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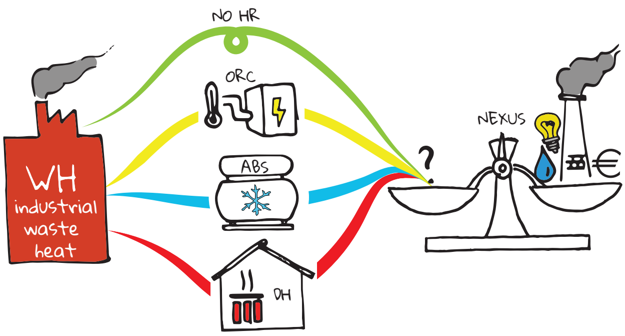
A Water-Energy-CO_2 Nexus Perspective on Industrial Heat Recovery Projects



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"Den som ger sig in i leken, får leken tåla"

Swedish proverb

Abstract

The water-energy nexus concept, i.e. the interdependence between energy conversion and water consumption, has only emerged as a research and policy issue in the last few years. However, although manufacturing is one of the sectors where the greatest increases in water consumption are expected in future, and is a major energy consumer, the literature on the water-energy nexus contains few studies focusing on industries other than electricity generation.

On the other hand, industrial heat recovery is an enabler for carbon emissions reduction and energy efficiency, and many studies demonstrate how using recovered heat for district heating is beneficial in these respects.

Other studies also demonstrate that waste heat recovery for internal direct use or for power generation, mainly through bottoming Organic Rankine Cycles, improves energy efficiency and carbon emission performances. Absorption cooling, in spite of being a well-known and mature technology, is mentioned or investigated in only very few studies, which mainly analyze economic feasibility and energy savings but do not explicitly assess carbon emission performance. Literature reviews show that the impact of all these options on water consumption has not been investigated to a great extent. To the best of the author's knowledge, only a couple of papers, appearing in the last two years, actually calculate both energy efficiency and water consumption indicators for a heat recovery project. One would expect synergistic effects to arise; however, interestingly enough, trade-off and pitfalls, leading to increasing water demand, have been reported for one case of heat recovery from the steelmaking industry in China.

This thesis contributes to fill the research gaps discussed above, i.e. the lack of studies on the water-energy nexus in industrial contexts, particularly concerning waste heat recovery projects, and the limited range of studies concerning absorption cooling as a waste heat recovery option, with special reference to low grade waste heat flows. For this purpose, two case studies of low grade heat recovery industrial settings are examined and compared from a Water-Energy Nexus perspective, within which CO_2 equivalent emissions, energy consumption, and economic feasibility indicators are calculated taking a life-cycle oriented approach.

Examined options for industrial waste heat recovery include power generation through an ORC cycle, heat upgrade via absorption cooling to

meet an internal cooling demand, and distribution through district energy networks to meet heating and air conditioning requirement of a remote office building. The waste heat sources are at a *low temperature* (<200°C) and the cases deal with flue gases cooling system at a steel-making plant (case studies *1a* and *1b*) and the cooling system of a biogas fuelled waste-to-energy (case study 2).

Case study *1* compares electricity generation from low grade waste heat via the Organic Rankine Cycle with internal uses of waste heat, namely the use of absorption cooling for cooling cabinets that house electric transformer, generator, and switch cabinets, which need air conditioning to preserve their functionality. Case study *1a* presents a general techno-economic analysis of alternative configurations. A preliminary energy balance model and a Monte Carlo model have been developed to undertake a generalization of the case study to the EU-15, based on a Europe-wide extensive review of energy and water prices, of energy sources and corresponding resource efficiency indicators. The influence of the national electricity mix and the climate zone in the various configurations is thus evaluated. It is shown how systems performance depends significantly on the residual heat dissipation systems used both originally and after the heat recovery intervention to dissipate residual waste heat, which can be based on wet-cooling or dry-cooling. Decisions by firms in favour of one technology option or another are assumed to be made on a purely economic basis.

