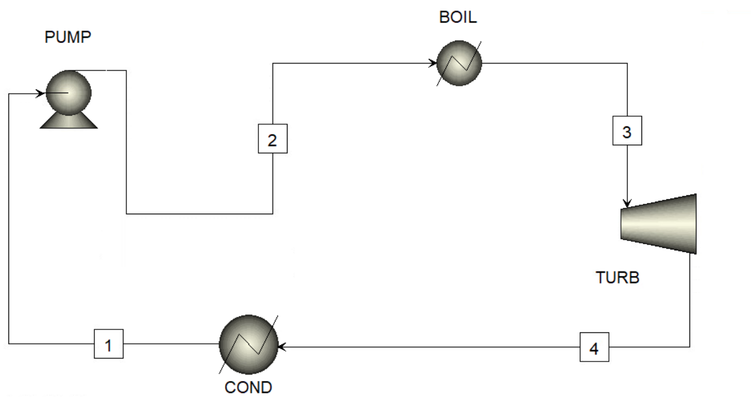


Figure 11: ORC model on Aspen Plus

The components represented in Figure 11 are a pump, a boiler, a turbine and a condenser and the four streams connects the various components. The following hypothesis have been considered:

- the boiler and the condenser have been assumed to be isobar;

- the pump has been considered as working with a high efficiency of 0.95;
- the turbine has been considered with high isentropic and mechanical efficiencies, both corresponding to 0.90.

In this cycle the liquid fluid is first rised in pressure by the pump, than it evaporates in the boiler, which in the reality is a counter current heat exchanger between the working fluid and the dielectric fluid coming from the data center, than the gaseous fluid is expanded in the turbine delivering power and finally the fluid is condensed in the condenser in order to bring it back to the inlet pump conditions. The cycle is represented in the T-s diagram in the following figure, considering as working fluid, for instance, R143a, which is the first refrigerant to be assessed in terms of thermal efficiency, as it can be seen in Figure 29:

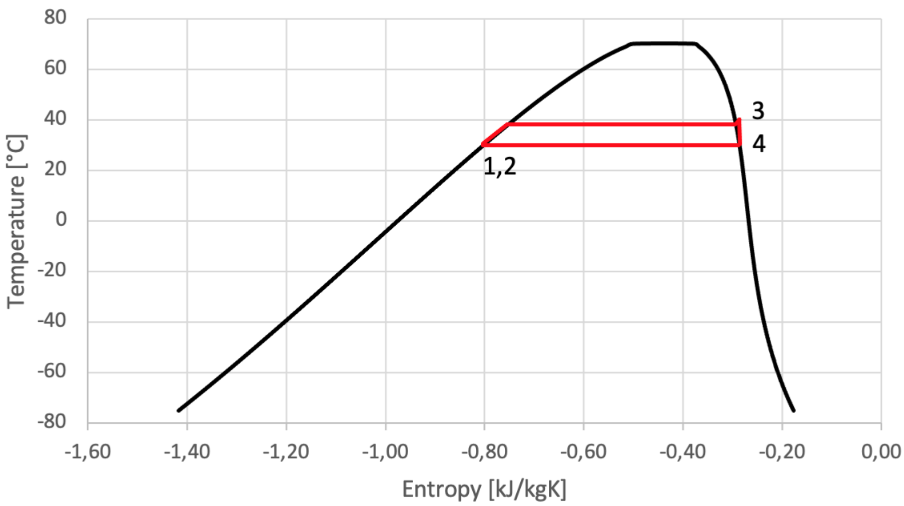


Figure 12: R143a ORC on T-s diagram

It is to be noted that in Figure 12 the values near the critical point are not represented well because the model on Aspen Plus doesn't represent the behaviour of the fluid faithfully in that points: so while simulating the behaviour of the saturation curve in the T-s diagram, points near the critical one have not been considered, and than while plotting the results the gap has been filled by a linear interpolation between the closest points on the two sides near the critical point that had been simulated.
Regarding the modeling on Aspen Plus, for the various components the following conditions have been set:

- for the pump the discharge pressure has been set by vaying it parametrically according to the sensitivity analysis, that will be discussed later in this subsection. The efficiency of the pump has been set according to the hypothesis previously stated in this part;
- for the boiler an outlet temperature of $40°C$ and a duty of 40 kW have been set;
- for the turbine an isentropic efficiency of 0.9 and a mechanical efficiency of 0.9 have been set according to the hypothesis previously stated in this paragraph;
- for the condenser an outlet temperature of $30°C$ and an outlet vapor fraction of 0 have been set.

Some "calculator" blocks on Aspen Plus has been set to model the cycle:

- "MF" calculator block is used to evaluate the mass flow rate in the cycle by using the formula $\dot{m} = \frac{\Phi}{h_3-h_2}$ where points 2 and 3 are the same as represented in Figure 12;
- "POUTBOIL" calculator block sets the pressure at the outlet of the pump equal to the pressure at the outlet of the boiler, according to the hypothesis of isobar evaporator;

- "POUTTURB" calculator block sets the pressure at the outlet of the turbine equal to the pressure at outlet of the codenser, according to the hypothesis of isobar condenser;
- "THERMEFF" calculator block is used to evaluate the thermal efficiency as $\eta_{therm} = \frac{W_{turb} - W_{pump}}{\Phi}$.

The modelling aims at finding the design conditions for each fluid, namely the ones that give the highest thermal efficiency for the fluid considered, and than repeating the analysis for each fluid in order to assess which one in its design conditions results in the highest thermal efficiency.
In order to find the design conditions for each refrigerant it has been taken into consideration that the boiling temperature and the condensing temperature are bounded by the constrains previously described in this subsection: in particular, the fluid should evaporate in a temperature range between a temperature higher than the minimum temperature of the cycle ($30°C$) and the maximum temperature of the cycle ($40°C$). Tacking into account the condensing temperature, in order to have the highest possible enthalpy step in the turbine, this has to be equal to the minimum temperature that can be reached in the cycle, namely $30°C$, assumed equal to the outlet air temperature in summer. Since at the exit of the condenser the fluid is assumed to be in saturated liquid conditions, at the temperature of $30°C$, the thermodynamic variables in this point are fixed, and so also the pressure is, which corresponds to the low pressure level of the cycle. Once the previous consideration has been done it is to be observed that the high pressure level of the cycle is a thermodynamic variable that can be changed in order to vary the thermal efficiency. This pressure is bounded between a minimum, which takes into account the low pressure level of the cycle, and a maximum, which takes into account the fact that expanding the fluid from $40°C$ it is required to have at the low pressure level a temperature higher that $30°C$, in order to have a subsequent cooling down to $30°C$ in the condenser. A sensitivity analysis regarding the thermal efficiency with respect to the pressure in the boiler has than been performed. For practical purposes the pressure in the boiler has been varied from a minimum, corresponding to the next integer higher than the value of the pressure in the condenser, in bar, and a maximum, which is obtained, increasing the pressure by 0.1 bar steps, up to the last value that satisfies the condition on the upper boundary previously said. The thermal efficiency has been evaluated for six points equally spaced and ranging between the minimum and the maximum just described.

3.1.2 Selection of the architecture

Once the optimal fluid has been selected, two more complex architectures of the cycle have been studied in order to maximize the thermal efficiency of the cycle, namely the ORC with recuperator and the reheat ORC. The ORC with recuperator has been assessed first. The some hypothesis regarding the pressure drop in the heat exchangers done for the basic architecture cycle has been considered in this phase. This cycle has been modeled by implementing the following variations with respect to the basic architecture cycle on Aspen Plus:

- two components have been added: the recuperator and the pre-heater. For the recuperator the outlet temperature has been set equal to $30.1°C$ as an extreme case in which most of the heat from the hot outlet of the turbine is recovered and the outlet vapor fraction has been set to 1;
- the calculator block "POUTPRHX" has been created in order to set the pressure in the pre-heater equal to the outlet pressure of the pump;
- a heat flux stream has been added in order to link the heat recovered by the recuperator to the heat provided to the pre-heater.

The layout of the ORC with recuperator on Aspen Plus is represented in the following figure:

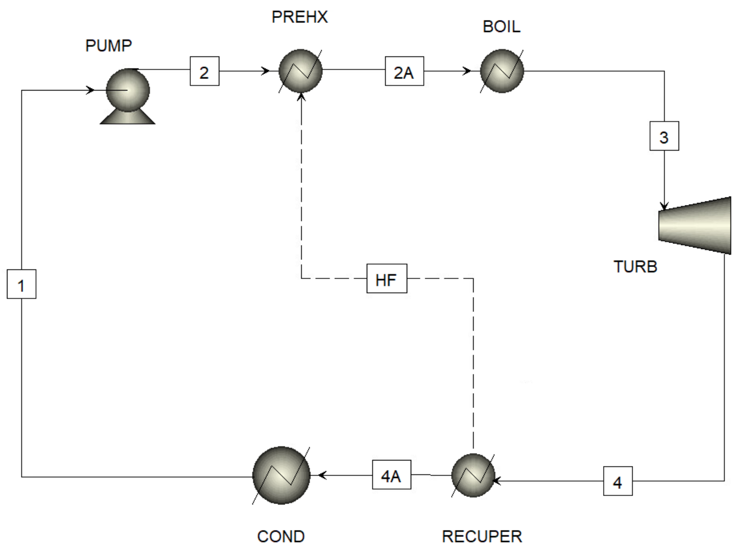


Figure 13: Model of ORC with recuperator on Aspen Plus

For this architecture the thermal efficiency has been evaluated and then compared with the one corresponding to the basic architecture cycle.
Secondly the reheat ORC has been assessed. The same hypothesis regarding the pressure drop in the heat exchangers and the efficiencies of the turbines, done for the basic architecture cycle, have been considered for the components added in this phase. This cycle has been modeled by implementing the following variations with respect to the basic architecture cycle on Aspen Plus:

- two components have been added: a post-boiler and a low-pressure turbine. The turbine which is preceded by the boiler have been then named high-pressure turbine, and its parameters has been modified. Also the parameters of the boiler have been modified. For the high-pressure turbine an outlet pressure of 7.81806 bar, namely the average pressure between the high and low pressure levels, has been set, while the isentropic and mechanical efficiencies have been assumed according to the hypothesis. For the low-pressure turbine efficiencies the same considerations done for the high-pressure turbine have been done. For the boiler the duty has been varied parametrically according to the sensibility analysis described later in this subsection;

- the "DPB" block sets the duty of the post-boiler equal to total heat duty delivered by the dielectric oil coming from the data center, namely $40kW$, minus the duty of the boiler;

- the "POUTPB" calculator block sets the pressure at the outlet of the post-boiler equal to the pressure at the outlet of the high-pressure turbine, according to the hypothesis;

- the "POUTTURB" calculator block has been modified since this time two turbines are present in the cycle. In particular the pressure at the outlet of the low-pressure turbine has been set equal to the pressure at the outlet of the condeser according to the hypothesis;

- the "THERMEFF" calculator block has been modified since this time the output power comes from two turbines and the thermal power supply comes from two heat exchangers. The thermal efficiency has been then defined as: $\eta_{therm} = \frac{W_{turbhp}+W_{turblp}-W_{pump}}{\Phi_B+\Phi_{PB}}$.

The layout of the reheat ORC is represented below:

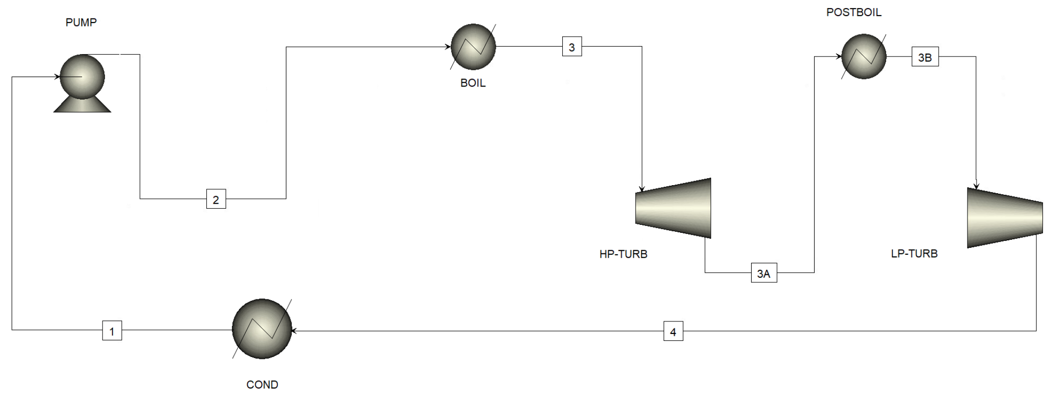


Figure 14: Layout of reheat ORC

For this architecture a sensitivity analysis has been done to determine how the thermal efficiency of the cycle changes varying the heat duty of the boiler from 20 to 40 kW with steps of 1 kW.

3.1.3 Analysis of ORC behaviour in all its working conditions

In this section an analysis of the behaviour of the ORC in all its working conditions is carried out. In order to assess the working conditions of the ORC it has been taken into account that:

- the hot fluid arriving from SYNAPSECOM towards the boiler has a temperature ranging between 50°C and 80°C and so, considering a temperature approach of 5°C, the working fluid in the ORC operates with a maximum temperature in the range from 45°C to 75°C;
- the finned condenser exchanges heat with air. The maximum summer temperature and the minimum winter temperature in Athens have been assessed respectively as the highest temperature in summer and the lowest temperature in winter in the last ten years. These has resulted to be 43°C and -4°C [34]. The behaviour of the maximum and minimum temperatures in Athens in the last ten years is represented in the following figure:

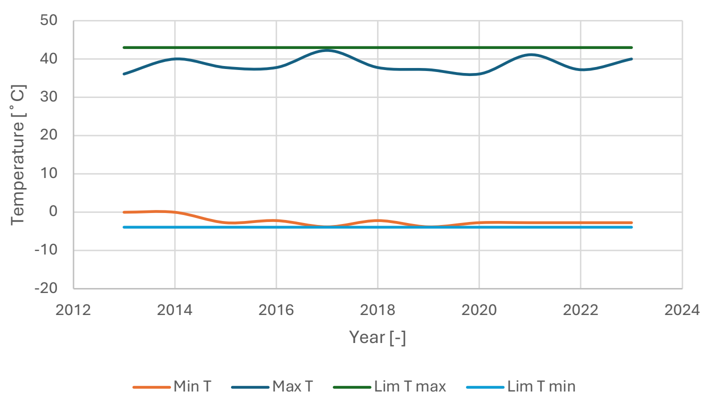


Figure 15: Behaviour of the maximum and minimum temperatures in Athes in the last ten years

After the working conditions of the ORC have been assessed, a model has been implemented on Aspen Plus in order to simulate the behaviour of the ORC in the conditions just found. This model is based on the basic architecture of the ORC described in Section 3.1.1. In the following the modifications applied to this model are reported. The boiler has been considered as it was constituted by three elements (theoretically, since in practice the heat exchanger is only one): the pre-heater, the evaporator and the super-heater. These are represented in the following scheme:

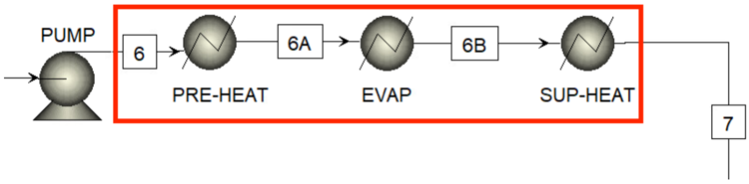


Figure 16: Modelling of the boiler

The boiler in the Figure 16 is constituted by the three elements inside the red rectangle. The temperature inside the evaporator has been set as the temperature at the outlet of the super-heater minus 1.5°C. This constant has been chosen since for values higher or equal to this it is obtained an expansion in the turbine that results in an outlet temperature from the turbine always higher than the condenser temperature in all the range of operating conditions of the ORC. The high pressure level of the cycle has been set as the pressure of the dry saturated vapour at the evaporator temperature. In the definition of the Aspen Plus model, the following variation has been done:

- for the pump an efficiency of 0.68 has been set, being this a more realistic value for the component than the one set in the modeling of the ORC developed in Section 3.1.1, where a really optimistic value for this parameter has been considered in order to carry out a preliminary study of the cycle;
- the efficiency of the turbine, comprehending both the mechanical and the isentropic contributions, has been set to 0.42. The reasoning is similar to the one developed in the previous point of the bulleted list for the efficiency of the pump;
- the boiler has been eliminated and substituted by three components: the pre-heater, the evaporator and the super-heater:
  - for the pre-heater it has been set a degree of subcooling equal to 0°C. This means that at the outlet of the pre-heater the fluid is in the saturated liquid phase;
  - for the evaporator it has been set a vapor fraction at the outlet equal to 1: this means that at the outlet of the evaporator the fluid is in dry saturated vapour conditions;
  - for the super-heater the temperature at the outlet has been set by varying it parametrically according to the sensitivity analysis that will be described in the following.

Regarding the "calculator blocks":

- the block "POUTBOIL" has been eliminated since the component "Boiler" has been eliminated;
- the block "DSUPHEAT" has been set, aiming at setting the power of the super-heater as the difference between the 40 kW released by the hot fluid and the sum of the powers of the pre-heater and the evaporator. This is equal to setting the total power exchanged at the boiler equal to 40 kW;
- in the block "PPUMP" the following operations are executed:
  - the temperature at the evaporator is set equal to the temperature at the outlet of the super-heater minus 1.5°C;
  - the pressure at the outlet of the pump, at the pre-heater and at the super-heater are set equal to the pressure at the evaporator.

A sensitivity analysis has been ran aiming at studying the behaviour of the ORC varying the temperature at the condenser and at the outlet of the super-heater in order to cover all the operating conditions. For each parameter the range of temperatures has been studied starting from the minimum value and incrementing it progressively of 1°C. The model on Aspen Plus is represented in the following scheme:

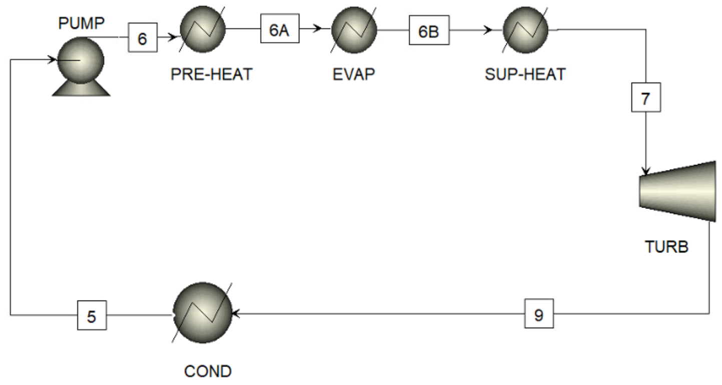


Figure 17: Modelling of the ORC aimed at studying all its operating conditions

3.1.4 Piping

The aim of the first part of this paragraph is to describe how the internal diameters of the pipes of the ORC system have been evaluated. First of all, in order to estimate the length of the pipe it has been necessary to make some assumptions regarding the dimensions of the components of the system. These are reported in the following scheme, that represent the layout of the ORC system. Note that the turbine and the generator in the scheme are represented as one block since the dimension of the former component are negligible with respect to that of the latter.

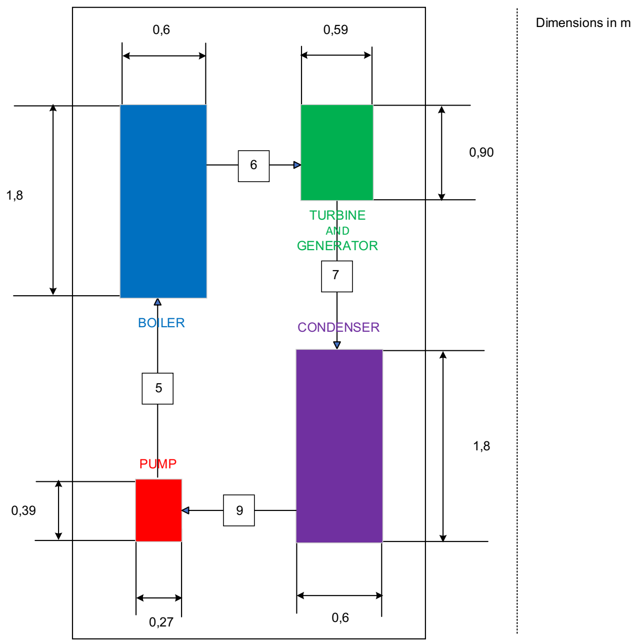


Figure 18: Space estimates of the ORC

Pipe 6 and 9 in the Figure 18 has been assumed to have a length equal to the sum of the width of the boiler and of the turbine and generator, which are assumed to be respectively 0.60 m and 0.59 m. The total length results consequently equal to 1.19 m for each of these pipes. For the pipes 5 and 7 it has been assumed a length equal to half of that said previously, namely 0.60 m.
For the mass flow rate it has been considered the value that allows to extract 40 kW from the hot fluid at the inlet of the boiler when the temperature difference between the condenser and the outlet of the boiler is minimum, namely when the temperature at the outlet of the boiler corresponds to 45° C and that at the

condenser to 43°C. This mass flow rate corresponds to 0.299 kg/s. In order to make a first estimate of the internal diameters of the pipes, the following procedure has been applied to each of them. The optimal velocity has been estimated for the pipe basing on the fluid phase (liquid or gas). In particular [35]:

- for the liquid phase an optimal velocity ranging between 1 m/s and 3 m/s has been considered;
- for the gas an optimal velocity ranging between 15 m/s and 30 m/s has been considered.

While performing calculations the values of velocities considered for the flow of the fluid in the two phases has been assumed as the average of the ranges just proposed. Regarding the density of the fluid the following reasoning has been applied. The pipes composing the ORC are four. Two of them contains fluid in liquid phase and two in gaseous phase. The density of the fluid in the liquid state has been evaluated as the average of the density of the fluid in the two pipes of the ORC where the liquid phase is found and an analogous procedure has been adopted to calculate the density of the fluid in the gaseous phase. Then, basing on the velocity, the mass flow rate and the density of the fluid, the internal diameter has been evaluated for the pipes containing gaseous and liquid phase as:

$$\dot{V} = \frac{\dot{m}}{\rho} \tag{1}$$

$$D_{in} = \sqrt{\frac{4 \cdot \dot{V}}{v \cdot \pi}} \tag{2}$$

The estimate of the internal diameter, as the evaluation of the volumetric flow rate, is based on the most critical condition in which the working fluid will operate, namely a condenser temperature of 43°C and an outlet boiler temperature of 45°C. The following step consists in the evaluation of the pressure drops in the pipes that, given the thermodynamic conditions of the fluid and the material by which the pipe is made, will be determined only by the internal diameter. The internal diameter evaluated in the first step of the procedure, by Equation 2, has been taken as initial value and then varied until values for pressure drops considered acceptable have been found. It has to be taken into account that, varying the internal diameter of a pipe, also the velocity in it, given the thermodynamic conditions and the pipe material, will vary. As a consequence, while trying to obtain acceptable pressure drops, attention has to be paid so that velocity remains in an acceptable range. The values that has been considered acceptable for the pressure drops are[35]:

- 0.5 kPa/m for pumped liquids;
- 0.02 per cent of line pressure for gases.

In the following the calculation methodology for the pressure drops in pipes is described. Input data required are:

- working pressure;
- working temperature;
- mass flow rate;
- density;
- phase;
- dynamic viscosity;
- molar weight of the fluid;
- roughness of the material constituting the pipes;
- internal diameter.

The former seven parameters indicated in the bulleted list has been evaluated basing on the thermodynamic model and considering the condenser temperature equal to 43 °C and the boiler outlet temperature equal to 45°C, namely considering the minimum temperature step between the condenser and the outlet of the boiler, since these are the conditions by which, aiming at extracting 40 kW from the hot fluid in the boiler, the highest mass flow rate will be needed, being the heat flux expressed as:

$$\Phi = \dot{m} \cdot c_p \cdot \Delta T \tag{3}$$

and being the specific heat, dependent on temperature, considerable in a first approximation constant for the small temperature difference assessed. The material chosen is INOX steel. The roughness of this material is equal to 0.00004572 m. It is to be underlined that, while varying the internal diameter in order to obtain acceptable values for the velocity and the pressure losses in the pipes, the nominal diameters established by the table from the ASTM are considered. This table is represented in the following:

	Schedule		Piping 5S		Piping 10S		Piping 40S		Piping 80S		Piping 160S	
			5		10		40		80		160	
Nominal	DN [mm]	Out.D [mm]	In D mm	Wall th. mm	In D mm	Wall th. mm	In D mm	Wall th. mm	In D mm	Wall th. mm	In D mm	Wall th. mm
1/8"	6	10,3	-	-	7,8	1,24	6,8	1,73	5,5	2,41	-	-
1/4"	8	13,7	-	-	10,4	1,65	9,2	2,24	7,7	3,02	-	-
3/8"	10	17,1	-	-	13,8	1,65	12,5	2,31	10,7	3,20	-	-
1/2"	15	21,3	18,0	1,65	17,1	2,11	15,8	2,77	13,8	3,73	-	-
3/4"	20	26,7	23,4	1,65	22,5	2,11	21,0	2,87	18,9	3,91	15,6	5,56
1"	25	33,4	30,1	1,65	27,9	2,77	26,6	3,38	24,3	4,55	20,7	6,35
1 1/4"	32	42,2	38,9	1,65	36,7	2,77	35,1	3,56	32,5	4,85	29,5	6,35
1 1/2"	40	48,3	45,0	1,65	42,8	2,77	40,9	3,68	38,1	5,08	34,0	7,14
2"	50	60,3	57,0	1,65	54,8	2,77	52,5	3,91	49,2	5,54	42,8	8,74
2 1/2"	65	73,0	68,8	2,11	66,9	3,05	62,7	5,16	59,0	7,01	53,9	9,53
3"	80	88,9	84,7	2,11	82,8	3,05	77,9	5,49	73,7	7,62	66,6	11,13
3 1/2"	90	101,6	97,4	2,11	95,5	3,05	90,1	5,75	85,4	8,08	-	-
4"	100	114,3	110,1	2,11	108,2	3,05	102,3	6,02	97,2	8,56	87,3	13,49
5"	125	141,3	135,8	2,77	134,5	3,40	128,2	6,55	122,2	9,53	109,5	15,88
6"	150	168,3	162,8	2,77	161,5	3,40	154,1	7,11	146,4	10,97	131,8	18,26
8"	200	219,1	213,6	2,77	211,6	3,76	202,7	8,18	193,7	12,70	173,1	23,01
10"	250	273,1	266,3	3,4	264,7	4,19	254,6	9,27	247,7	12,70	215,9	28,58
12"	300	323,9	316,0	3,96	314,8	4,57	304,8	9,53	298,5	12,70	257,3	33,32
14"	350	355,6	347,7	3,96	346,0	4,78	333,3	11,13	317,5	19,05	284,2	35,71
16"	400	406,4	398,0	4,19	396,8	4,78	381,0	12,70	363,5	21,44	325,4	40,49
18"	450	457,2	448,8	4,19	447,6	4,78	428,7	14,27	409,5	23,83	366,7	45,24
20"	500	508,0	498,4	4,78	496,9	5,54	477,8	15,09	455,6	26,19	408,0	50,01
22"	550	558,8	549,2	4,78	547,7	5,54	527,0	15,88	501,6	28,58	450,8	53,98
24"	600	609,6	598,5	5,54	596,9	6,35	574,7	17,46	547,7	30,96	490,5	59,54
26"	650	660,4	-	-	644,5	7,94	-	-	-	-	-	-
28"	700	711,2	-	-	695,3	7,94	-	-	-	-	-	-
30"	750	762,0	749,3	6,35	746,2	7,92	-	-	-	-	-	-

Figure 19: Dimensions of the pipes for various thicknesses and nominal diameters [10]

The Reynolds number is obtained as:

$$Re = \frac{\rho \cdot v \cdot D}{\mu} \tag{4}$$

The friction factor has been evaluated by using the Haaland formula, which is [36]:

$$\frac{1}{\sqrt{fric}} = -1.8 \log \left[\left(\frac{\epsilon/D}{3.7} \right)^{1.11} + \frac{6.9}{Re} \right] \tag{5}$$

The pressure drop due to the distributed losses is evaluated as [36]:

$$\Delta p = fric \cdot \frac{l}{D} \cdot \frac{\rho \cdot V^2}{2} \tag{6}$$

For each 90° curve a loss coefficient of 0.8 m has been associated.
In order to verify if the thickness considered in a first stage of the analysis was suitable to prevent mechanical damage at the working pressure, the minimum thickness required by the pipes has been evaluated. This has been done by using the following procedure. The input data are:

- external diameter;
- reference regulation;

- material;
- design temperature;
- design pressure;
- corrosion factor;
- quality factor.

The outer diameter has been chosen coherently with the dimensioning made regarding the pressure drops and the velocity. The regulation that has been taken into account is the EN 10217. This regulation is related to welded tubes for applications in pressure. The material chosen is the AISI 316L. Basing on the material and the regulation, it is possible to obtain:

- the yield stress with a permanent deformation of 1% of the total length;
- the ultimate tensile strength

for various values of temperature, by using proper tables [37]. The working temperatures and pressures considered for each pipe are higher than that for which the highest mechanical and thermal stress is achieved. For what concerns the design pressures, for each pipe the first value of this parameter higher than the maximum operating pressure of the fluid in the pipe in the following table has been considered:

PN2.5	PN6	PN10	PN16	PN20	PN25	PN40	PN50
PN64	PN100	PN150	PN160	PN250	PN320	PN400	PN420

Table 3.2: Nominal pressures [12]

For the design temperature values higher than the working temperature that determines the maximum thermal stress have been considered. The values used as design temperature and pressure in each pipe are represented in the following table:

Pipe	5	6	7	10
p_{workmax} [bar]	10.96	22.12	22.12	10.96
Design p [bar]	16	25	25	16
T_{workmax} [°C]	43	45	75	43
Design T [°C]	50	50	80	50

Table 3.3: Nominal temperatures and pressures for the ORC's pipes

The corrosion factor, related to the type of the fluid that flows in the pipe, has been assumed equal to 0.3 mm [35]. The quality factor, basing on what is indicated in the regulation EN 10217, has been considered equal to 0.7, taking into account the worst case scenario, namely a poor quality welding. Basing on parameters, material and regulation previously discussed, the maximum allowable stress has been evaluated as:

$$\sigma_{\text{allow}} = \max\left\{\frac{R_{P1.0}}{1.5}; \min\left[\frac{R_{P1.0}}{1.2}; \frac{R_m}{3}\right]\right\} \tag{7}$$

Basing on the Barlow's formula reported in the following, the minimum thickness has been evaluated:

$$t_{min} = \frac{p_{\text{design}} \cdot D_{\text{in}}}{2 \cdot \sigma_{\text{allow}} \cdot QF + p_{\text{design}}} + F_{\text{corr}} \tag{8}$$

3.1.5 Battery Limits

The battery limit is used in order to establish clear boundaries and responsibilities inside an industrial or process plant. In the following the battery limit is described. Regarding the data transmission the quantities reported in Figure 39 are given by a technical office and so they won't be analysed in detail. Regarding the electricity required in input 1.5 kW has been considered, of which 0.5 kW concerns the pump consumption in the case of maximum pressure step in the component, namely in the case in which the temperature at the condenser is equal to -4°C and that at the outlet of the boiler 75°C, while 1 kW takes into account the request from the automation system, as the electrically actuated valves and the auxiliary systems. Regarding the mass flow rate of hot fluid entering the boiler the maximum and minimum values are associated with the cases of minimum and maximum temperature step between the outlet of the boiler and the condenser respectively, namely to the cases in which the temperatures previously said are 45.00°C-43.00°C and 75.00°C-2.33°C. In order to evaluate these mass flow rates the following procedure has been adopted. Firstly, the thermodynamic cycle of the refrigerant fluid has been simulated in the two cases of interest. Then another model, still on Aspen Plus, has been developed, aiming at simulating the behaviour of the hot fluid. The layout of the model is represented in the following scheme:

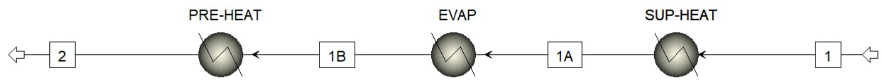


Figure 20: Layout of the hot fluid flow model

The hot fluid has been considered as glycolated water, represented on Aspen Plus as a mixture composed, in molar fraction, by 70% of water and 30% of propylene glycol. The three components represented in Figure 20, in the direction of the flow are: the super-heater, the evaporator and the pre-heater. The following data have been set on Aspen Plus:

- the temperature of flow 1 at the inlet of the super-heater has been set equal to the temperature at the exit of the boiler plus 5°C, in accordance to the hypothesis of minimum temperature approach of 5°C in the boiler;
- in the super-heater the pressure has been set to 4 bar and the heat exchanged has been given by the simulation of the same component in the thermodynamic cycle of the refrigerant;
- for the evaporator the pressure has been set to 4 bar and the heat exchanged has been given by the simulation of the same component in the thermodynamic cycle of the refrigerant;
- for the pre-heater the pressure has been set to 4 bar and the heat exchanged has been given by the simulation of the same component in the thermodynamic cycle of the refrigerant.

A "design specification" has been set aiming at varying the mass flow rate until the temperature of the hot fluid entering the pre-heater was higher by around 1°C than the temperature of the refrigerant exiting the pre-heater, which has been obtained by the simulation of the thermodynamic cycle of the working fluid. This aims at respecting the second law of thermodynamics, for which the heat flows, without any external work, from a higher temperature source to a lower temperature body and the vice-versa never occurs. The minimum and maximum volumetric flow rate have resulted to be respectively equal to 3000 L/h and 6500 L/h. The hot fluid feed temperature ranges, basing on the working conditions of the project, between 50°C and 80°C. As minimum design temperature a temperature lower than the minimum temperature registered in Athens in the last ten years has been considered, namely -10°C. As the maximum design temperature a temperature higher than 80°C has been considered, being this the maximum working temperature in the operating conditions of the ORC. The working pressure is 3 bar_g, corresponding, considering Table 3.2, to a design pressure of 6 bar. For the pipes of the hot water feed and return plate flange for welding pipes has been considered, according to the regulation EN ISO 1092-1. For the piping of the hot fluid analogous considerations has been done to that already exposed for the piping of the tubes where the refrigerant flows in the ORC.

In particular, for the analysis of the hot fluid feed pipe:

- the pressure drops and the velocity in the pipe have been studied in the situation of entering temperature in the super-heater equal to 50°C, namely in the case in which the mass flow rate is maximum;
- the thickness of the pipe has been studied in the case of highest design temperature, namely 100°C.

The indications regarding the diameter and the pressure of this pipe has so resulted to be DN40 and PN6. Regarding the hot fluid return pipes, the same considerations done for the feed hot fluid pipes are valid. In this case the minimum temperature is 41°C, corresponding to the case in which the temperatures at the outlet of the boiler and at the condenser are respectively 45°C and -4°C. The maximum temperature is instead 75°C, corresponding to the case of boiler outlet temperature and condenser temperature equal respectively to 75°C and 43°C. For the maximum and minimum design temperatures, the working pressure, the design pressure and the connection the same considerations done for the hot fluid feed pipe are valid. For the sizing of the hot fluid return pipe:

- the pressure drop and the velocity have been studied in the case of maximum mass flow rate, namely in the situation of temperature at the outlet of the pre-heater of 44°C;
- the thickness of the pipe has been studied in the case of maximum design temperature, namely 100°C.

For the electricity towards SYNAPSECOM it has been considered a maximum of 2.6 kW, deriving from the simulations developed on Aspen Plus in the different working conditions.

3.1.6 Fluids List

A fluid list refers to a technical document that compiles and lists all the fluids used within a particular system or process. In the fluid list it is indicated the fluid type and phase, the composition, the design and working temperatures and pressures and the material compatibility of the various fluids. The aim of the fluid list in process engineering is to provide essential data about the fluids in the system, supporting safe design, operation, and maintenance while ensuring regulatory compliance. The fluids that work in the ORC are:

- 2,3,3,3-Tetrafluoropropene. This fluid is flammable and works both in the liquid and in the gaseous phase. The working temperatures are in the range -4°C (lowest temperature at the condenser) and 75°C (highest temperature at the outlet of the boiler). The minimum and maximum design temperatures are -10°C and 100°C respectively and they have been assigned considering a temperature slightly lower than the minimum working temperature and a temperature slightly higher than the maximum working temperature. The design pressure is 25 bar, assigned considering that the maximum working pressure is 22.12 bar and the Table 3.2. The working pressure ranges between 1.70 bar_g and 21.11 bar_g, approximated in Figure 40 to 2 bar_g and 23 bar_g respectively. A process insulation is required for the pipe containing this fluid;
- glycolated water. This mixture is not flammable and works only in the liquid phase. The composition, considering the molar fractions, is 70% water and 30% propylene glycol. Regarding the design temperatures, the working temperatures, the design pressures and the working pressures, the same considerations done in the Section 3.1.5 are valid. A process insulation is required for the pipes containing this fluid.

In the following table it is indicated the compatibility of the fluids utilized with several materials. The compatibility is classified in four types, each of them is characterized by a letter:

- A: minor effects;
- B: moderate effects;
- C: major effects;
- D: not recommended.

Material	2,3,3,3-Tetrafluoropropene	70% water+30% propylene glycol
303 SS	A	A
304 SS	A	A
440 SS	A	A
Aluminium	A	A
Titanium	A	A
Hastelloy C.	A	A
Cast bronze	A	A
Brass	A	A
Cast Iron	B	B
Carbon Steel	B	B
PVC	A	A
Teflon	A	A
Nylon	A	A
Polyethylene	A	A
Polypropylene	A	A
Carbon	A	A
Ceramic	A	A
Viton	A	A
Buta N	B	A
Silicone	A	A
Neoprene	A	B
Ethylene Propylene	A	A
Natural rubber	D	D

Table 3.4: Compatibility of ORC fluids with several materials, [13], [14]

3.1.7 Block Flow Diagram

A block flow diagram is a simple, graphical representation of a process or system, depicting the major steps or blocks involved and the flow of materials or energy between them. In the control volume of the ORC enter:

- hot fluid coming from the servers;
- hot fluid coming from the TES;
- electricity;
- data;

and exits:

- hot fluid towards the servers;
- hot fluid towards the TES;
- electricity;
- data.

In the BFD the company involved in the development of the ORC is represented.

3.1.8 Process Flow Diagram

A process flow diagram is a graphical representation of a process or system that shows the sequence of steps or operations involved. In the PFD of the ORC all the components that constitute the system are represented:

- the pump;
- the boiler;
- the turbine;
- the electric generator;
- the condenser;
- the refrigerant collector tank.