The author proposes an evaluation of both direct and indirect blue water consumption and carbon emissions, with the aim of determining whether the net overall water balance, also considering the indirect consumption associated with electricity demand at various level, is always favorable.

Under the examined technical and economic conditions, absorption cooling is found to be an attractive option in all cases, and to be preferable to electricity generation via Organic Rankine Cycles in all countries and considering all nexus-related performance indicators. It is also found that local conditions and carbon reduction incentives may generate trade-offs and pitfalls. While all indicators are favorable when dry cooling systems are used, net water consumption may increase as much as tenfold in some countries when a technology switch occurs, because wet cooling systems become economically preferable. This largely depends on local economic conditions and electricity generation mix, and typically happens where water prices are low and power prices are high.

Having demonstrated the overall economic feasibility of absorption cooling as a heat recovery technology, the problem is of practical interest for electric steelmaking sites, and possibly also for similar applications, which are located all over the world.

It is of both theoretical and practical interest to compare such energy recovery based solutions with energy saving alternatives based on so called free cooling. In order to evaluate the economic and water-energy performance of such configurations worldwide, a TRNSYS simulation model is developed and mechanical compression cooling, free cooling with mechanical compression, and absorption cooling alternatives, are simulated for the 16 out of 17 worldwide climate zones defined by ASHRAE 90.1.

Looking at the energy-environmental performance from a water-energy nexus perspective, heat recovery based solutions are found to outperform even free cooling based alternatives in virtually all climate zones when dry coolers are used as dissipators. Water consumption, however, can increase in absorption cooling configurations in cold climates when cooling towers are used. From an economic viewpoint, however, free cooling is the best option exactly in those climate zones.

In case study 2, heat recovery coupled with district heating is taken into account. This configuration is also used to understand the difference between internal use and external use of recovered waste heat.

It was found that climate does not substantially affect heat recovery configurations for cooling purposes, while the national energy mix (indirect emissions/consumption) and market prices could play a valuable role. If decision makers aim to minimize systems costs as well as carbon footprint and primary energy consumption, then absorption chilling for internal cooling demand is the best option for those systems. If it is also desired that overall blue water footprints are minimized, or at least their deterioration avoided, then decision makers should also consider the footprint of the energy flows to be replaced and compare them with the footprints of the recovery opportunities and how much waste heat should be dissipated before and after energy recovery, and how dissipation is performed, i.e. by consuming water or not.

In general terms, our results for water footprints lead us to agree with the literature recommendation that the carbon footprint should not be taken as a proxy for environmental impact.

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Outline

Chapter 1	Gives an introduction to the water-energy nexus, the scientific background, the case studies and the environmental indicators used. The aim of the thesis as well as the research questions are also presented here.
Chapter 2	Describes the technical background required for a better understanding of the components and the design problems related to the various configurations analyzed.
Chapter 3	Describes the methodology, the systems studied, the environmental indicators, and the tools used for the evaluation of the nexus indicators. The model building is also described here as an introduction to the next chapter.
Chapter 4	Describes the case studies developed and the model building.
Chapter 5	Provides the results obtained for each of the case studies developed, and a comparative analysis is also given.
Chapter 6	Includes a brief discussion of the results. The answers to the specific research questions and the main limitations of the research are also presented.

Contents

List of Figures

List of Tables

List of Abbreviations

ABS(c)	**Abs**orption chiller
AUX	**A**uxiliaries
BF	**B**last **F**urnace
BOF	**B**asic **O**xygen **F**urnace
CHP	**C**ombined **H**eat (and) **P**ower
COP	**C**oefficient **O**f **P**erformance
CT	**C**ooling **T**ower
DC	**D**ry **C**oolers,
DH	**D**istrict **H**eating
DH+C	**D**istrict **H**eating and **C**ooling
EAF	**E**lectric **A**rc **F**urnace
EER	**E**nergy **E**fficiency **R**atio
ETS	**E**mission **T**rading **S**chemes
EU	**E**uropean **U**nion
FC	**F**ree **C**ooling
GHG	**G**reen**h**ouse **G**ases
GUI	**G**raphical **U**ser **I**nterface
GWP	**G**lobal **W**arming **P**otential
HO	**H**eating **O**il
HR	**H**eat **R**ecovery
HVAC	**H**eating, **V**entilation and **A**ir **C**onditioning
LCA	**L**ife **C**ycle **A**pproach
LCC	**L**ife **C**ycle **C**ost
LGWHR	**L**ow **G**rade **W**aste **H**eat **R**ecovery
MVC(c)	**M**echanical **V**apor **C**ompression chiller
NG	**N**atural **G**as
NTU	**N**umber of **T**ransfer **U**nits
ODP	**O**zone **D**eplenting **P**otential
O(F)AT	**O**ne (**f**actor) **A**t **T**ime

O-LCA	**O**rganizational **L**ife **C**ycle **A**ssessment
ORC	**O**rganic **R**anking **C**ycle
QT	**Q**uenching **T**ower
TOE	**T**on (of) **O**il **E**quivalent
SA	**S**ensitivity **A**nalysis
SEER	**S**easonal **E**nergy **E**fficiency **R**atio
STD	**St**an**d**ard
TOE	**T**on (of) **O**il **E**quivalent
TMY	**T**ypical **M**eteorological **Y**ear
UA	**U**ncertainty **A**nalysis
UNEP	**U**nited **N**ations **E**nvironment **P**rogramme
WCD	**W**ater **C**ooled **D**uct
WEN	**W**ater **E**nergy **N**exus
USGS	**U**nited **S**tates **G**eological **S**urvey

CHEMICAL SYMBOLS

CH_4	Methane
CO	Carbon Monoxide
CO_2	Carbon Dioxide
H_2O	Water
Li-Br	Lithium Bromide
NH_3	Ammonia

Chapter 1

Introduction

This chapter describes the scientific background, and in particular the concept of a water-energy nexus. Starting with the main definitions, the concept will be presented and expanded to also take into account carbon emissions. The literature review is also discussed, highlighting how the nexus theme is relevant for industrial settings, but only partially investigated in the literature, how it relates to the equally important subject of waste heat recovery, and which research gaps exist in this field. Based on such gaps, the research questions in this work are posed and the case based approach adopted here is introduced. The chapter ends with a brief introduction to the case studies analyzed and the rationale for their selection and comparison.

1.1 Scientific background and research questions

1.1.1 The nexus between water and energy

The interdependencies between water and energy requirements have long been recognized in the international scientific community [1] and in policy making, especially in the US [2]. Following the Bonn Nexus Conference in 2011 [3], the water-energy nexus has also come into focus in Europe, and an increasing body of research has been developed globally to integrate the traditionally separate issues of water and energy across the spectrum of policy, planning, design and operation [4].

The evaluation of the water intensity of the energy sector, from fuel extraction to energy conversion and distribution, and of the energy intensity of the water sector, including production, distribution and wastewater management, have represented the core of nexus research so far [5]. Empirical and model-based studies have been performed for individual countries and regions, including the US [6, 7], China [8, 9], the Middle East and North Africa [10], Brazil [11], and Thailand [12].

Water is required for nearly all production and conversion processes in the energy sector, including fuel extraction and processing (fossil and nuclear fuels as well as biofuels) and electricity generation (thermoelectric, hydropower, and renewable technologies). Especially for power generation, water is required during plant construction and operation, particularly for cooling and waste heat dissipation at thermal power plants (including biomass based and solar ones).

European studies on the water-energy nexus are relatively few in number, and their focus is more on agriculture and food production, in the framework of the expanded water-energy-food nexus concept discussed within the Bonn Conference [13], or on the other side of the water-energy nexus problem, i.e. energy consumption and water management. Some studies concerning Spain and Germany are mentioned in a literature review concerning energy consumption for water use cycles [14]. A consumptive water footprint of electricity and heat was presented only at an aggregated European level in Ref. [15]. However, individual estimates of water footprints of electricity generation could not be retrieved in literature for individual European countries.