It is to be noted how in Figure 42 the turbine and the generator are numbered both with the code 103, since they are in the same loop. The liquid refrigerant collector tank at the outlet of the condenser is not strictly required, but it can be advantageous for several reasons:

- operating stability: it can supply a buffer that helps maintaining a continuos and stable provision of working fluid, even when there are variations in the flow rate and temperature at the inlet of the condenser. This can improve the reliability and the stability of the entire cycle;
- management of the load variations: it can help managing the variations in the thermal load and the operating conditions;
- maintenance and security: during maintenance or in emergency situations, it can work as a working fluid reserve, allowing to isolate parts of the system without interrupting the cycle completely;
- fluid level control: it allows mainteining a precise control on the level of the fluid in the cycle, which is crucial in order to avoid problems like cavitation in the pumps or the accumulation of fluid in excess in undesired parts of the system;
- impurity management: in some cases, it can also facilitate the removal of impurities or the separation of the different phases, improving the efficiency of the cycle and reducing the maintenance costs.

3.1.9 Mass and Energy Balance

In process engineering, mass and energy balances are fundamental concepts used to analyze and design chemical, mechanical, and industrial processes. They help engineers track the flow of materials and energy in a system to ensure that inputs, outputs, and transformations are properly accounted for. The mass and energy balance has been made considering the average temperature in Athens in the last ten years [34], equal to around 18°C, as the condenser temperature, while 45°C as the temperature at the outlet of the boiler.

3.1.10 Process Data Sheet

A process data sheet gives specific and detailed pieces of information regarding the components of a system used in an industrial process. The main aims of a PDS are:

- to describe the technical specifications: it contains technical details such as the dimensions, the materials used for the construction and the temperature and pressure requirements of the components;
- to guide the design: it gives pieces of information necessary for the design of the components, assuring that all the requirements of the industrial process are satisfied;
- to assure the compliance with the regulations: it helps assuring that the components respect the relative regulations as well as the safety standards;
- facilitate the communication: it serves as a common reference among the various engineers team, assuring that all have a uniform comprehension of the technical specifications;

- support the selection and the buying: it gives crucial pieces of information that helps in the selection of the suppliers and in the preparation of the offers, specifying exactly what is requested.

In the following the process data sheet for each component of the ORC will be analyzed in detail.
Regarding the inlet and outlet flows of the pump three cases have been assessed regarding the main parameters, namely the mass flow rate, the pressure, the temperature, the pressure ratio, the work required and the head: the normal (NOR), minimum (MIN) and maximum (MAX) operating conditions. These conditions referes to, respectively:

- NOR: it corresponds to a temperature at the condenser of $18°C$ and at the outlet of the boiler of $45°C$;
- MIN: it refers to the conditions for which the pump requires the minimum work, namely the case for which the temperature at the condenser is $43°C$ and the temperature at the outlet of the boiler is $75°C$;
- MAX: refers to the conditions for which the pump requires the maximum net power, namely the case in which the temperature at the condenser is $21°C$ and the temperature at the outlet of the boiler is $75°C$.

The operating conditions corresponding to MIN and MAX have been deduced from the results of the Aspen Plus model of the ORC in all the operating conditions, in particular assessing the power map of the pump in all the working range. This map is represented in the following:

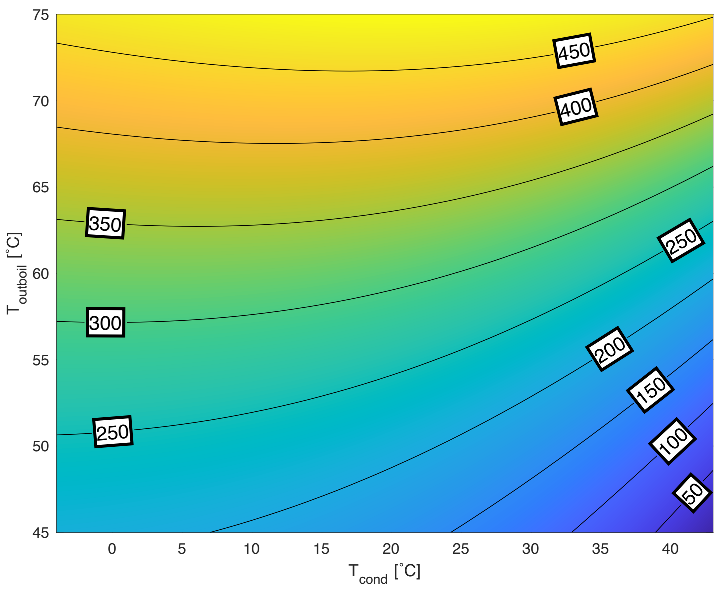


Figure 21: Power map of the pump

The head in the various operating conditions has been evaluated using the following formula:

$$H = \frac{p_2 - p_1}{\gamma} + y \tag{9}$$

where y is given by the Stevin formula [38], namely:

$$y = \frac{\Delta p}{\rho g} \tag{10}$$

It is to be noted how, regarding the contribution of the pressure drops in the evaluation of the head, the conditions of maximum pressure drop in the pipes are always assessed, corresponding to the case of maximum mass flow rate achieved for a temperature at the condenser of $43°C$ and a temperature at the outlet of the boiler of $45°C$ (conservative hypothesis). Anyway the contribution of this quota is small compared to the contribution

of the pressure increases.
The specific gravity, viscosity, specific heat and thermal conductivity are reported only in normal conditions. The efficiency has been considered constant in all the working conditions. Regarding the materials for the casing, the impeller and the flanges, AISI 316 has been chosen as a first instance, while for the gaskets the VITON. Regarding the power supply 220V and 50 Hz have been considered, according to the European standard. The values of maximum and minimum design temperature and pressure are slightly higher than the relative maximum values and slightly lower than the relative minimum ones in order to guarantee the correct functioning of the plant in the expected working conditions. For the temperatures $\pm 20°C$ has been summed to the maximum and minimum values, while for the pressures the first higher value in the Table 3.2 for the case of maximum pressure and the first lower value in the same table for the case of minimum pressure have been chosen.
The ATEX classification is the following: II 3G IIB T3. In the following the explanation is shown:

- II: group of the component (for environments with gas, vapor and fog);
- 3G: category of the component (zone 2, gas);
- IIB: explosion group;
- T3: temperature class (minimum autoignition temperature of the gas ranging between $200°C$ and $300°C$). The autoignition temperature of R1234yf has been assessed at 22 bar, being this the highest pressure at which the system works, namely the working condition associated to the minimum autoignition temperature, equal to around $250°C$ [4]. This classification suggests that the maximum allowable surface temperature of the system must be $200°C$.

This component should be compliant with the PED directive. The PED directive (Pressure Equipment Directive) is an European regulation regarding the design, production and conformity of the pressure equipments. A component of a plant is compliant with the PED classification if it respects the requirements established by this directive. The type of pump chosen as a first instance is the multistage centrifugal pump. This type of pumps are ideal for the applications that requires a significant increase of the pressure in the liquid phase. They are often used in refrigerant fluid transfer systems, where it is necessary to manage an increase in pressure with a minimum impact on the temperature of the fluid. These pumps offer a good energy efficiency, in particular when working with moderate mass flow rate as in the case study. Furthermore, the multistage centrifugal pumps offer a significant increase in pressure in a gradual and controlled manner, reducing the wear to the minimum and maintaining a long operational life. Moreover, these pumps are robust and reliable, ideal for systems that operates continuosly.
Regarding the inlet and outlet flows of the boiler the case corresponding to the normal operating conditions has been assessed, namely the situation in which the temperature at the condenser is $18°C$ and the temperature at the outlet of the boiler is $45°C$. It is to be noted that the power exchanged at the boiler is constant and equal to 40 kW for all the operating conditions, since the mass flow rate in the ORC is varied basing on the temperature at the condenser and on the temperature of the hot fluid entering the boiler, with the aim of maintaining the power exchanged constant indeed. It is also to be noted that the hot fluid has been considered as completely coming from the data center. This is not always true since varying the working conditions of the integrated system the hot fluid entering the boiler could come from TES, both partially or completely. Anyway, even varying the source of the hot fluid, the working conditions at the inlet of the boiler are the same and consequently the design can be done basing on this conditions, neglecting the provenance of the hot fluid. Regarding the materials for the plates and the frame it has been chosen as a first instance AISI 316, while for the gaskets VITON. Regarding the design values of pressure and temperature the same considerations already developed for the pump are valid. It is to be noted that in this case the design pressure considered is only one since the boiler is supposed to be isobar. In particular the maximum pressure is taken into account, for which the mechanical stress will be the highest. The boiler works in countercurrent. The ATEX classification is the same of the pump. The considerations regarding this are the same done for the previous component analysed, apart from that regarding the temperature class. In this case infact also the temperature class of the mixture of propylene glycol and water has to be assessed since this fluid flows on one side of the plate heat exchanger. The autoignition temperature of R1234yf anyway determines the temperature class of this component since the hot fluid is not flammable. This component should be compliant with the PED directive. Regarding the type of heat exchanger to be used as a first instance a plate heat exchanger has been chosen, because of:

- high heat transfer efficiency: thanks to the big contact area between fluids, this type of heat exchanger is very efficient in the heat transfer, that is useful considering the small temperature difference between the two fluids in the case study;
- compact dimensions: this type of heat exchanger have compact dimensions, which allows to save space;
- pressure resistance: this kind of heat exchanger is able to manage relatively high pressures, as those of the gas flow in the case study.

Regarding the inlet and outlet flows of the turbine three cases have been considered for the main parameters, namely the mass flow rate, the pressure, the temperature and the power produced: the normal (NOR), minimum (MIN) and maximum (MAX) operating conditions. These corresponds to:

- NOR: corresponds to a temperature at the condenser of 18$°C$ and a temperature at the outlet of the boiler of 45 $°C$;
- MIN: it refers to the conditions in which the turbine produces the minimum power, namely the case in which the temperature at the condenser is 43 $°C$ and the temperature at the outlet of the boiler is 45 $°C$;
- MAX: it refers to the conditions for which the turbine produces the maximum power, namely the case in which the temperature at the condenser is -4 $°C$ and the temperature at the outlet of the boiler is 75 $°C$.

These values have been taken from the results of the Aspen Plus model of the ORC in all the operating conditions, in particular from the power map of the turbine in all the working range. This is reported in the following:

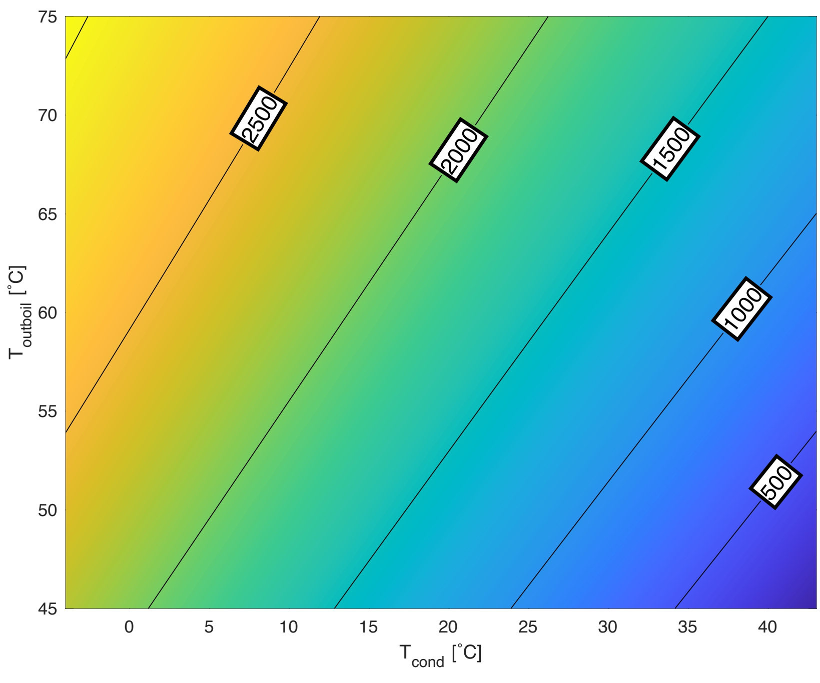


Figure 22: Power map of the turbine

The specific gravity, viscosity, specific heat and thermal conductivity have been reported only in normal operating conditions. The efficiency has been considered constant in all the operating conditions. Regarding the material for the spirals, the body and the shaft it has been considered AISI 316, while for the gaskets VITON. Assessing the design values the same considerations done for the analysis of the pump are valid. The ATEX classification is the same as done for the pump and also the same consideration regarding it are valid. This component should be compliant with the PED directive.
Regarding the type of turbine as a first instance a scroll expander has been chosen, since:

- they are efficient at handling low to moderate pressure differentials, which aligns with the pressure drops in the case study;
- they often achieve high isentropic efficiencies, particularly in applications with working fluids like R1234yf;
- they work well with vapor-phase refrigerants, as in the case study.

Regarding the electric generator the efficiency has been reported, which has been considered constant and equal to 0.85, and the values of mechanical power at the inlet of the shaft and electrical power at the outlet of the generator in the three cases corresponding to the operating conditions NOR, MIN and MAX described previously for the turbine.
Regarding the inlet and outlet flows of the condenser three cases for the main parameters, namely the mass flow rate, the pressure, the temperature and the power exchanged, have been considered: the normal (NOR), minimum (MIN) and maximum (MAX) operating conditions. These corresponds to:

- NOR: temperature at the condenser of 18°C and at the outlet of the boiler of 45°C;
- MIN: conditions for which the absolute value of the power exchanged is minimum, namely the case in which the temperature at the condenser is -4°C and the temperature at the outlet of the boiler is 75°C;
- MAX: conditions for which the absolute value of the power exchanged is maximum, namely the case in which the temperature at the condenser is 43°C and the temperature at the outlet of the boiler is 45°C.

These values have been taken from the results of the model of the ORC on Aspen Plus in all the operating conditions, in particular from the power map of the condenser in all the working range. This is reported in the following:

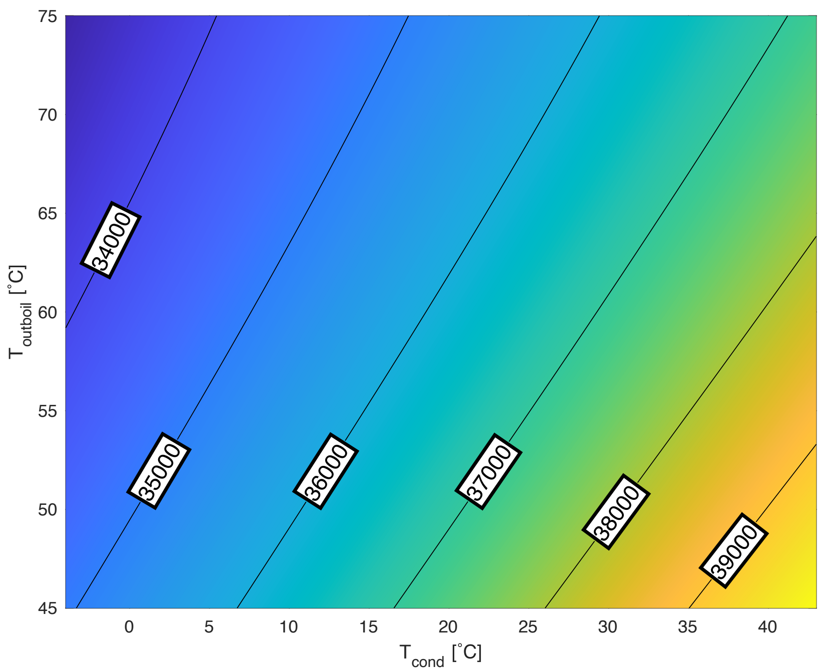


Figure 23: Power map of the condenser

The specific gravity, viscosity, specific heat and thermal conductivity are reported only in the normal operating conditions. Regarding the materials for the pipes, the fans and the frame it has been chosen as a first instance AISI 316. Regarding the design values the same considerations done for the analysis of the boiler are valid. Regarding this component the ATEX classification is not necessary. The air cooled condensers work cooling down the fluid without any direct contact with the air in such a way to reduce the probability of explosive

atmospheres. This component should be compliant with the PED directive.
Regarding the refrigerant collector tank the normal operating conditions have been considered, namely that corresponding to a temperature at the condenser of 18°C and a temperature at the outlet of the boiler of 45°C. While designing the refrigerant collector tank firstly the total volume of fluid contained in the pipes, given the internal diameters and the length of each of them, has been evaluated. Subsequently, a multiplying factor of 1.2 has been applied in order to be conservative. An height of the tank equal to 0.2 m has been considered and basing on this it has been evaluated the internal diameter of the tank. The calculations are reported in the following tables:

Pipe	D_{in} [m]	Length [m]	Volume [m^3]
5	0.018	0.6	0.000153
6	0.018	1.19	0.000303
7	0.0234	0.6	0.000258
9	0.0234	1.19	0.000512

Table 3.5: Pipes' volume evaluation

total volume pipes [m^3]	0.001
design volume tank [m^3]	0.001
design volume tank [L]	1.5
height [m]	0.2
cross section [m^2]	0.007
D_{in} [m]	0.1

Table 3.6: Sizing of the refrigerant collector tank

The unloading time has been evaluated knowing the mass flow rate, the density and the volume of fluid contained in the tank. The minimum thickness required for the tank has been evaluated with the same method used to evaluate the minimum thickness of the pipes of the ORC. Regarding the toroidal ends of the tank the minimum thickness has been evaluated as described in the following. First of all the parameter β has been evaluated from the following graph:

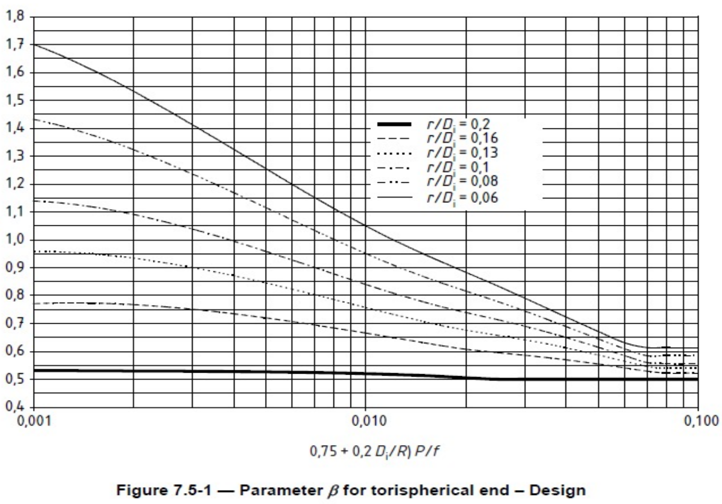


Figure 24: Parameter β for torispherical end - Design [11]

In order to evaluate β it is necessary to know the ratio between the internal diameter and the radius of curvature of the torispherical end ($\frac{D_i}{R}$), the design pressure (p_{design}) and the maximum allowable stress (f). The ratio between the internal diameter and the radius has been fixed equal to 10, while the other parameters have been calculated analogously to what has been done for the evaluation of the thickness of the pipes of the ORC. The other parameters necessary for the calculation of the minimum thickness are the internal diameter, the quality factor and the factor that takes corrosion into account, which has been evaluated analogously to what has been done for the evaluation of the thickness of the pipes of the ORC. Finally, the minimum thickness of the torispherical end has been calculated as [11]:

$$t = \frac{p_{\text{design}} \cdot D_{\text{in}}}{2 \cdot \theta_{\text{allow}} \cdot \beta \cdot QF + 0.2 \cdot P_{\text{design}}} + F_{\text{corr}} \tag{11}$$

The values of the minimum thicknesses for the cyclindrical body and the torispherical end are reported in the following table:

Minimum thickness of the cylindrical body [mm]	1.6
Minimum thickness of the torispherical end [mm]	1.5

Table 3.7: Minimum thicknesses of the refrigerant collector tank

For the design temperature and pressure the maximum and minimum values of these parameters in the working range have been considered and the same considerations done in the case of the pump has been applied. The orientation of the tank is vertical. Regarding the material, as a first instance, for the body and the torispherical ends AISI 316 has been chosen. This component requires a safety valve which is to be opened for pressure levels higher than 25 bar, since this is the value of pressure basing on which the minimum thickness of the tank is designed. The Atex classification is II 3G IIB T3 and the same considerations done for the Atex classification of the pump are valid. The tank is compliant with the SEP (Sound Engineering Practice), basing on the European directive on the pressure equipments (PED), since the product between its volume and the maximum allowable pressure is equal to 37.5 $bar \cdot L$.