1.1.2 The water-energy nexus in industrial settings

Although manufacturing is one of the sectors where the greatest increases in water consumption are expected in future [16] and accounts for 25.88% [17] of European primary energy consumption, there are few case studies in the literature focusing on the water-energy nexus in industries other than electricity generation. Recent work by Ref. [18] uses input-output analysis to investigate the nexus between water saving and energy conservation for the Chinese industrial sector as a whole.

A framework for extending the energy diagnosis and management approaches of the ISO 50001 standard to industrial water management was recently proposed in Reference [19]. However a recent contribution by Varbanov [20] underlines that the explicit treatment of the nexus in the industrial context is still not apparent, and it mainly appears in the form of the development of process integration methodologies for the simultaneous optimization of water use and energy efficiency in the design or refurbishment of process plants [21].

Indeed, at the operational level of single industries and factories, energy, carbon, and water-related indicators are commonly calculated in industrial LCA studies, but practical case studies discussing the interdependencies of these flows are rare, and are mainly derived from the food [22, 23], and the textile industries [24].

Some informal discussion with industrial specialists carried out by the author during joint industrial heat recovery projects performed during the Ph.D. program have confirmed that, at the industrial level, there is some awareness of the nexus between water and energy consumption. There is also a position paper from the steelmaking industry [25] which expresses a nexus view of the sector, and in that it calls for a holistic approach, enabling the evaluation of additional energy requirements and all environmental aspects when introducing water management policies and evaluating discharge reduction projects. However, at the beginning of the Ph.D. program no case studies on the interdependencies between water and energy consumption could be found in literature on industries other than food, textile and electricity. In late 2017, a paper by [26] highlighted that most energy saving measures in the Chinese steelmaking industry also had benefits for water consumption, but that there were some pitfalls leading to an increase in water consumption. This finding further consolidated the already chosen research path towards the investigation of the water-energy nexus in industrial settings, with special focus on how the links between water consumption and energy consumption are affected by energy saving projects, particularly by low grade waste heat recovery.

1.1.3 Low grade waste heat recovery

Industrial waste heat refers to energy that is generated in industrial processes without being put to practical use [27]. Waste heat sources can be of various types depending on the process and the machinery involved. Examples range from hot combustion gases of internal combustion engines, heated products exiting industrial processes, and heat transfer from hot equipment surfaces [27]. Heat losses and inefficiencies in industrial processes are inevitable. Losses are related to inefficiencies of equipment, thermodynamic limitations and the production process itself. Some losses can be avoided by retrofitting or upgrading the production systems, while others are necessary, especially when related to the manufacturing process e.g. some steel forged elements need to be cooled slowly to avoid internal imperfections, reducing the possibilities of recovering heat efficiently.

Historically, industrial waste heat sources have been classified as low grade if their temperature is below 232 °C [28]. At this point, waste heat recovery flux should be categorized based on temperature. Three categories are used in [28], low, medium, and high quality based on temperature, and are shown in Table 1.1.

TABLE 1.1: Categorized waste heat sources based on temperature.

Type	Temperature
High	$T >650\,°C$
Medium	$232 < T \leq 650°C$
Low	$T \leq 232\,°C$

Materials used in heat exchangers, turbines and more in generally in the high temperature zones, play a relevant role in heat recovery. In particular, the main limitations derive from the exposure to high temperatures and corrosive streams. Material selection is not analysed in depth, though it is considered the second most significant limitation after the theoretical limits described.

A high proportion of industrial excess heat has a low temperature: it is estimated that, globally, 42% of industrial waste heat is available at less than 100 °C, and in particular 28% of global industrial waste heat comes at temperatures between 60 °C and 140 °C [29].

The recovery of low grade waste heat from industries is considered an enabler for CO_2 emission reduction, and recent literature is therefore rich in contributions reviewing recovery technologies for specific processes and industry sectors (see e.g. [25, 30, 31]), and even more recently in works aimed at estimating waste heat potentials at regional, international and global level (see, in particular [29, 32, 33]). While Miró et al. [33] point out that the reliability of such estimates is inevitably limited, the magnitude of potentials is sizeable: for the EU-27, those authors reported values between 1000 and 3000 PJ/year.

Depending on the heat temperature and flow, heat can be reused in multiple ways, e.g. for pre-heating, space heating, absorption cooling and for electricity generation. Suitable technologies for the conversion of low grade heat into power are reviewed for example in [33] [34] [35] [36] [37] [38] [39].

Excess heat can be distributed to district heating systems or exported to other purchasers with a demand for heat, but the economic feasibility of such projects is constrained by the existence of suitable heat sinks within an economically feasible distance. In particular, the number of operation hours at full load is a determinant of the economic viability of heat distribution to users, either directly through heat exchangers, or through active technologies (as defined by [40]) such as mechanical or absorption heat pumps, which allow the transfer of heat from a low temperature heat source to a higher temperature heat sink. In the presence of a demand for cooling, particularly for space or process cooling processes requiring chilled water at about 10 °C, and of waste heat at suitable temperature levels (typically above 70 °C), water/lithium-bromide absorption cooling can also be considered as an active waste heat conversion technology. Brückner et al. [40] report that, assuming an operation time of 2500 h/year, absorption cooling is of little interest for industrial consumers requiring high returns on their investments. However it should be observed that longer operating times are usually required for process cooling applications; this would enhance the attractiveness of this heat recovery technology for industrial users, who represent the major share of the European cooling demand [41].

When heat or cool sinks are not available at a convenient distance, power generation becomes an attractive option. Suitable technologies for the conversion of low grade heat into power are reviewed for example in [37]. Organic Rankine Cycles represent the most economically attractive technology in the typical ranges of industrial heat recovery, in part because of their commercial readiness [38]. Although the efficiency of ORCs at low temperature is necessarily low, "even technologies with low conversion efficiencies can be of interest if there is no other use for the excess heat" [37].

Considering the studies related to:

- Waste heat recovery potentials [42, 43].
- District heating coupled with heat recovery [35, 36]
- Power generation through ORC [37, 38].
- Absorption cooling [39].

It was shown that energy recovery generates overall benefits also for other sources. The literature review discussed above has highlighted two research gaps, which are:

- The lack of studies with a water-energy nexus perspective on industrial heat recovery opportunities.
- The limited range of studies considering absorption cooling as opportunity for low grade waste heat flows usage.

1.1.4 Research questions

The objective of this work is to contribute toward filling research gaps by testing the general assumption that energy recovery generates overall benefits (CO_2 emissions, water, and primary energy reduced consumption) in selected test cases. Thus, to assess the impact of selected energy recovery options, particular attention is paid to low-grade waste heat recovery from cooling systems in for steelmaking and industrial plants. Taking a nexus view, this work will analyze the implications of different economic conditions in various countries and the possible effect of carbon reduction policies on the feasibility of different technology options, and their impact on water and energy consumption. The main research questions, which will be addressed using the test cases, can be summarized as follow:

1. Which forms and uses of Low-Grade Waste Heat Recovery (LGWHR) are economically preferable in process industries, considering internal and external use of heat recovery products?
2. Considering the lack of literature concerning low grade heat recovery for cooling purposes, is this configuration competitive or even preferable to other configurations considering the economic feasibility and the water-energy-GHG perspective?

3. Since it is generally accepted that heat recovery is a good solution for CO_2 reduction and energy efficiency, does LGWHR always generate synergies for the CO_2 footprint, primary energy consumption, and water footprint?

4. Following on from the previous question, how do the performances of LGWHR options depend on local conditions such as market prices, local energy mix and climate?

While work starts from particular cases in steelmaking industry in Italy, the analysis is successively extended to encompass:

- generic industrial low grade waste heat flows
- cross border applications in the Italy-Austria framework
- European applications in the EU-15 context
- a global assessment of some implementations in the 17 worldwide climate zones
- a cross comparison of the selected cases

In this way, the thesis aims to set the basis for answering the research questions from a more general perspective.