3.1.11 Mechanical Data Sheet

The mechanical data sheet is a fundamental device to assure that all the mechanical and performance details of a component are carefully defined, documented and understood by all the parties involved in the project. The difference between a PDS and a MDS is explained in the following:

- MDS: it deals with the mechanical and structural specifications of a component;
- PDS: it deals with the operational and process specifications of a component inside a production system.

The mechanical data sheet is assessed only for the refrigerant collector tank, which will be built in home by the company where this thesis has been developed. Assessing the MDS of the refrigerant collector tank all the pieces of information regarding this component already discussed in the process data sheet has been taken into account. In addition to this, regarding the connections, the following types have been considered for the various inlet and outlet pipes:

- loading and purging line: threaded connections for ease of use and accessibility;
- inlet and outlet line of the working fluid during the functioning: flanged connections for higher safety, resistence and ease of maintenance.

For the loading line of the refrigerant fluid the values of nominal pressure and diameter of the pipe exiting the condenser have been taken into account. For the purging line the diameter of the pipe in which the gas flows, obtained as described in the Section 3.1.4, and the nominal pressure corresponding to the maximum pressure in the tank have been taken into account. Regarding the inlet and outlet lines of the refrigerant collector tank, for the nominal pressures the maximum value of the pressure in the connected line has been assessed and then the first higher value present in Table 3.2 has been selected, while for the nominal diameter the same nominal diameter of the connected lines has been considered.

3.1.12 Equipment List

An equipment list is a detailed inventory of the tools, machinery, devices or other items needed for particular tasks, project or operation. It helps ensure that all necessary tools are available and in proper working condition, aiding in smooth operation and planning. In the equipment list for every equipment constituting the system it is indicated: the ID, the service, the installed power, the pressure at inlet in normal conditions, the availability for heating, cooling and electricity production and additional notes when needed.

3.1.13 Instrument List

In process engineering an instrument list is a comprehensive document that details all the instruments used within a particular plant. It is used by engineers, technicians, and operators to understand the instrumentation involved and to facilitate troubleshooting, maintenance and upgrades. The parameters to be measured by the measuring instruments in the ORC are:

- the pressure:
  - at the outlet of the pump;
  - at the inlet of the turbine;
  - at the outlet of the turbine;
- the temperature:
  - at the inlet of the boiler;
  - at the outlet of the boiler;
  - at the inlet of the condenser;
  - at the outlet of the condenser;
- the level of fluid in the refrigerant collector tank.

The pressure is measured with the "gauge", the temperature with the "PT100" and the level with the "Level switch low". For each instrument it is indicated: the design temperature and pressure, the fluid whose parameters are to be measured and the range of values in which the instrument must be able to operate. The instruments transmit a signal by a current which has an intensity in the range 4-20 mA. 4 mA is associated to the minimum value measurable, while 20 mA is associated to the maximum value measurable. Regarding the connections: for each instrument they are threaded and have a diameter equal to 1/4" M NPT. This means:

- 1/4": this indicates the nominal dimension of the pipe. Even if it is indicated as 1/4" the effective diameter is different from the nominal measure;
- M: it stands for "male", indicating that this is an external thread;
- NPT: National Pipe Tapered, a conical thread standard.

In order to establish the nominal pressures, the maximum working pressures of the fluid in the zone where the instrument operates have been taken into account and in particular the first pressure higher than the maximum one in Table 3.2 has been selected. Regarding the level switch low it is to be underlined how this instrument doesn't indicate the absolute value of the refrigerant level in the tank, but it is limited to indicate when this fall below a certain threshold.

3.1.14 Valve List

In process engineering a valve list is a comprehensive document or database that details all the valves used in a particular system. A valve list is essential for process engineers because it helps in designing, operating and maintaining process system efficiently. It ensures that the right type of valve is used for specific applications and helps in troubleshooting and upgrading systems. The main aims of the valves in the ORC are to allow the loading of the refrigerant fluid in the system and the maintenance. Two different types of valve are used for these specific purposes. In order to allow the loading of the refrigerant fluid in the system a needle valve is used.

In order to perform maintenance ball valves are used to isolate the component which requires maintenance and needle valves are used to purge the pipe of the component that requires maintenance from air and liquid. In the following the two types of valves just presented will be discussed more in detail. A needle valve offers a precise control of the flow thanks to its structure with a small opening and an adjustable conical piston. It is ideal for fine adjustments and low flow rate fluxes. It allows an accurate control while filling the system. Ball valves consist of a spherical ball with a hole through the middle, which is mounted in a valve body. A ball valve provides a complete and reliable shutoff when closed. The ball inside the valve has a solid, smooth surface that fits tightly against the seats, ensuring that no fluid pass through when the valve is in the closed position. This makes them highly effective for isolating sections of a system. Regarding the positioning of the valves:

- ball valves (isolate): they must be installed on both sides of the component to be isolated. This permits to isolate completely the component from the system during the maintenance;
- needle valves (purge): two for each component. One is to be installed on the highest point of the component in order to remove the air and one on the lowest point of the component in order to remove the liquid;
- needle valve (loading): to be installed on the lateral surface of the refrigerant collector tank.

For each valve it is indicated the type, the connection, the nominal diameter and the nominal pressure of the connection, the material by which the body of the valve is constituted, the material by which the gasket is constituted, the material for the wet parts, the type of fluid in contact with the valve, the nominal working temperature and pressure and the type of actuation. The connection is performed by compression for each valve. This has the following advantages:

- ease of installation: it doesn't require any welding or thread, making the installation quick and simple;
- maintenance: the compression connection can be easily disassembled and assembled, facilitating the maintenance;
- versatility: suitable for a variety of materials and dimensions of the pipes.

The internal diameter is assumed equal to that of the pipe in which the fluid flows. The nominal pressure is selected considering the first pressure higher than the maximum operating pressure in the pipe in Table 3.2. As a first instance AISI 304 has been chosen for each valve's body and Ethylene-Propylene Diene Monomer (EPDM) for each valve's gasket. For each wet part AISI 304 has been chosen. The fluid in contact with the valves is 2,3,3,3-Tetrafluoropropene. The nominal temperatures and pressures are refered to the conditions in which the ambient temperature is 18 $°C$ and the temperature of the hot fluid at the inlet of the boiler is 50 $°C$. In the case of the analysis of the needle valves it is considered, among the inlet and the outlet, the condition which is associated with the highest temperature and pressure. The actuation is manual.

3.2 Analysis of the integrated system

The integrated system is composed by a DHW tank, a SH system, a TES, an ORC, a heat dissipation fan and a circuit where hot water is circulated by a pump and extract heat from the dielectric fluid that cools the data center.

3.2.1 Working logic

The idea behind the logic of the integrated system is that of using the TES as a storage to compensate the oscillations of power coming from the data center. Regarding the domestic hot water tank and the space heating system, it can help better exploit the energy transported by the hot fluid by storing it when in excess and use it when in lack. The ORC is designed with a flow rate so that it can receive 40 kW by the hot fluid in the boiler. Being the aim of the servers cooling system that of maintaining them at a constant temperature, and being that these heats up while working in a way such that this effect varies with time, it follows that the heat power extracted from them is variable with time. Since in order to work the ORC requires a power constant and equal to 40 kW, the oscillations of power in the hot fluid circuit for which values of power higher than this are achieved will be exploited sending 40 kW to the ORC and charging the storage with the remaining part (charging of the TES). In the moment in which the power transported by the hot fluid circuit is lower than 40 kW, the power coming from the storage will be summed to this (discharge of the TES), if available, in order to reach the 40 kW necessary for the ORC to work correctly. This concept is graphically explained in the following:

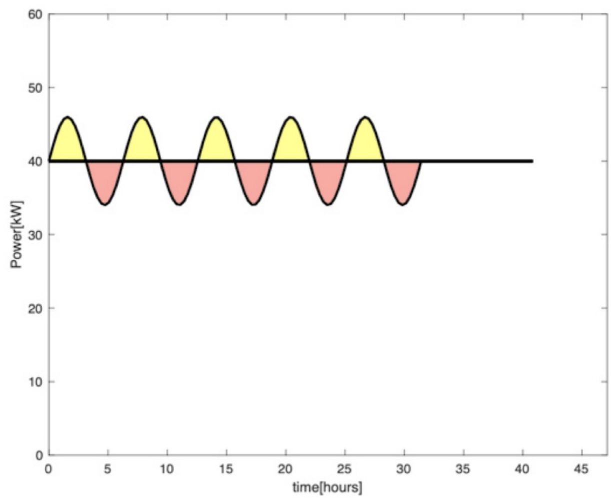


Figure 25: Management of power coming from the data center

The yellow ares in the graph indicates the moments in which the power transported by the hot fluid circuit is higher than the power required by the ORC (charging of the TES), while the red ones indicates the moments in which the power transported by the hot fluid circuit is lower than the power required (discharge of the TES). Furthermore it is to be noted that the TES is not considered for the seasonal storage, namely it is not used to store heat in summer and then reuse it in winter, but it is for low-time storage. This is because of the charging and the discharging times of the storage, that are a consequence of its phycal-chemical characteristics. Another important consideration is that the five system constituting the project (heat dissipation circuit, DHW, SH, TES and ORC) works respecting a hierarchical order, in which the priority is given first to the DHW, than to the SH, then to the ORC, then to the TES and finally to the heat dissipation circuit.

The hierarchical order in which the different systems are supplied is represented in the following scheme:

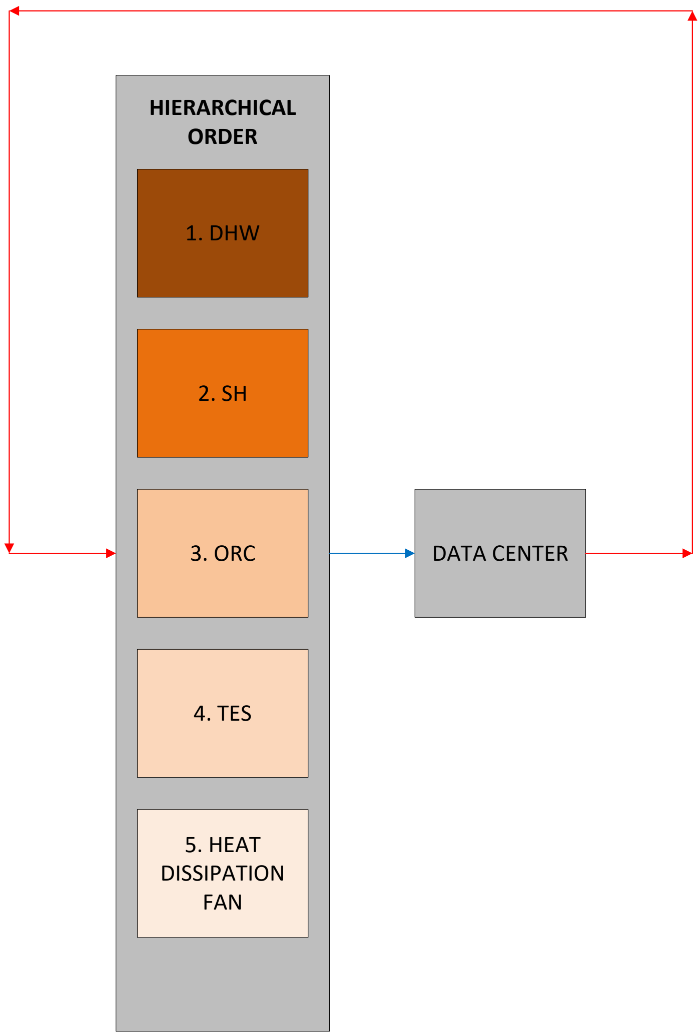


Figure 26: Hierarchical order of the IS

In the following it is described how the logic of the integrated system works. It is to be noted that the DHW and the SH systems will be analysed as one in a first moment and the separation will be assessed later in the following reasoning. Also, it is to be noted that while discussing the power coming from the servers, this has an absolute value that does not depend from the interaction of it with the various systems constituting the IS, since the temperature of the hot fluid sent to the IS can be measured, its mass flow rate can be measured too and the temperature of the hot fluid returning to the servers is set by the owners of the data center. Finally, it is to be noted that at each step the value of power coming from the TES and power required from the DHW+SH system are assumed to be absolute values, not depending from the interaction with the various systems. While analyzing the working logic of the IS, firstly it is to be said that the time is assumed as divided in several steps that last a short amount of time. At each step the logic of the IS operates by evaluating the conditions of the system and establishing the consequent operations. In the several subsequent time steps the logic operates performing the actions just said, which form a loop. At the beginning of each loop the power demand of the DHW+SH system is evaluated. Then a comparison between the power coming from the servers and the power demand from the DHW+SH system is carried out:

- if the former is higher than the latter, the power remaining from the supply of the DHW+SH system is calculated ($P_{1,remain}$) and then:
  - a power equal to the power demand of the DHW+SH system is supplied to this;
  - the following condition is evaluated: "The TES at the previous loop was charging or the State of Charge of the TES (SOC_{TES}) is lower than 30% and the ORC was OFF in the previous loop".
    - if the condition is true: the ORC is bypassed;
    - if the condition is wrong: a comparison between the sum of $P_{1,remain}$ and the power coming form the TES and the minimum power required by the ORC in order to be able to work is draw:
      - if the former is higher or equal than the latter:
        1. if the sum of $P_{1,remain}$ and the power coming from the TES is higher or equal than 40 kW: the power coming from the TES necessary, while summed to $P_{1,remain}$, to give 40 kW is evaluated and then the sum of this power and $P_{1,remain}$ is sent to the ORC;
        2. if the sum of $P_{1,remain}$ and the power coming from the TES is lower than 40 kW: the sum of $P_{1,remain}$ and the power coming from the TES is sent to the ORC;
      - if the former is lower than the latter the ORC is bypassed;

    Then $P_{2,remain}$ is evaluated as the difference between $P_{1,remain}$ and the power sent to the ORC. Subsequently, the following condition is evaluated: "$P_{2,remain}$ is higher than 0 kW and the TES is not fully charged":
    - if this condition is satisfied: $P_{2,remain}$ is sent to the TES;
    - if this condition is not satisfied: the TES is bypassed;

    Then $P_{3,remain}$ is evaluated as the difference between $P_{2,remain}$ and the power coming from the TES. Furthermore:
    - if $P_{3,remain}$ is higher than 0 kW: this power is sent to the Heat Dissipation Fan;
    - if $P_{3,remain}$ is equal to 0 kW: the Heat Dissipation Fan is bypassed;
- if the former is lower or equal than the latter:
  - if the sum of the power coming from the servers and the power coming from the TES is higher or equal than the power demand from the DHW+SH system: the power required from the TES that summed to the power coming from the servers gives the power demand of the DHW+SH system is evaluated and then the sum of this power and the power coming from the servers is given to the DHW+SH system;
  - if the sum of the power coming from the servers and the power coming from the TES is lower than the power demand from the DHW+SH system: the sum of the power coming from the servers and the power coming from the TES is given to the DHW+SH system.

The working logic of the IS, which has just been described, is graphically represented in the following flow chart:

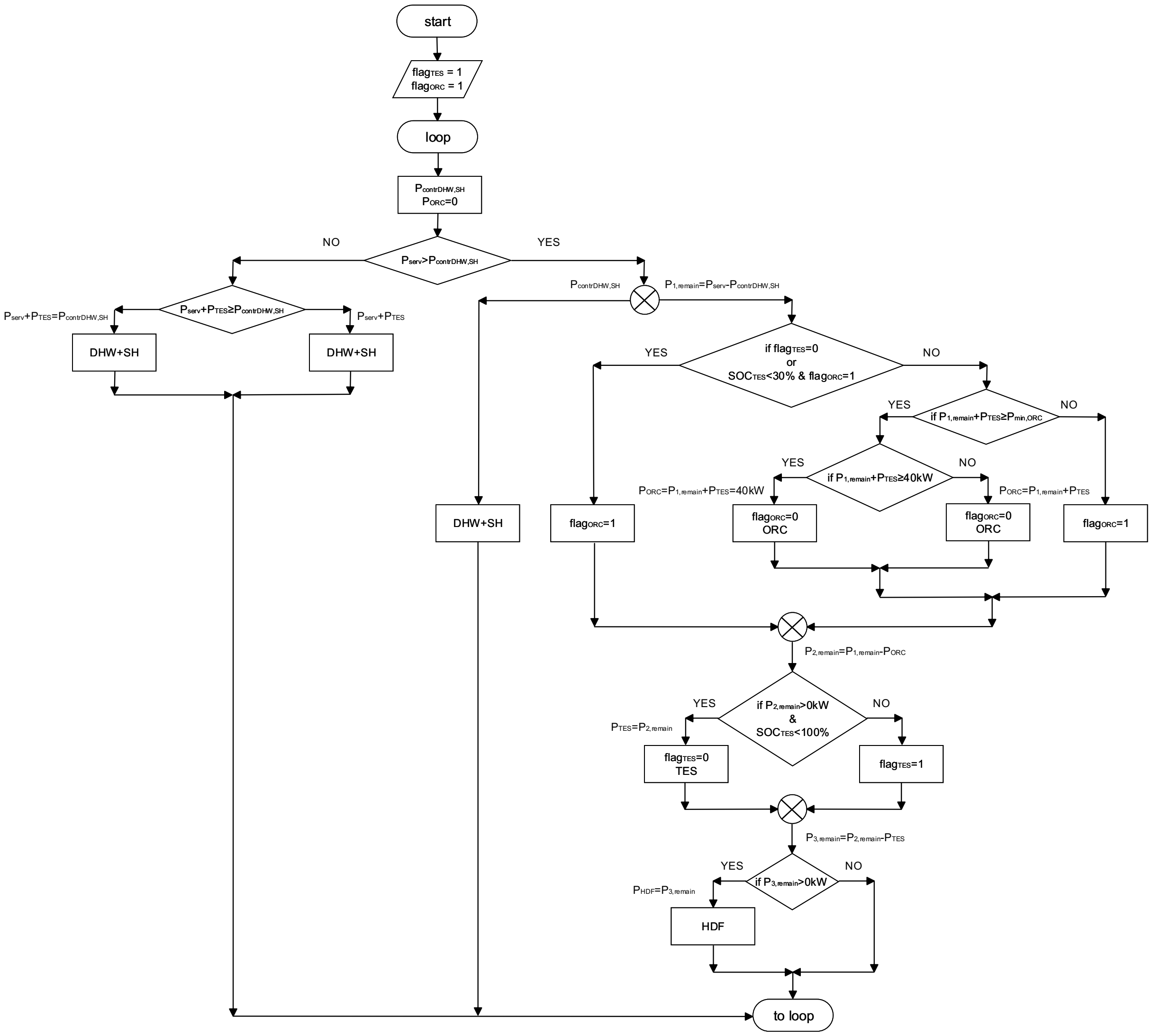


Figure 27: Flow chart of the working logic of the IS

Regarding the system composed by the DHW+SH, the DHW has the priority above the SH.