1.2 Case studies

The case studies come from Italian industrial plants located in North-East Italy, where options for recovering low-grade waste heat from cooling systems are being evaluated to improve energy and carbon efficiency as well as primary energy consumption.

The case studies are based on a real electric steelmaking plant where heat recovery is supposed to be realized at the cooling plant used for the hot gases of a Electric Arc Furnace (EAF), while in the other case a generic waste heat flow is considered from a industrial facility 6.6 km far from a small town of 11000 inhabitants.

The best available technologies to exploit waste heat at industrial have been analyzed based on scientific literature and industrial technical reports. Low grade waste heat can be used for:

- Power generation through Organic Rankine Cycle [30, 31, 38, 44, 45] or through Kalina Cycle [46].
- Direct heat reuse for hot water, space heating, and low temperature process heating as well as district heating or cooling [25, 47–49].
- Cooling applications using absorption chillers [40].

To better understand the idea of this work a *rich picture* of the problem is given in Figure 1.1. An industrial site where waste heat is available and ready for recovery is shown in red on the left side of the picture. Two ways to use waste heat from industries are shown. The direct dissipation of the excess heat through cooling towers or dry coolers as can be seen on the "dissipation" side, where no heat recovery is performed. The use of excess heat to generate electricity, cooling or district heating is shown on the "recovery" side. All the thermodynamic cycles studied in this work require condensers. Condensation can be performed taking water from the sea, circulating it through the plant heat exchangers and returning it to the local source. This method is characterized by high water withdrawals, but relatively low water consumption. The typical plants studied in this thesis on the other hand, are relatively small systems located close to end markets, usually in inland areas, which often use closed-loop cooling systems. Cooling systems commonly used for this purpose are cooling towers (CTs) and dry coolers (DCs), and consequently all the configurations proposed are linked to these dissipation systems.

When heat recovery is not performed, cooling, electrical, and heating demands are supplied using electricity from the national energy mix, via a mechanical vapor compression chiller, the national energy mix, and a boiler respectively. This configuration is essentially the base case, where all energy demands are supplied with the conventional technology. Base case is represented on the right side of the colored logos indicating the types of demand and the related technologies utilized. The national energy mix and the conventional technologies are also used during heat recovery, in fact it can happen that heat recovery configurations cannot supply the demand completely. Also, condensers use electricity provided by the national mix during operation.

The national energy mix is shown in the dashed-line box. Every nation has its own mix, and consequently depending on the technologies used

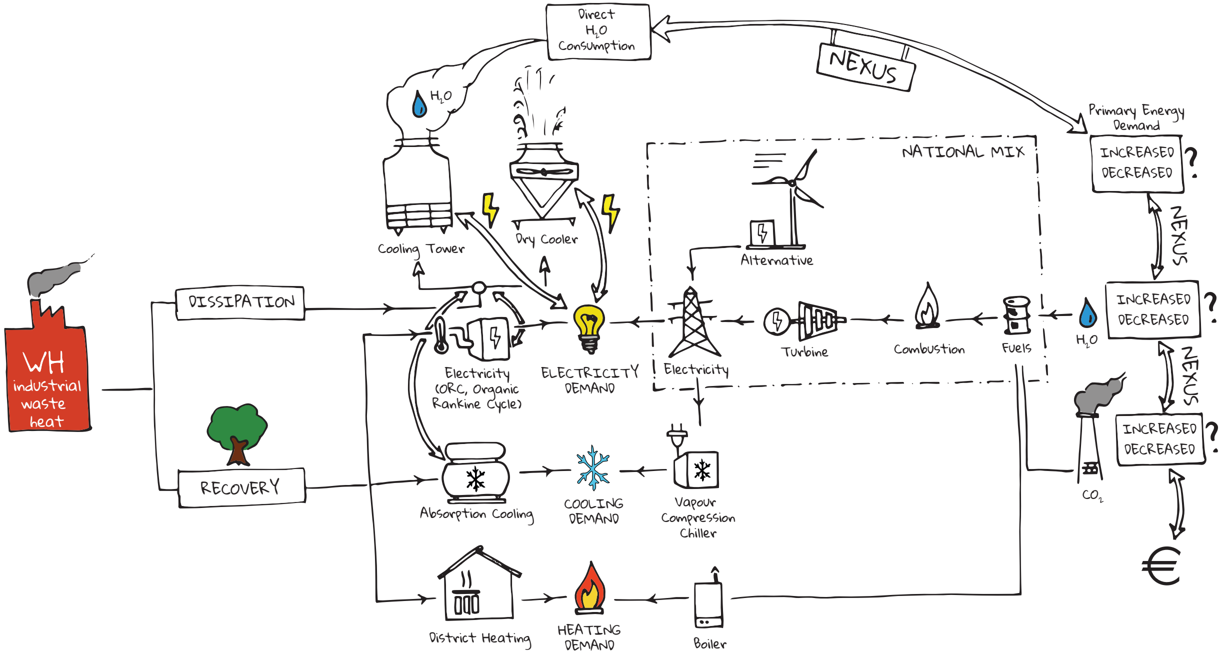

FIGURE 1.1: Rich picture of Water-Energy-GHG nexus perspective on case studies.

for electrical energy generation, has a different consumption of fossil fuels, water, and emissions of GHG gases. The *indirect* use/emissions of the water, electricity and CO_2 will be explained in Chapter 3.

The nexus indicators are represented by the set of arrows on the top and right of the picture. Based on each configuration chosen, analyses of these indicators have been performed in order to answer the research questions. Moreover, in order to take into account different climates and national energy mixes, this picture can be "moved" over different nations thus helping with understanding the effects of climates and markets in the analyzed configurations.

In order to answer to the research questions and get a generalization of the results, the cases introduced in Table 1.2 have been evaluated in different countries with the aim to assess the effects of local conditions such as market prices, local energy mix and climate.

As was already introduced a common element required in most of the thermodynamic cycles, and therefore also in low-grade heat recovery cycles, is the condenser. Condensers are heat rejection devices used to reject heat when heat transfer fluid is required to be at the liquid state, e.g. before the expansion valve and evaporator in a refrigeration cycle, or when they are not used in refrigeration cycles, industrial process fluid cooling applications are very common.

In the heat recovery configurations analyzed in this work two types of condensers are considered:

- Evaporative towers or *cooling towers* (CT) use the evaporation of water to remove process heat and cool the working fluid to near the wet-bulb air temperature.
- Forced air coolers or *dry cooling* towers (DC) use the outside air to cool the working fluid to near the dry-bulb air temperature.

To the best of the author's knowledge, these are the main devices used in heat recovery projects.

This leads to the introduction of the three configurations analyzed in this work, which are summarized in Table 1.2 where case studies are specified with letter C followed by the relative number. As can be noted, all the case studies have a base case in which heat recovery is not performed, and it is used for economic and environmental comparison. Along the first row of the table, cooling configurations of heat recovery are listed. The use of waste heat for cooling purposes can be performed through an *absorption refrigerator*. Electricity is produced through an Organic Rankine Cycle (ORC). The last column refers to waste heat recovery coupled with district heating. The acronym shown in the last column of the first row stands for district heating (*DH*) with an absorption chiller for cooling (*C*) demand resulting in *DH+C*.

Case study *C1a* is the first study proposed in Table 1.2. Cooling and electricity configurations are analyzed and compared with the base case ("no HR") in order to give an assessment of the techno-economic feasibility. The focus in this case is more on economic assessment in different countries with different economic conditions i.e. water, electricity prices and carbon taxes than on the thermodynamic behavior of the systems. The objective of this case is therefore to answer to the first and second research questions, also introducing a link to the third and the fourth questions through the introduction of the water-energy GHG perspective. For this case, economic data for significant number of countries were collected and analyzed in order to understand the economic influence, as well as how effects of the national energy mix relate to the consumption of primary energy, blue water, and the carbon dioxide emissions.