3.2.2 Battery Limits

Regarding the battery limits of the integrated system the same considerations made for that of the ORC in Section 3.1.5 are valid. In addition to this the consumption of the auxiliary systems has to be taken into account, being this constituted of: the temperature meters in the DHW, SH and TES, the valves (gate and three-way), the temperature and flow meter of the hot fluid flow and the circulation pumps.

The values considered for the power consumption of each component are represented in the following table:

Instrument	Quantity	$W_{\rm el}$ per Component [W]	W_{el} per Component Class [W]
Temperature sensor in DHW	1	0.1	0.1
Temperature sensor in SH	1	0.1	0.1
Temperature sensor in TES	1	0.1	0.1
Gate valve actuator	6	50	300
Three-way valves actuator	3	40	120
Hot fluid temperature sensor	1	0.2	0.2
hot fluid flow meters	1	20	20
Circulating pumps	2	2	4

Table 3.8: Power consumption of auxliaries [15], [16], [17], [18], [19]

3.2.3 Block Flow Diagram

Concerning the Block Flow Diagram of the integrated system the mass and energy fluxes at the inlet and at the outlet of the control volume of the integrated system are the same described for the block flow diagram of the ORC, discussed in Section 3.1.7. It is to be underlined that the TES interact with the DHW, the SH and the ORC: in particular it exchanges heat with them by circuits in which a solution of water and propylene glycol flows. The company involved in the development of heat dissipation system is SYNAPSECOM, in the development of the domestic hot water and space heating systems is HYSYTECH and the TES are HSLU and COWA.

3.2.4 Process Flow Diagram

The Process Flow Diagram of the integrated system implements the working logic described in Section 3.2.1.

4 Description of the case study

All the fluids included in the preliminary list, represented in Table 3.1, have been firstly analysed by the pre-screening process. The assessment of the thermal efficiency has been carried out only for fluids resulting from the pre-screening. Once the working fluid has been selected, an analysis of different architectures regarding the thermal efficiency by using this fluid has been performed. In order to further analyse the performance of the ORC, its behaviour has been studied in all its operating conditions. The design of the pipes and components has been performed basing both on normal operating conditions, which takes into account the average outlet temperature in Athens, namely $18°C$, and the most probable temperature of the hot fluid coming from SYNAPSECOM, suggested by the SYNAPSECOM partner of the project, of $50°C$ and on limit conditions. While designing the integrated system, the interaction of all the components of the system has been studied in order to maximize the energy recovery from data center. While assessing the technical documentation, for each component normal operating conditions has been analysed together with limit conditions.

5 Results

5.1 Results of the ORC

In the following bulleted list the results from the pre-screening assessment are concerned:

- the results from the critical points analysis are show in the following figure:

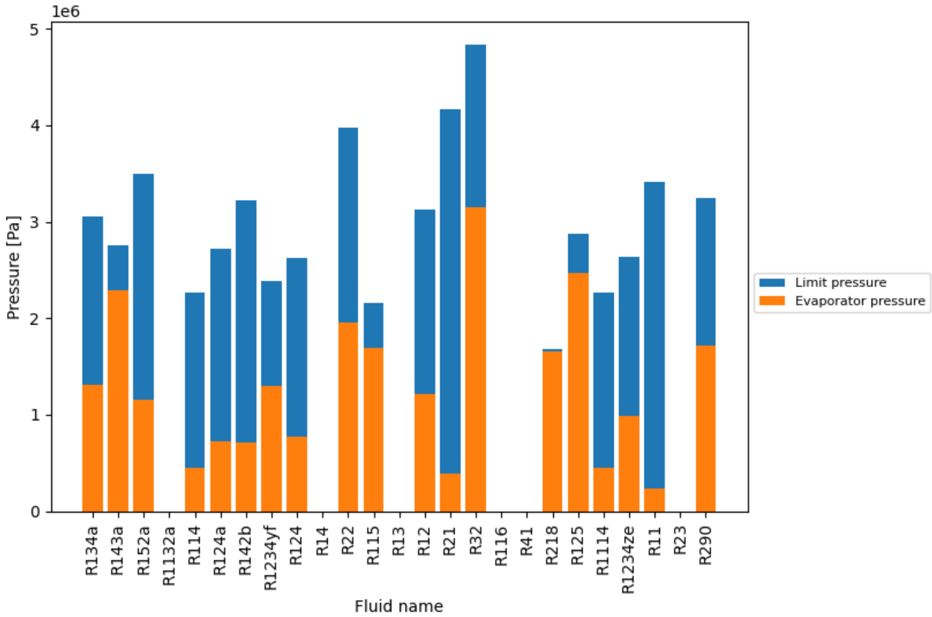


Figure 28: Limit vs evaporation pressure

- the values of ODP for different fluids and the status regarding the acceptability of the compounds are represented in Table A.1;
- the values of GWP for different fluids and the status regarding the acceptability of the compounds are represented in Table A.2;
- the ASHRAE safety classification for each fluid is shown in the Table A.3. Considering the refrigerants associated with the "−" symbol in Table A.3, by looking at their data sheets, R41 and R1114 have been considered not acceptable, while for R124a the safety data sheet is not available online. For this last case the refrigerant has been considered "not acceptable" since it makes no sense to perform an analysis on a fluid whose informations are really hard to be found and which than will probably be difficult to be found on the market. Basing on the last two considerations the status "acceptable" and "not acceptable" has been defined in Table A.3.

In the following summarizing figure the results of the pre-screening are reported. For each fluid the status regarding critical points ("acceptable" if satisfying both temperature and pressure constrains, "not acceptable" if not satisfying at least one one of the two), ODP, GWP and toxicity are reported and the result of the assessment for each fluid has been considered as:

- "acceptable" (green in Figure 29): if the status corresponding to each of the four category is "acceptable";
- "acceptable with warning" (orange in Figure 29): if the status corresponding to each category is never "not acceptable" and the status regarding at least one category is "acceptable with warning";

- "not acceptable" (red in Figure 29): if the status corresponding to at least one category is "not acceptable".

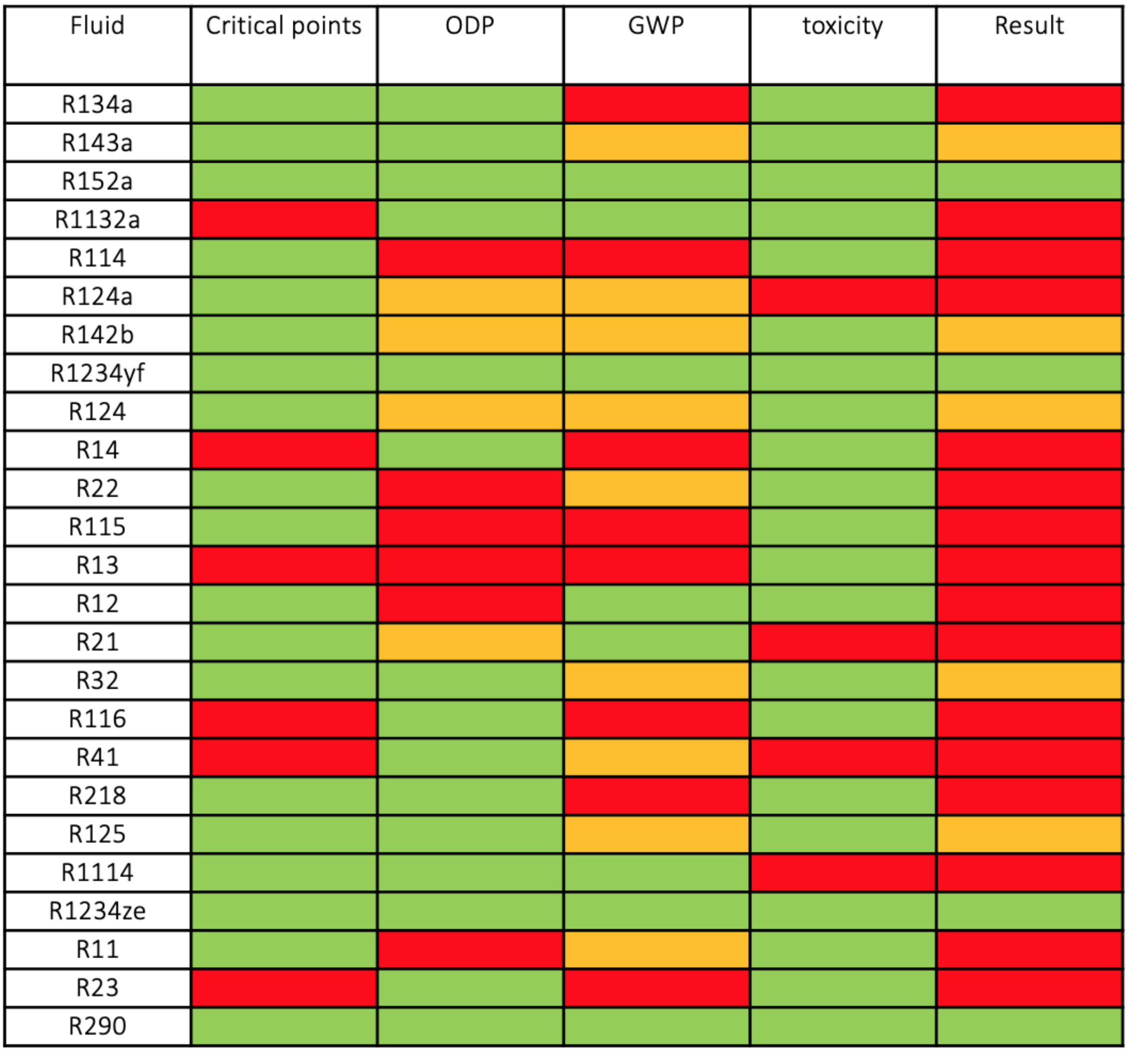


Figure 29: Summarizing figure on refrigerant status

The fluids to be analysed regarding thermal efficiency are given by the pre-screening and are:

- R143a;
- R152a;
- R142b;
- R1234yf;
- R124;
- R32;
- R125;
- R1234ze;
- R290.

In the following image the results of the sensitivity analysis of the thermal efficiency for the various fluids with respect to the pressure in the boiler are represented:

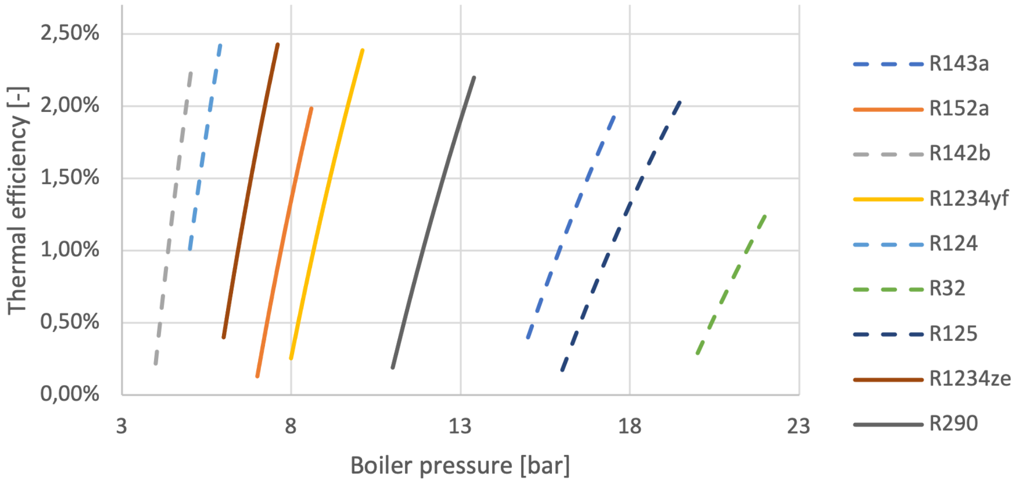


Figure 30: Sensitivity analysis of the thermal efficiency with respect to the boiler pressure

The dotted lines in the Figure 30 represents the fluids which are "acceptable with warnings", while the other lines represents the "acceptable" fluids.
In the following figure the thermal efficiency in the design conditions for each fluid is represented:

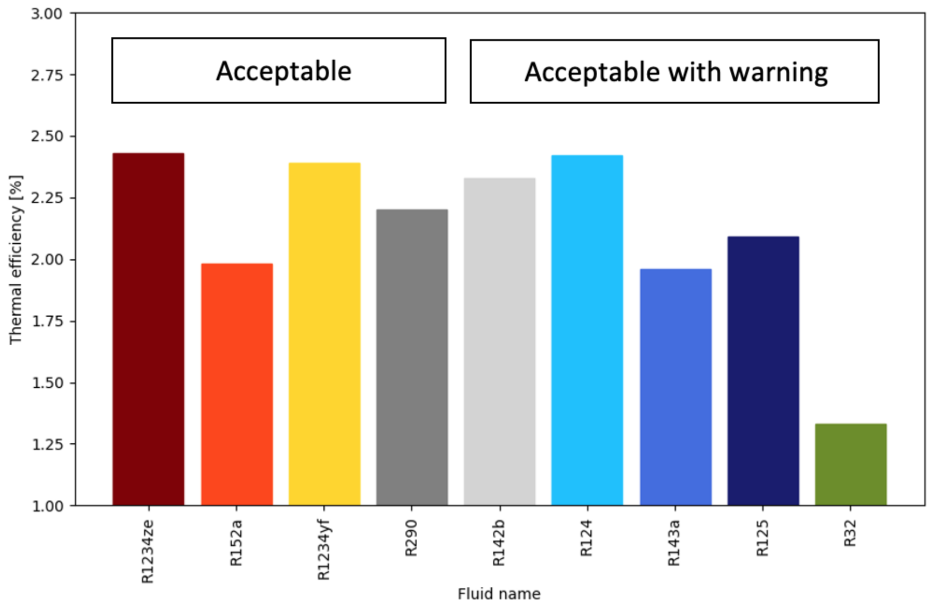


Figure 31: Thermal efficiency in the design point for each fluid

The fluids in Figure 31 are divided between "acceptable" and "acceptable with warning".

The thermal efficiency of the ORC with recuperator is compared with the one of the ORC basic architecture cycle in the following figure:

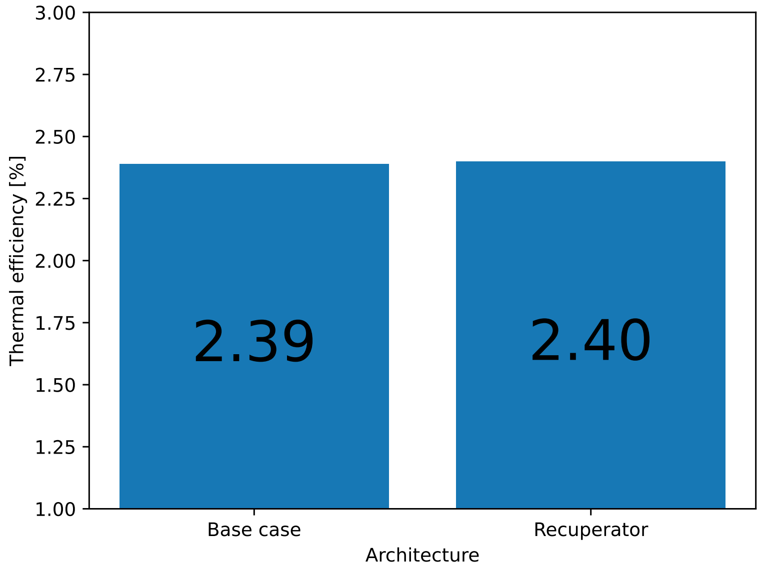


Figure 32: Thermal efficiency of basic architecture vs recuperator ORC

The behaviour of the thermal efficiency varying the duty of the boiler in the reheat ORC is represented below:

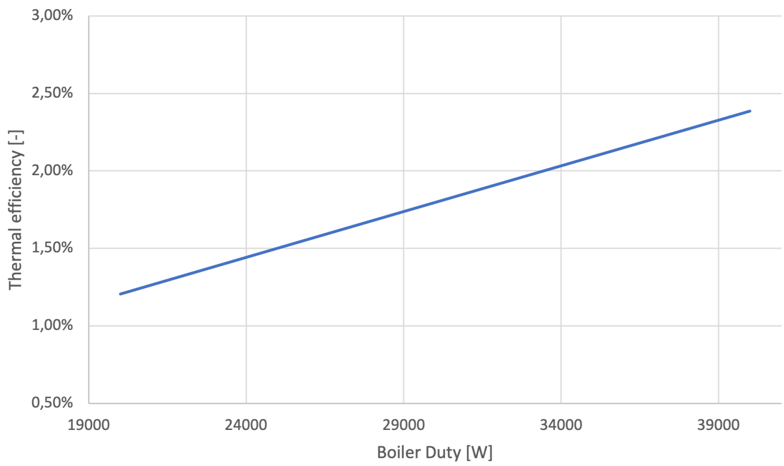


Figure 33: Thermal efficiency vs boiler's duty

The behaviour of the net power produced by the ORC in all its operating conditions is represented in the following contour map:

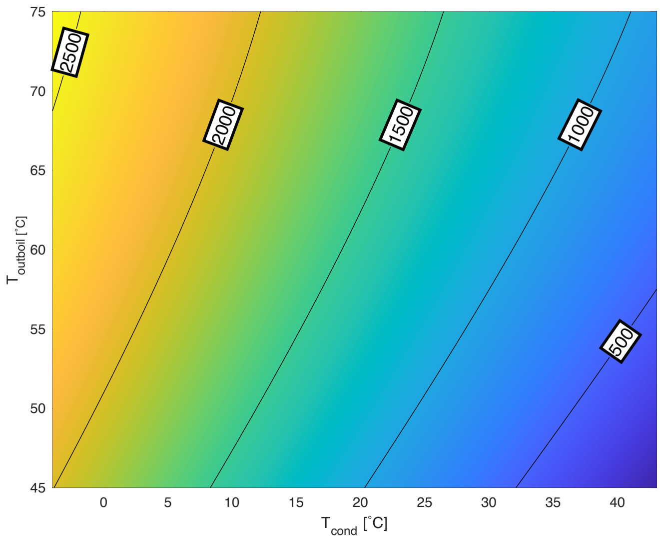


Figure 34: Behaviour of the net power [W] in all the operating conditions of the ORC

The behaviour of the thermal efficiency of the ORC in all its operating conditions is represented in the following contour map:

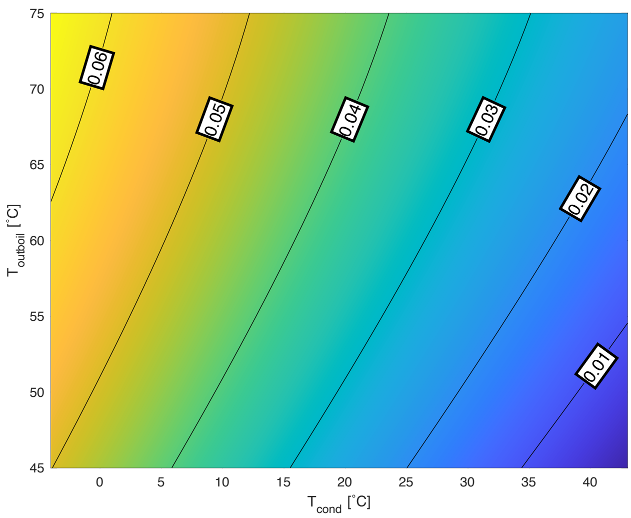


Figure 35: Behaviour of the thermal efficiency [-] in all the operating conditions of the ORC

The needed mass flow rate to abstract the 40 kW from the hot fluid in all the operating conditions is represented in the following contour map:

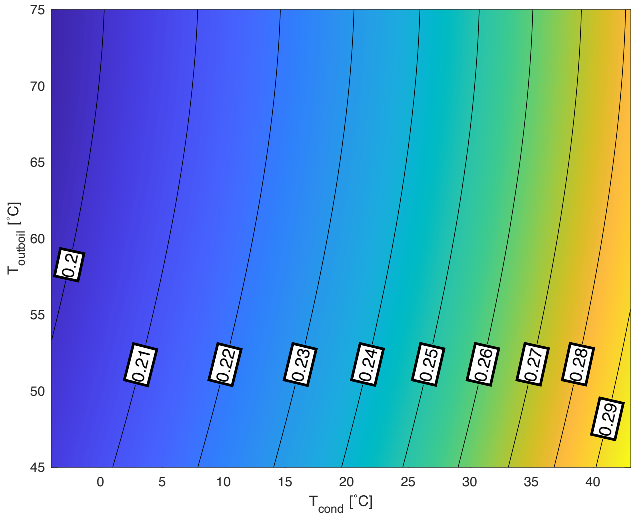


Figure 36: Behaviour of the mass flow rate [kg/s] in all the operating conditions of the ORC

The trend of the evaporator pressure varying the temperature at the outlet of the super-heater is represented in the following graph:

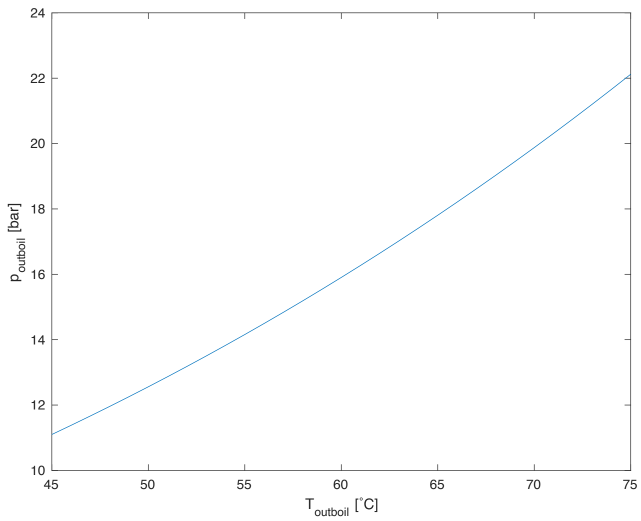


Figure 37: Evaporator pressure-outlet temperature of the super-heater

The trend of the condenser pressure varying the temperature at the outlet of the same components is represented in the following graph:

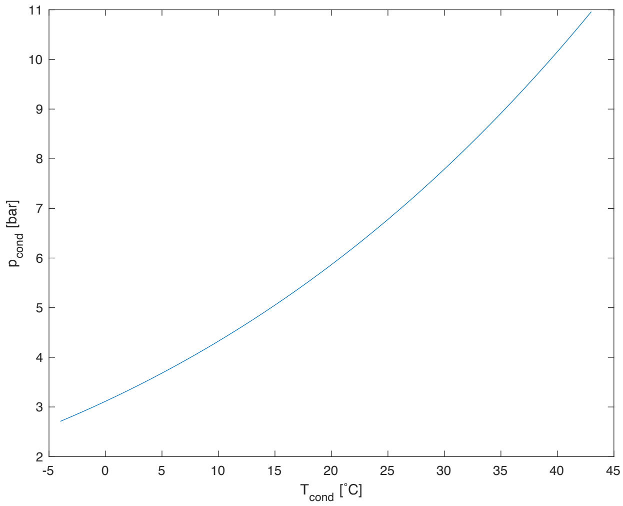


Figure 38: Condenser pressure-outlet temperature of the condenser

The results obtained for the piping are represented in the following table:

Pipe	5	6	7	10
D_{in} [mm]	18.0	18.0	23.4	23.4
Thickness [mm]	1.65	1.65	1.65	1.65
Δp_{lin} [kPa/m]	2.24	1.09	3.93	4.04
v [m/s]	1.09	1.09	11.19	11.34

Table 5.1: Results of the piping

In the following table the minimum thickness required and the thickness chosen for each pipe are shown:

Pipe	5	6	7	10
t_{min} [mm]	0.5	0.6	0.6	0.5
t_{chosen} [mm]	1.65	1.65	1.65	1.65

Table 5.2: Minimum and chosen thickness

In the following the battery limit is represented:

ID	SHORT DESCRIPTION	CODE/PFD CODE	IN/OUT	TYPE	AMOUNT	WORKING T
1	DATA TRANSMISSION	-	Input/Output	Service	**Minimum**: 6 Mbps Download / 1 Mbps Upload **Recomended**: 12 Mbps Download / 3,5 Mbps Upload	-
2	Electric supply	-	Input	Service	1,14 kW	-
3	Hot glycolated water	1	Input	Process	Min: 3000 L/h Max: 6500 L/h	Min: 50°C Max: 80°C
4	Hot glycolated water	2	Output	Process	Min: 3000 L/h Max: 6500 L/h	Min: 40°C Max: 75°C
5	Electricity to SYNAPSE	9	Output	Process	Max: 2,6 kW	-

ID	SHORT DESCRIPTION	DESIGN T	WORKING P	DESIGN P	Q.ty	Connection
1	DATA TRANSMISSION	-	-	-	1	TBD
2	Electric supply	-	-	-	1	TBD
3	Hot glycolated water	Min: -10 °C Max: 100 °C	Nor: 3 barg	5 barg	1	Flanged EN ISO 1092-1 DN40 PN6
4	Hot glycolated water	Min: -10 °C Max: 100 °C	Nor: 3 barg	5 barg	1	Flanged EN ISO 1092-1 DN40 PN6
5	Electricity to SYNAPSE	-	-	-	1	TBD

Figure 39: Battery limit of the ORC

In the following the fluid list is represented:

	SUBSTANCE DEFINITION				TEMPERATURE	
Streams	**Code**	**Definition**	**Fluid Type and Phase**	**Characteristics / Composition**	**Design [°C]**	**Working [°C]**
	TFP	2,3,3,3-Tetrafluoropropene	Flammable Gases liquid/gas	2,3,3,3-Tetrafluoropropene	-10 / 100	-4 / 80
	GW	Glicolated water	Flammable - Liquid	70 % Water+ 30% propylene glicol	-10 / 100	40 / 80

		PRESSURE		CONSTRUCTION	
Code	**Definition**	Design [barg]	Working [barg]	Piping class	Insulation
TFP	2,3,3,3-Tetrafluoropropene	24,0	2 / 23	TBD	P
GW	Glicolated water	5,0	3,0	TBD	P

Figure 40: Fluid list of the ORC

The BFD of the ORC is represented in the following:

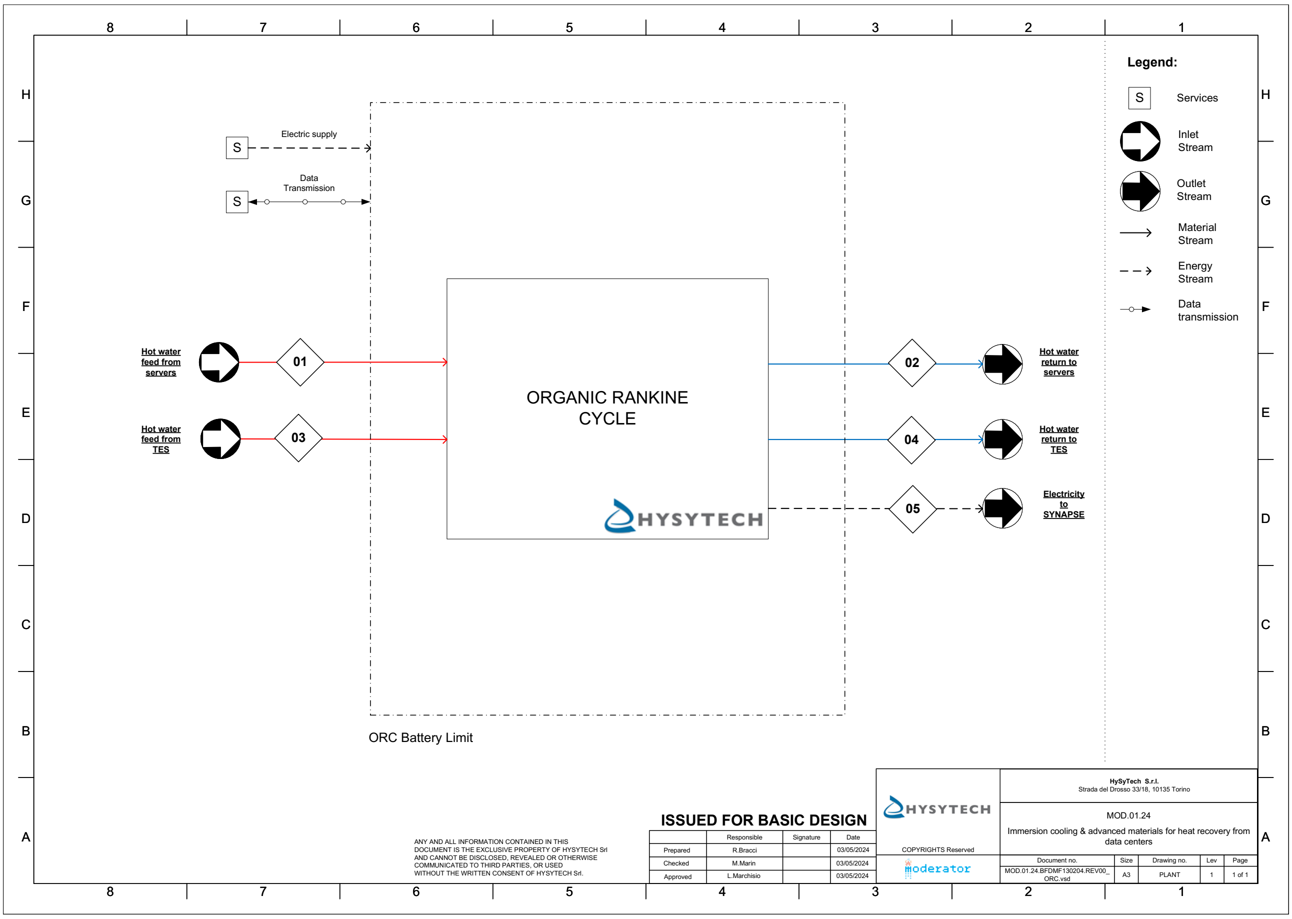


Figure 41: BFD of the ORC

The process flow diagram of the ORC is represented in the following:

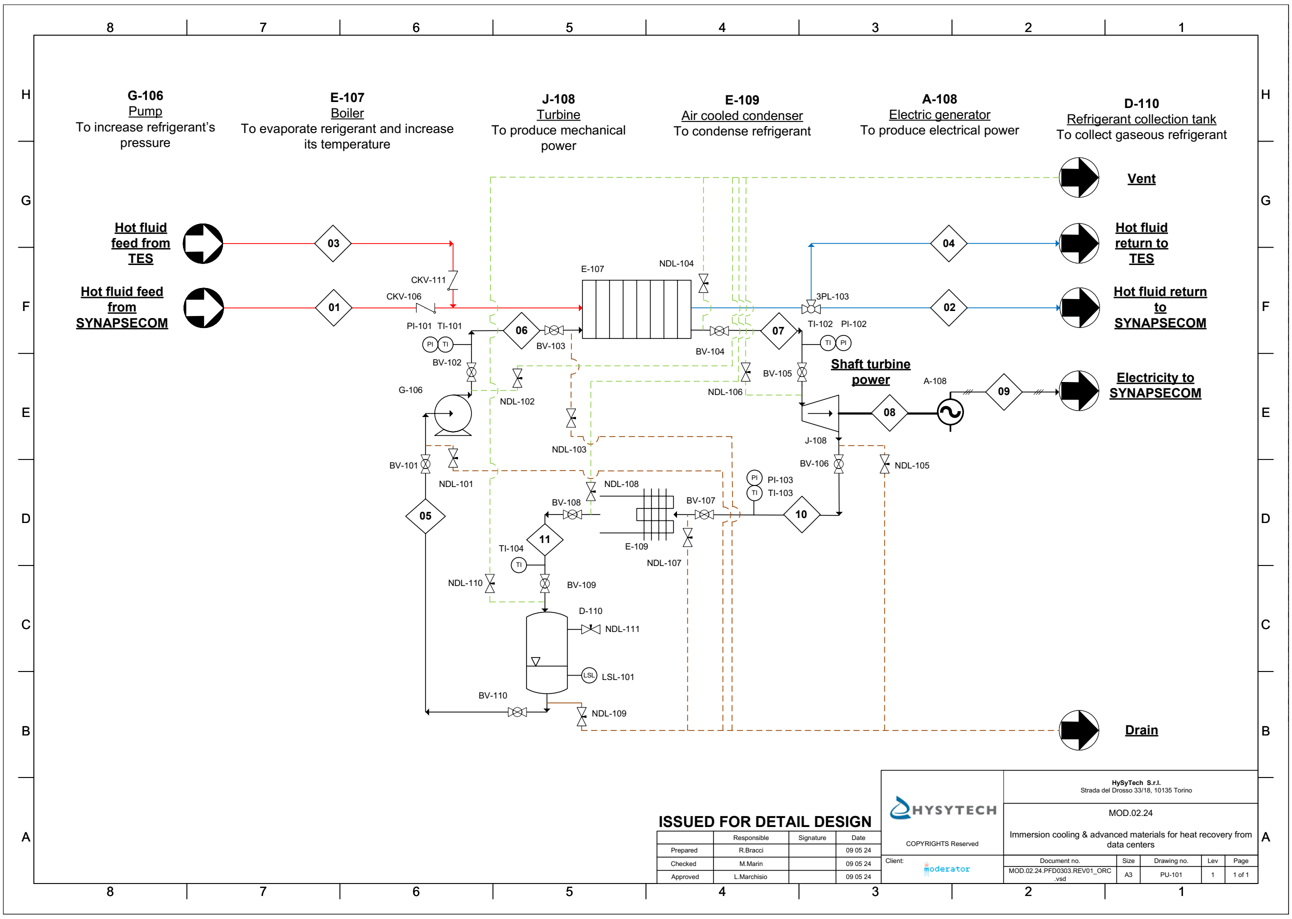


Figure 42: PFD of the ORC

The mass and energy balance is represented in the following:

		Stream	**1**	**2**	**5**	**6**	**7**	**10**
		Fluid name	Glicolated water	Glicolated water	TFP	TFP	TFP	TFP
		Type of fluid	Flammable Liquids	Flammable Liquids	Flammable Liquids	Flammable Liquids	Flammable Gases	Flammable Gases
		Phase	Liquid	Liquid	Liquid	Liquid	Gas	Gas
T	Nor	°C	50	42	18	19	45	21
P	Nor	barg	3,0	3,0	4,5	10,1	10,1	4,5
Density	Nor	kg/m3	1021,48	1030,13	1115,07	1112,95	62,14	29,82
MW	Nor	kg/kmol	35,4	35,4	114,0	114,0	114,0	114,0
Specific enthalphy	Nor	kJ/kg	-9774,16	-9801,50	-1131,43	-1130,69	-961,98	-974,73
Flow rate	Nor	kmol/h	148,66	148,66	7,48	7,48	7,48	7,48
	Nor	m3/h	5,16	5,11	0,77	0,77	13,74	28,62
	Nor	kg/s	1,46	1,46	0,24	0,24	0,24	0,24
Mass composition	Water	kg/kg	70%	70%				
	Propylene glycol	kg/kg	30%	30%				
	TFP	kg/kg			100%	100%	100%	100%
		tot	100%	100%	100%	100%	100%	100%
Molar composition	Water	mol/mol	70%	70%				
	Propylene glycol	mol/mol	30%	30%				
	TFP	mol/mol			100%	100%	100%	100%
		tot	100%	100%	100%	100%	100%	100%

Description	Pump
Name	**G-106**
Efficiency [-]	0,68
Discharge pressure [barg]	10,1
Mass flow rate [kg/s]	0,24
Electric power required [kW]	0,2
Head [m]	60,0

Description	Turbine
Name	**J-108**
Efficiency	0,4
Mass flow rate [kg/s]	0,24
Discharge pressure [barg]	4,5
New power output [kW]	1,3

Description	Boiler
Name	**E-107**
Outlet temperature [°C]	45
Heat duty [kW]	40,0
Mass flow rate [kg/s]	0,24
Discharge pressure [bara]	10,1

Description	Condenser
Name	**E-109**
Outlet tempearture [C°]	18
Outlet vapor fraction [-]	0
Discharge pressure [barg]	4,5
Mass flow rate [kg/s]	0,24
Heat exchanged [kW]	37,2

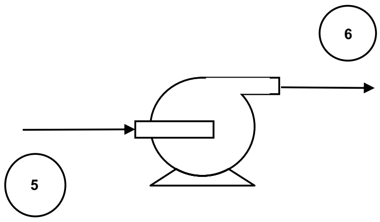
Figure 43: Mass and energy balance of the ORC

The process data sheets of the components of the ORC are represented in the following:

PERFORMANCE OF ONE UNIT:

Stream		5	6
Fluid Name		TFP	TFP
Fluid Type		Flammable Liquids	Flammable Liquids
Phase		Liquid	Liquid
Mass flow rate (NOR)	[kg/h]	853,6	853,6
Mass flow rate (MIN)	[kg/h]	1076,16	1076,16
Mass flow rate (MAX)	[kg/h]	830,8	830,8
Molecular Weight	[kg/kmol]	114,0	114,0

		INLET	OUTLET
Pressure (NOR)	[barg]	4,5	10,1
Pressure (MIN)	[barg]	9,9	10,1
Pressure (MAX)	[barg]	5,0	21,1
Temperature (NOR)	[°C]	18,0	18,6
Temperature (MIN)	[°C]	43,0	43,0
Temperature (MAX)	[°C]	21,0	22,8
Density (NOR)	[kg/m3]	1115	1113
Density (MIN)	[kg/m3]	1018,53	1018,43
Density (MAX)	[kg/m3]	1104	1098
Specific Gravity (NOR)	[-]	1,12	1,11
Viscosity (NOR)	[cP]	0,169	0,168
Specific Heat (NOR)	[kJ/kgK]	1,34	1,33
Thermal Cond. (NOR)	[W/(m°C)]	0,072	0,072



PRESSURE RATIO (NOR)	2,01	-
PRESSURE RATIO (MIN)	1,01	-
PRESSURE RATIO (MAX)	3,66	-
EFFICIENCY (NOR/MIN/MAX)	0,68	-
WORK (NOR)	0,174	kW
WORK (MIN)	0,006	kW
WORK (MAX)	0,468	kW
HEAD (NOR)	53	m
HEAD (MIN)	4	m
HEAD (MAX)	151	m

CERTIFICATION:

PED: ● ATEX: ● II 3G IIB T2

DESIGN:

Pump Type	Multistage centrifugal pump	
Design inlet pressure	[barg]	1,4
Design inlet temperature	[°C]	-20
Design outlet pressure	[barg]	23,9
Design outlet temperature	[°C]	60
Power supply type	220 V, 50 Hz	

MATERIALS:

Casing:	AISI 316
Impeller:	AISI 316
Flanges:	AISI 316
Gasket:	VITON

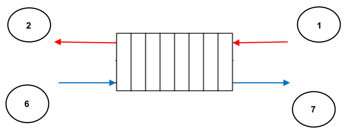
Safety valve: ☐ Relief pressure [barg] NA

NA: Not Apply

Figure 44: PDS of the pump

PERFORMANCE OF ONE UNIT:

Stream		1	2	6	7
Fluid Name		GW	GW	TFP	TFP
Fluid Type		Flammable liquids	Flammable liquids	Flammable liquids	Flammable gases
Phase		Liquid	Liquid	Liquid	Gas
Mass flow rate (NOR)	[kg/h]	7488,0	7488,0	853,6	853,6
Molecular Weight	[kg/kmol]	31,2	31,2	114,0	114,0
Pressure (NOR)	[barg]	4,0	4,0	10,1	10,1
Temperature (NOR)	[°C]	50	44	19	45
Density (NOR)	[kg/m3]	1048	1054	1113	62
Specific Gravity (NOR)	[-]	1,05	1,05	1,11	0,06
Viscosity (NOR)	[cP]	1,185E-03	1,355E-03	1,685E-04	1,415E-05
Specific Heat (NOR)	[kJ/kgK]	3189,50	3179,73	1334,39	1013,90
Thermal Cond. (NOR)	[W/(m°C)]	0,316	0,315	0,072	0,015



HEAT DUTY (NOR)	40,00	kW
Fooling resistance	0,0002	m^2K/W

CERTIFICATION:

PED: ATEX: II 3G IIB T2

DESIGN:

Heat exchanger type Plate heat exchanger

Design pressure for working fluid flow	[barg]	23,9
Design pressure for hot fluid flow	[barg]	5,0
Design temperature at working fluid at inlet	[°C]	-20
Design temperature at hot fluid at inlet	[°C]	100
Design temperature at working fluid at outlet	[°C]	90
Design temperature at hot fluid at outlet	[°C]	20

MATERIALS:

Plates:	AISI 316
Gasket:	VITON
Frame:	AISI 316

Safety valve: Relief pressure [barg] NA

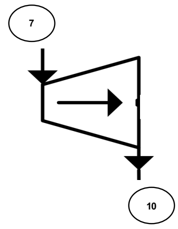
NA: Not Apply

Figure 45: PDS of the boiler

PERFORMANCE OF ONE UNIT:

Stream		7	10
Fluid Name		TFP	TFP
Fluid Type		Flammable Gases	Flammable Gases
Phase		Gas	Gas
Mass flow rate (NOR)	[kg/h]	853,56	853,56
Mass flow rate (MIN)	[kg/h]	1076,04	1076,04
Mass flow rate (MAX)	[kg/h]	701,6	701,6
Molecular Weight (NOR)	[kg/kmol]	114,0	114,0
Pressure (NOR)	[barg]	10,1	4,5
Pressure (MIN)	[barg]	10,1	9,9
Pressure (MAX)	[barg]	21,1	1,7
Temperature (NOR)	[°C]	45	21
Temperature (MIN)	[°C]	45	45
Temperature (MAX)	[°C]	75	-2
Density (NOR)	[kg/m3]	62	30
Density (MIN)	[kg/m3]	62	61
Density (MAX)	[kg/m3]	144	15
Specific Gravity (NOR)	[-]	0,06	0,03
Viscosity (NOR)	[cP]	1,415E-05	1,263E-05
Specific Heat (NOR)	[kJ/kgK]	1013,90	882,51
Thermal Cond. (NOR)	[W/(m°C)]	0,015	0,014

EFFICIENCY (NOR/MIN/MAX)	0,42	-
NET POWER OUTPUT (NOR)	1096	W
NET POWER OUTPUT (MIN)	22	W
NET POWER OUTPUT (MAX)	2579	W



CERTIFICATION:

PED: ■ ATEX: ■ II 3G IIB T2

DESIGN:

Type of turbine Scroll expander

Design inlet temperature	[°C]	90
Design inlet pressure	[barg]	24,0
Design outlet temperature	[°C]	-20
Design outlet pressure	[barg]	1,5

MATERIALS:

Spiral:	AISI 316
Body:	AISI 316
Gasket:	VITON
Shaft:	AISI 316

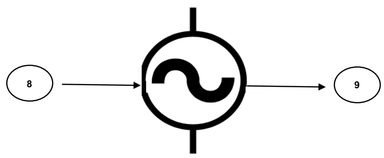
Safety valve: ☐ Relief pressure [barg] NA

NA: Not Apply

Figure 46: PDS of the turbine

PERFORMANCE OF ONE UNIT:

		8	9
Stream type		Mechanical power	Electrical power
Power (NOR)	kW	1270	1079,5
Power (MIN)	kW	28	23,8
Power (MAX)	kW	3047	2589,95



ELECTRICAL EFFICIENCY (NOR/MIN/MAX) 0,85 -

Safety valve: ☐ Relief pressure [barg] NA

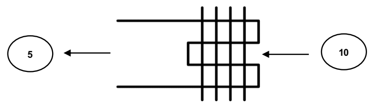
NA: Not Apply

Figure 47: PDS of the generator

PERFORMANCE OF ONE UNIT:

Stream		10	5
Fluid Name		TFP	TFP
Fluid Type		Flammable Gases	Flammable Liquids
Phase		Gas	Liquid
Mass flow rate (NOR)	[kg/h]	853,56	853,56
Molecular Weight (NOR)	[kg/kmol]	114,0	114,0
Pressure (NOR)	[barg]	4,5	4,5
Temperature (NOR)	[°C]	21	18
Density (NOR)	[kg/m3]	30	1115
Specific Gravity (NOR)	[-]	0,03	1,12
Viscosity (NOR)	[cP]	1,263E-05	1,695E-04
Specific Heat (NOR)	[kJ/kgK]	882,51	1341,53
Thermal Cond. (NOR)	[W/(m°C)]	0,014	0,072

HEAT DUTY (NOR)	37151	W
HEAT DUTY (MIN)	33213	W
HEAT DUTY (MAX)	39939	W



CERTIFICATION:

PED:

DESIGN:

Type of heat exchanger	Air cooled heat exchanger	
Design pressure	[barg]	15,0
Design inlet temperature	[°C]	60
Design outlet temperature	[°C]	-20

MATERIALS:

Tubes:	AISI 316
Fans:	AISI 316
Frame:	AISI 316

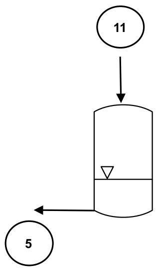
Safety valve: ☐ Relief pressure [barg] NA

NA: Not Apply

Figure 48: PDS of the condenser

PERFORMANCE OF ONE UNIT:

Stream		11	5
Fluid Name		TFP	TFP
Fluid Type		Flammable liquids	Flammable liquids
Phase		Liquid	Liquid
Liquid Fow (NOR)	[kg/h]	853,56	853,56
Pressure (NOR)	[barg]	4,51653	4,51653
Temperature (NOR)	[°C]	18	18
Density (NOR)	[kg/m3]	1115	1115
Viscosity (NOR)	[cP]	1,69E-04	1,69E-04
Molecular Weight	[kg/kmol]	114,0	114,0



Storage capacity	1,5	liter
Emptying time (NOR)	0,117574	min

DIMENSIONS:

Height	200,0	mm
Internal diameter	100,0	mm
Minimum thickness cylindrical body	1,6	mm
Minimum thickness toroidal end	1,5	mm

CERTIFICATION:

PED: ● SEP ATEX: ● II 3G IIB T2

DESIGN:

Maximum design temperature	[°C]	60
Maximum design internal pressure	[barg]	15,0
Minimum design temperature	[°C]	-20
Minimum design internal pressure	[barg]	1,5

Safety valve: ● Relief pressure [barg] 24,0

MATERIALS:

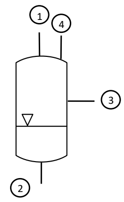
Body:	AISI 316
Toroidal ends:	AISI 316

Figure 49: PDS of the refrigerant collector tank

The MDS of the refrigerant collector tank is represented in the following:

PROJECT DATA

Maximum design temperature	[°C]	60
Maximum design internal pressure	[barg]	15,0
Minimum design temperature	[°C]	-20
Minimum design internal pressure	[barg]	1,5
Storage capacity	[L]	1,5



DIMENSIONS

Height	200,0	mm
Internal diameter	100,0	mm
Minimum thickness cylindrical body	1,6	mm
Minimum thickness toroidal end	1,5	mm

CERTIFICATION

PED: ● SEP ATEX: ● II 3G IIB T2

Safety valve: ● Relief pressure [barg] 24,0

CONNECTIONS

n°	ID	Service	Location	Type	DN	PN	Note
1	1	Working fluid inlet	top	Flanged	15	16	
2	2	Working fluid outlet	bottom	Flanged	15	16	
3	3	Working fluid charge	lateral	Threaded	15	16	
4	4	Bleed	top	Threaded	20	16	

Figure 50: MDS of the refrigerant collector tank

The equipment list of the ORC is represented in the following:

Tag ID	PFD N°	Qty.	Description	Service
G-106	1	1	Pump	Increase working fluid pressure
E-107	1	1	Boiler	Evaporate and heat up working fluid
J-108	1	1	Turbine	Produce mechanical power from working fluid expansion
E-109	1	1	Air cooled condenser	Condenses working fluid
A-108	1	1	Electric generator	Transforms mechanical power in electriity
D-110	1	1	Refrigerant collection tank	Collect liquid fluid from condenser

Tag ID	Charat. Size	Installed Power [kW]	Pressure [barg]	Heat.	Cool.	Elect.	Notes
G-106	-	0,5	4,5	Y	-	Y	
E-107	-	40	10,1	Y	-	-	
J-108	-	3	10,1	-	-	Y	
E-109	-	39,9	4,5	-	Y	-	Finned tube
A-108	-	2,6	-	-	-	Y	
D-110	V = 1,5 L	-	4,5	-	-	-	

Figure 51: Equipment list of the ORC

The instrument list of the ORC is represented in the following:

Pos.	Foglio PFD	Tag	P.U.	Q.ty	Description	Instrument type	Type I/O	Type signal	Temperature [°C]	Pressure [barg]
1	1	PI-101	106	1	Pressure indicator at pump outlet	Gauge	-	-	19	10,1
2	1	TI-101	107	1	Temperature indicator at boiler inlet	PT100	AI	4-20 mA	18	10,1
3	1	TI-102	107	1	Temperature indicator at boiler outlet	PT100	AI	4-20 mA	45	10,1
4	1	PI-102	108	1	Pressure indicator at turbine inlet	Gauge	-	-	45	10,1
5	1	PI-103	108	1	Pressure indicator at turbine outlet	Gauge	-	-	21	4,5
6	1	TI-103	109	1	Temperature indicator at condenser inlet	PT100	AI	4-20 mA	21	4,5
7	1	TI-104	109	1	Temperature indicator at condenser outlet	PT100	AI	4-20 mA	18	4,5
8	1	LI-101	110	1	Level indicator in refrigerant collector tank	Level Gauge	-	-	19	4,5

Pos.	Fluid	From	To	Unit	Connection	DN	PN	Immersion length [mm]	Insulation [mm]
1	2,3,3,3-Tetrafluoropropane	0	24,0	[barg]	Threaded	1/4" M NPT	24,0	-	-
2	2,3,3,3-Tetrafluoropropane	-20	60	[°C]	Threaded	1/4" M NPT	15,0	-	-
3	2,3,3,3-Tetrafluoropropane	20	90	[°C]	Threaded	1/4" M NPT	24,0	-	-
4	2,3,3,3-Tetrafluoropropane	0	24,0	[barg]	Threaded	1/4" M NPT	24,0	-	-
5	2,3,3,3-Tetrafluoropropane	0	15,0	[barg]	Threaded	1/4" M NPT	15,0	-	-
6	2,3,3,3-Tetrafluoropropane	-20	60	[°C]	Threaded	1/4" M NPT	15,0	-	-
7	2,3,3,3-Tetrafluoropropane	-20	60	[°C]	Threaded	1/4" M NPT	15,0	-	-
8	2,3,3,3-Tetrafluoropropane	0	300	[mm]	Threaded	1/4" M NPT	15,0	-	-

Figure 52: Instrument list of the ORC

The valve list of the ORC is represented in the following:

ID	Unit	Tag	Quantity	Type	Connection	DN	PN	Body Material
1	110	NDL-111	1	Needle	Compression	15	15,0	AISI 304
2	106 / 107	BV-102 / BV-103	2	Ball	Compression	15	24,0	AISI 304
3	107 / 108	BV-104 / BV-105	2	Ball	Compression	20	24,0	AISI 304
4	108 / 109	BV-106 / BV-107	2	Ball	Compression	20	15,0	AISI 304
5	109 / 110 / 110 / 106	BV-108 / BV-109 / BV-110 / BV-101	4	Ball	Compression	15	15,0	AISI 304
6	107 / 108	NDL-103 / NDL-104 / NDL-105 / NDL-106	4	Needle	Compression	20	24,0	AISI 304
7	109	NDL-107 / NDL-108	2	Needle	Compression	20	15,0	AISI 304
8	110	NDL-109 / NDL-110	2	Needle	Compression	15	15,0	AISI 304
9	106	NDL-101 / NDL-102	2	Needle	Compression	15	24,0	AISI 304

ID	Gasket	Wetted parts	Fluid	Temperature [°C]	Pressure [barg]	Actuator	Normal Position	Pilot
1	EPDM	AISI 304	2,3,3,3-Tetrafluoropropene	18	4,5	Manual	-	-
2	EPDM	AISI 304	2,3,3,3-Tetrafluoropropene	18,6	10,1	Manual	-	-
3	EPDM	AISI 304	2,3,3,3-Tetrafluoropropene	45	10,1	Manual	-	-
4	EPDM	AISI 304	2,3,3,3-Tetrafluoropropene	20,8	4,5	Manual	-	-
5	EPDM	AISI 304	2,3,3,3-Tetrafluoropropene	18	4,5	Manual	-	-
6	EPDM	AISI 304	2,3,3,3-Tetrafluoropropene	45	10,1	Manual	-	-
7	EPDM	AISI 304	2,3,3,3-Tetrafluoropropene	20,8	4,5	Manual	-	-
8	EPDM	AISI 304	2,3,3,3-Tetrafluoropropene	18	4,5	Manual	-	-
9	EPDM	AISI 304	2,3,3,3-Tetrafluoropropene	18,6	10,1	Manual	-	-

Figure 53: Valve list of the ORC

5.2 Results of the integrated system

The battery limits of the integrated system are represented in the following:

ID	SHORT DESCRIPTION	CODE/PFD CODE	IN/OUT	TYPE	AMOUNT	WORKING T
1	DATA TRANSMISSION	-	Input/Output	Service	**Minimum**: 6 Mbps Download / 1 Mbps Upload **Recomended**: 12 Mbps Download / 3,5 Mbps Upload	-
2	Electric supply	-	Input	Service	2 kW	-
3	Hot glycolated water	1	Input	Process	Min: 3000 L/h Max: 6500 L/h	Min: 50°C Max: 80°C
4	Hot glycolated water	2	Output	Process	Min: 3000 L/h Max: 6500 L/h	Min: 40°C Max: 75°C
5	Electricity to SYNAPSE	9	Output	Process	Max: 2,6 kW	-

ID	SHORT DESCRIPTION	DESIGN T	WORKIN G P	DESIGN P	Q.ty	Connection
1	DATA TRANSMISSION	-	-	-	1	TBD
2	Electric supply	-	-	-	1	TBD
3	Hot glycolated water	Min: -10 °C Max: 100 °C	Nor: 3 barg	5 barg	1	Flanged EN ISO 1092-1 DN40 PN6
4	Hot glycolated water	Min: -10 °C Max: 100 °C	Nor: 3 barg	5 barg	1	Flanged EN ISO 1092-1 DN40 PN6
5	Electricity to SYNAPSE	-	-	-	1	TBD

Figure 54: Battery limits of the integrated system

The block flow diagram of the integrated system is represented in the following:

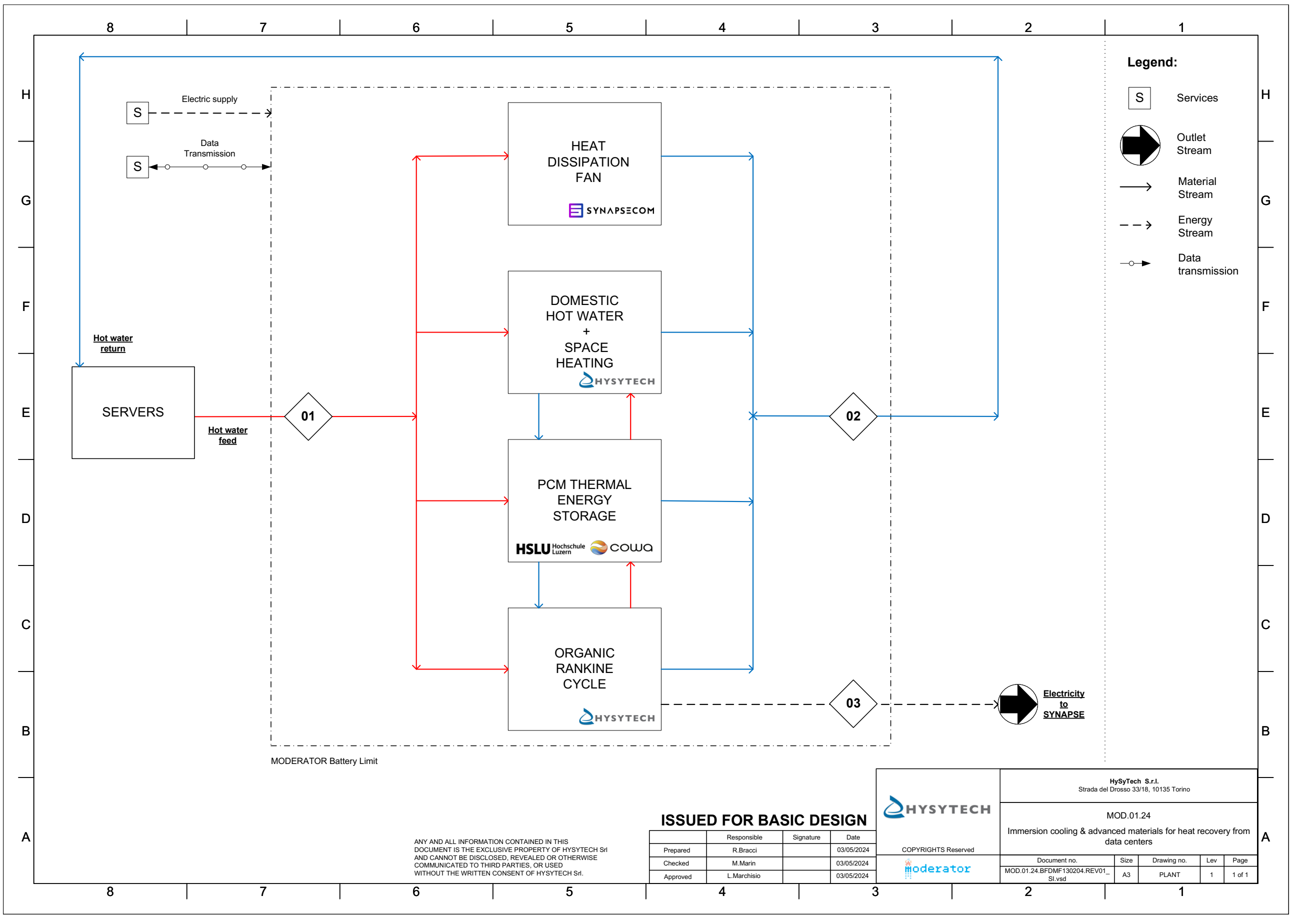


Figure 55: BFD of the integrated system

The process flow diagram of the integrated system is represented in the following:

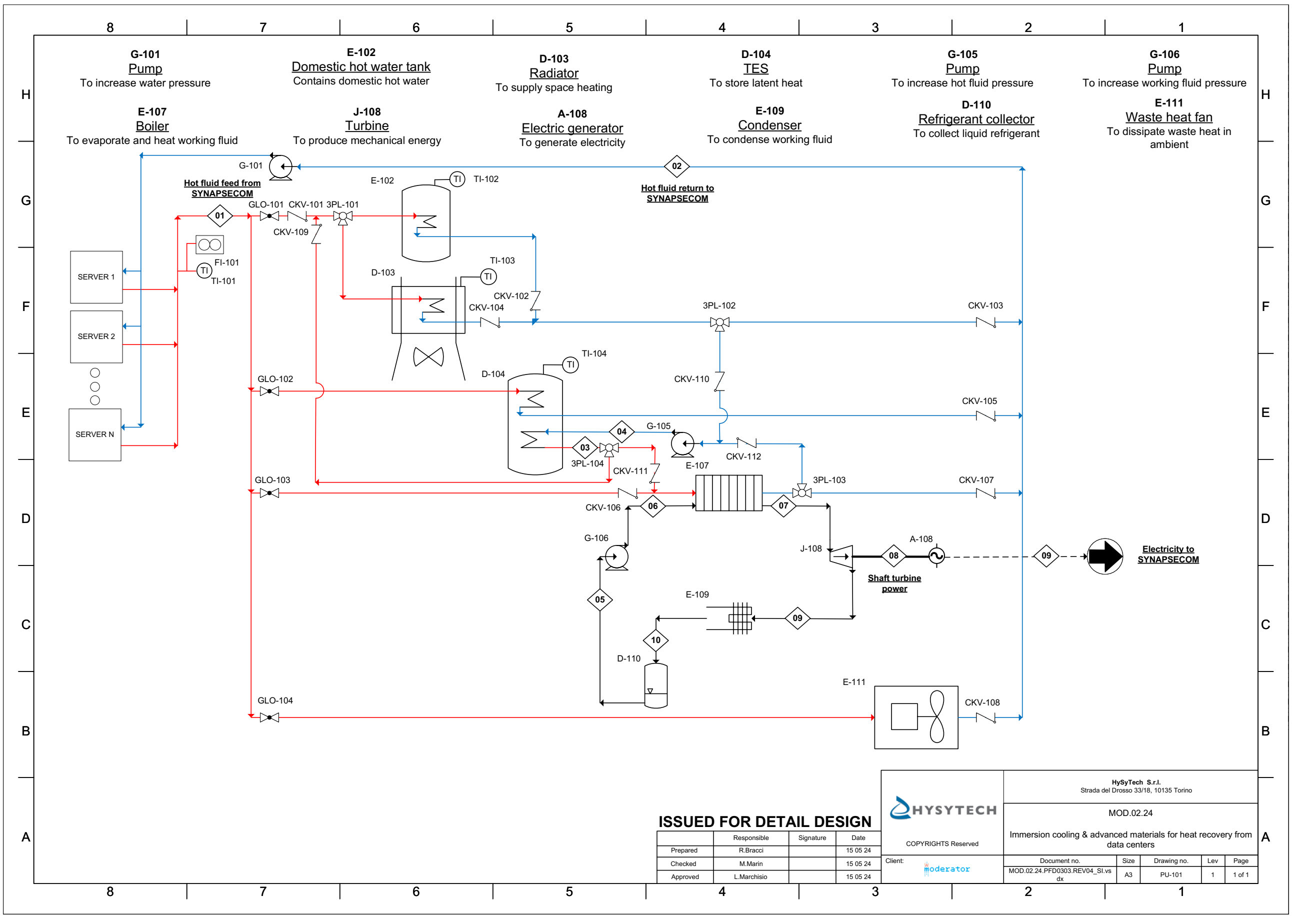


Figure 56: PFD of the integrated system

6 Discussion

The check of the pressure criteria is performed by analyzing Figure 28. As it can be seen from it the fluids R1132a,R14,R13,R116,R41,R23 pressure values are set to zero meaning they don't respect the temperature criteria. For the remaining refrigerants, which satisfy the temperature criteria, the pressure criteria is respected: in fact the orange bar, which represents the evaporation pressure at $40°C$, is always below the blue bar representing the limiting pressure for the given fluid.
As it can be seen from the Figure 30 for each fluid the thermal efficiency increases by increasing the pressure in the boiler, as it was expected, since the enthalpy drop increases and so also the output power from the turbine. In particular this increases more than the work required by the pump because of the divergence of the isobars and so the thermal efficiency increases too. The Figure 30 is also interesting because, considering how the minimum and maximum pressure level in the boiler have been defined in Subsection 3.1.1, it gives an idea of the pressure range in which the ORC works with each specific fluid.
Regarding Figure 31, the best three fluids in terms of thermal efficiency are, in order, R1234ze, R124 and R1234yf, having respectively a value for the characteristic parameter equal to 2.43%, 2.42% and 2.39%. R124 is "acceptable with warning", meaning it presents some criticalities, and since the value of thermal efficiency associated to this fluid is not so high to justify the higher environmental cost it implies, it has been excluded from the selection at this stage of the process. R1234ze and R1234yf, instead, are both "acceptable". In order to decide which of the two latter fluids was the best choice for the ORC a techno-economic evaluation has been carried out. The prices of R1234ze and R1234yf are $40€/kg$ and $26€/kg$ respectively [39]. Since the significant positive difference in price of R1234ze with respect to R1234yf is not compenseted by the small positive difference in thermal efficiency, R1234yf has been chosen as the working fluid.
As it can be seen from Figure 32 the thermal efficiency of ORC with recuperator is slightly higher than the one of the ORC with basic architecture. Anyway the increase in thermal efficiency is really small and it may be due to the parameters of the models utilized and so we can't say that the former architecture is better than the latter from the point of view of thermal efficiency. Furthermore the recuperator ORC implies additional expenses with respect to the basic architecture and, since this is not justified by a gain in thermal efficiency, it is not convenient from a techno-economic point of view.
As it can be seen from Figure 33, regarding the comparison of the thermal efficiencies in ORC with basic architecture and reheat ORC, the highest thermal efficiency is reached when the boiler delivers 40 kW, namely when the post-boiler doesn't provide any thermal power. From this observation we can state that the reheat ORC decreases the thermal efficiency with respect to the basic architecture ORC.
As it can be seen from Figure 34 the net power produced by the ORC increases by increasing the outlet temperature of the boiler and decreasing the temperature at the condenser, this because an higher temperature difference between the boiler and the condenser assures, in the model developed, an higher pressure difference between the pressure at the boiler and at the condenser, and so an higher enthalpy step at the turbine. As it can be seen from Figure 35, the thermal efficiency has the same trend of the net power output, this because the increase of the power output from the turbine consequent to the higher enthalpy step is more influential on the performance parameter than the increased work required by the pump to assure an higher pressure step. It is to be noted that the thermal efficiency has resulted having a maximum value of 6.4%, which is a reasoning value compared to the ones that can be found in literature, and a net power output associated to it of 2.5 kW. The normal operating conditions has been defined as the ones corresponding to the average temperature in Athens in the last ten years and the minimum temperature of the hot fluid coming from the data center, and the thermal efficiency and net power output obtained with this conditions correspond respectively to 2.74% and 1.1 kW, which are still reasoning values compared to the ones found in literature.

The comparison between the thermal efficiencies in the conditions just mentioned with the thermal efficiencies obtained in the case studies analyzed in Section 2.5 is represented in the following graph:

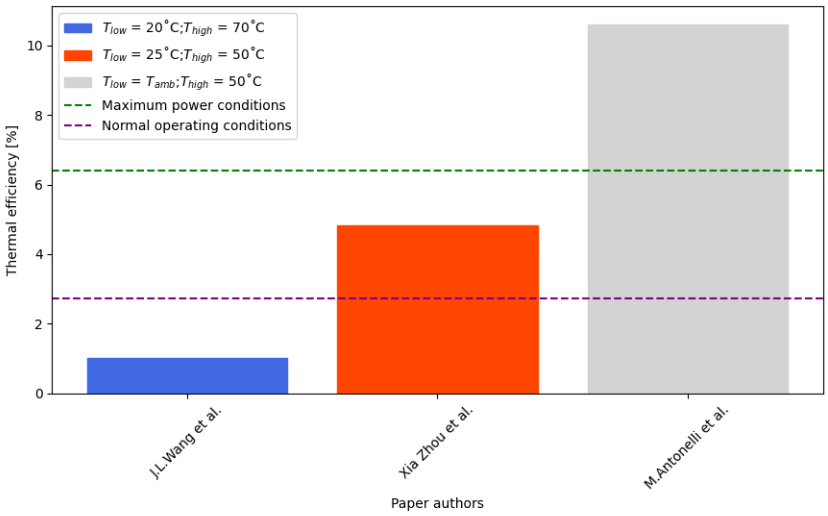


Figure 57: Thermal efficiency of ORC in different operating conditions vs thermal efficiencies obtained in case studies

As it can be seen from Figure 36, the behaviour of the mass flow rate is the opposite with respect to that of the net power output and the thermal efficiency. This is caused by the fact that, for bigger temperature differences between the outlet and the inlet of the boiler, considering almost constant the specific heat capacity of the working fluid in the range of temperature analysed, the mass flow rate required to abstract 40 kW of heat from the hot fluid coming from SYNAPSECOM will be lower. This can be noted by analysing the Equation 3. Regarding Figure 37, it is to be noted that the pressure at the boiler increases with the temperature at the outlet of the boiler, according to the modelling of this component that has been performed. The same reasoning can be applied to Figure 38.
Regarding Table 5.1, concerning the linear pressure drops, both in the liquid and in the gaseous lines, these are one order of magnitude higher than the value initially considered as acceptable. Anyway, in order to reduce this value, it would be needed to increment the diameter of the pipe, and this would cause a decrease in the velocity. The diameter that has been chosen is thus a compromise that allows to both the parameters in analysis (velocity and pressure drop) to approach their acceptable values as much as possible. Regarding the velocities, these are in the acceptable range in the case of the pipes containing liquid, while they are slightly lower in the case of gaseous lines, for the same reason described in the analysis of the pressure drop in the pipes. As it can be seen from Table 5.2, all the chosen thicknesses are higher than the minimum thickness and so they are suitable for this application. The part regarding the preparation of the technical documentation of the ORC will not be discussed in this section, since it is based on the analysis performed and doesn't provide any additional result to be discussed.
The same reasoning just described can be applied also for the definition of the preliminary working logic and the preparation of the technical documentation of the integrated system.

7 Conclusions

7.1 Conclusions for the ORC

The aim of this analysis was to design an ORC able to produce electricity from the waste heat of a data center and to produce technical documentation needed in order to purchase the various components of the system and build it. In the first part a pre-screening of various fluids has been performed in order to identify some candidates as the working fluid of the system. Once a short list has been created, an analysis regarding the thermal efficiency of the ORC working with the various candidates has been performed and the working fluid of the system has been chosen basing on the maximization of the parameter just said. Once the working fluid has been selected an analysis regarding the architecture has been performed. In particular the ORC with recuperator and the reheat ORC architectures have been analysed comparing them with the basic one. Later, after having identified the operating conditions of the system, the performance of the system in all the working range has been assessed. Then, knowing the behaviour of the ORC in all the operating conditions, the design of the pipes and of the components has been performed. Finally, basing on the data obtained in the analysis, some technical documentation useful in order to purchase the components and to build the system has been produced. In particular the technical documentation includes: the battery limit, the fluid list, the block flow diagram, the process flow diagram, the mechanical data sheet, the equipment list, the instrument list and the valve list.
The pre-screeing is based on four main criteria which are: threshold regarding critical points, ozone depleting potential, global warming potential and toxicity. The fluids resulting from the pre-screeing has been divided in acceptable (satifying all the requirements) and acceptable with warnings (meaning that they don't satisfy all the requirements, but regarding the thresholds that are not satisfied, they are not far from the acceptable values). The fluids resulting from this preliminary analysis are: R143a, R152a, R142b, R1234yf, R124, R32, R125, R1234ze, R290. Then the thermal efficiency of the ORC operating with each of these fluids has been studied and a ranking of this critical parameter as a function of the operating fluid has been obtained. Basing both on this ranking, on the acceptability derived from the pre-screening and on the cost of each refrigerant fluid R1234yf has been chosen as a working fluid. In particular this fluid has resulted in the third highest value of thermal efficiency, behind, in order, R1234ze and R124. Anyway, R124 didn't respect all the acceptability criteria since it was classified as "acceptable with warning" and R1234ze, despite being "acceptable" as R1234yf, resulted in a small thermal efficiency increase with respect to the big difference in price. Then the two previously said architectures have been analysed. The recuperator ORC has resulted in the same thermal efficiency of the base case. In the former case it is needed to buy, additionally with respect to the base case, the pre-heater and the recuperator and since no gain in thermal efficiency is achieved, this is not convenient from a techno-economic point of view. Then the reheat ORC architecture has been assessed, but also in this case, it has resulted that it implied no increase in the thermal efficiency with respect to the basic architecture. Basing on these reasonings the basic architecture has been selected. Then the operating conditions of the ORC have been defined, basing on the external temperature in Athens (where the ORC system will work) during the year and on the expected temperature of the hot fluid coming from the data center. In particular the external temperature in Athens has been found ranging between -4°C and 43 °C and the temperature of the hot fluid coming from the data center between 50°C and 80°C. In order to perform the analysis of the ORC in all its operating conditions the temperature at the condenser has been considered equal to the external temperature and the temperature at the outlet of the boiler has been considered equal to the temperature of the hot fluid minus 5°C. The thermal efficiency has resulted having a maximum value of 6.4%, which is a reasoning value compared to the ones that can be found in literature, and a net power output of 2.5 kW. The normal operating conditions has been defined as the ones corresponding to the average temperature in Athens in the last ten years and the minimum temperature of the hot fluid coming from the data center, and the thermal efficiency and net power output obtained with this conditions correspond respectively to 2.74% and 1.1 kW, which are still reasoning values compared to the ones found in literature. Then the pipes have been designed basing on the optimization of the pressure drops and of the velocity. The components have been designed basing on both the most critical and the normal operating conditions. The technical documentation which has been produced is based on the analysis that has been performed and will be used by the company where this thesis has been developed to purchase the equipment and build the system. In particular the aim of the various technical documents produced is to:

- Battery Limits: establish clear boundaries and responsibilities inside the process plant;

- Fluids List: compile and list all the fluids used within the system;
- Block Flow Diagram: depict the major steps or blocks involved and the flow of materials or energy between them;
- Process Flow Diagram: show the sequence of steps or operations involved;
- Mass and Energy Balance: track the flow of materials and energy in the system to ensure that inputs, outputs, and transformations are properly accounted for;
- Process Data Sheet: give specific and detailed pieces of information regarding the components of a system used in the process;
- Mechanical Data Sheet: assure that all the mechanical and performance details of a component are carefully designed, documented and understood by all the parties involved in the project;
- Equipment List: ensure that all the necessary tools are available and in a proper working condition, aiding in smooth operation and planning;
- Instrument List: understand the instrumentation involved and facilitate troubleshooting, maintenance and upgrades;
- Valve List: help in designing, operating and maintaining process system efficiency.

7.2 Conclusions for the integrated system

The aim of this analysis is to define a preliminary working logic of the integrated system and to prepare the basic technical documentation, comprehending the Battery Limits, the Block Flow Diagram and the Process Flow Diagram. Firstly the working logic has been defined and then, basing on it, the Battery Limits, Block Flow Diagram and Process Flow Diagram has been prepared.
The main idea behind the working logic of the integrated system is to optimize the recovery of energy coming from the data center by exploiting the storage to supply domestic hot water, space heating and ORC in a more uniform way with respect to how this exploitation could be performed with only the flow in the hot fluid circuit. In particular the surplus energy is used to fill the storage and in the case of deficit of energy, this is provided by the storage when available. The working logic is based on a hierarchical order, since the priority list of the systems to be supplied by the hot water coming from the data center are the Domestic Hot Water, the Space Heating, the Organic Rankine Cycle and the Thermal Energy Storage. Only if none of these systems needs (in the case of domestic hot water and space heating) or can (in the case of the organic Rankine cycle and the thermal energy storage) accept the heat power from the hot fluid, this will be dissipated by the Heat Dissipation Fan. The working logic has then been used to determine the layout of the system that is represented in both the block flow diagram (Figure 55) and the process flow diagram (Figure 57) of the integrated system and to compile the Battery Limits. The aim of these technical documents is the same as described in Section 7.1.

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Appendices

A Pre-screening of the working fluid

```
    Tc[ii], Vc[ii], pc[ii] = eos.critical(x)
     if Tc[ii]>=T_lim_cr:
         p_ev[ii],y_bub_ev[ii] = eos.bubble_pressure(T_ev, x)
         p_lim[ii] = pc[ii]-10e5
     elif Tc[ii]<T_lim_cr:
         print('Il fluido',fluid_name[ii],'non rispetta il limite sulla temperatura critica')
         p_lim[ii] = 0
         p_ev[ii] = 0
         y_bub_ev[ii] = 0
         pc[ii] = 0
```

Listing 1: Python code implementing critical points' criteria

Fluid name	$ODP[-]$	Status
R134a	0	acceptable
R143a	0	acceptable
R152a	0	acceptable
R1132a	0	acceptable
R114	0.58	not acceptable
R124a	0.022	acceptable with warning
R142b	0.07	acceptable with warning
R1234yf	0	acceptable
R124	0.022	acceptable with warning
R14	0	acceptable
R22	12.0	not acceptable
R115	0.44	not acceptable
R13	1	not acceptable
R12	1	not acceptable
R21	0.04	acceptable with warning
R32	0	acceptable
R116	0	acceptable
R41	0	acceptable
R218	0	acceptable
R125	0	acceptable
R1114	0	acceptable
R1234ze	0	acceptable
R11	1	not acceptable
R23	0	acceptable
R290	0	acceptable

Table A.1: ODP and status of refrigerants

Fluid name	$GWP[-]$	Status
R134a	1300	acceptable with warning
R143a	4800	acceptable with warning
R152a	16	acceptable
R1132a	1	acceptable
R114	8590	not acceptable
R124a	609	acceptable with warning
R142b	2310	acceptable with warning
R1234yf	4	acceptable
R124	527	acceptable with warning
R14	7390	not acceptable
R22	1760	acceptable with warning
R115	7670	not acceptable
R13	13900	not acceptable
R12	1	acceptable
R21	148	acceptable
R32	677	acceptable with warning
R116	12200	not acceptable
R41	116	acceptable with warning
R218	8830	not acceptable
R125	3170	acceptable with warning
R1114	9	acceptable
R1234ze	6	acceptable
R11	4660	acceptable with warning
R23	12400	not acceptable
R290	0.02	acceptable

Table A.2: GWP and status of refrigerants

Fluid name	ASHRAE safety classification	Status
R134a	A1	acceptable
R143a	A2L	acceptable
R152a	A2	acceptable
R1132a	A2	acceptable
R114	A1	acceptable
R124a	-	not acceptable
R142b	A2	acceptable
R1234yf	A2L	acceptable
R124	A1	acceptable
R14	A1	acceptable
R22	A1	acceptable
R115	A1	acceptable
R13	A1	acceptable
R12	A1	acceptable
R21	B1	not acceptable
R32	A2L	acceptable
R116	A1	acceptable
R41	-	not acceptable
R218	A1	acceptable
R125	A1	acceptable
R1114	-	not acceptable
R1234ze	A2L	acceptable
R11	A1	acceptable
R23	A1	acceptable
R290	A3	acceptable

Table A.3: ASHRAE safety classification of refrigerants