Case study *C1b* can be seen as an in-depth analysis of case study *C1a*, where the focus is more on the thermodynamic behavior of the heat recovery configurations than on the economic analysis. This case is developed in order to understand in more detail the effects of climates on different configurations. Climates ranges from very hot and humid, e.g. Singapore, to very cold e.g. Östersund (Sweden) and the effects on energy and water consumption are analyzed more in detail.

Finally, the last case study, *C2*, is used to understand and give an overview of all the effects related to the heat recovery configurations, taking into account a new parameter: the possibility of supplying an external energy demand.

Within the frame of an INTERREG Italia - Österreich project that funded this research, Italy and Austria are used as pilot cases to understand the implications of different low grade waste heat recovery configurations.

All the case studies are analyzed using either the dry cooler or cooling tower as condensers/heat rejection devices. The main differences between the two condensers are not just to do with the method of heat dissipation, but also in the electricity and water consumption during the operation. Differences between the case studies are also related to the internal (on-site) or external (e.g. remote user - district heating) use of the heat recovery products. In particular case studies *C1a* and *C1b* look at fulfilling the internal demand for electricity and cooling, while case study *C2* looks at the external use of heating and cooling.

TABLE 1.2: The case studies analyzed in this thesis and the related placement depending on main output and condenser type.

Low Grade Heat Recovery Opportunity

<table>
<tr><th rowspan="2">Condenser type</th><th colspan="2">No HR</th><th colspan="2">Cooling</th><th colspan="2">Electricity</th><th colspan="2">*DH+C</th></tr>
<tr><th>DC</th><th>CT</th><th>DC</th><th>CT</th><th>DC</th><th>CT</th><th>DC</th><th>CT</th></tr>
<tr><td rowspan="3">Case study</td><td>C1a</td><td></td><td colspan="2">C1a</td><td colspan="2">C1a</td><td colspan="2"></td></tr>
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*District Heating (DH) with Cooling (C) using an absorption chiller

Chapter 2

Technical Background

This chapter gives an overview of the main components of the technologies identified in the introduction that are used in low-grade waste heat recovery configurations. Figure 1.1 is used as guideline to introduce the components and the cycles of interest. First, particular attention is paid to the main heat rejection devices, which are relevant for this study because some of them are water-intensive. Then, the heat recovery technologies for heating, cooling, and electricity production are examined, followed by an examination of the conventional technologies used for the same purposes.

2.1 Technologies for heat dissipation

In the first branch of Figure 1.1 dissipation devices are introduced. Dry coolers and cooling towers are used both for condensation purposes and for the purpose of cooling process fluids. Dry coolers will be presented in Section 2.1.1, while cooling towers are presented in Section 2.1.2.

2.1.1 Dry coolers

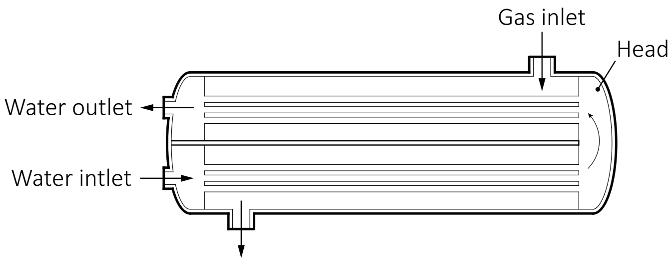
A dry cooler (DC) usually refers to a device used for heat rejection of waste heat into the atmosphere using outside air. DCs are used widely for industrial cooling, process cooling, data center cooling and air conditioning (usually with mechanical vapor compression chillers). In this case refrigerant condensation is performed by the refrigerator using a water cooled condenser, which is a heat exchanger where in one side refrigerant is condensed, and in the other side, cooling water from a dry cooler is consequently heated through gas condensation. Figure 2.1 shows a dry cooler used for industrial applications while Figure 2.2 shows a water cooled condenser.

FIGURE 2.1: Industrial dry coolers.

A dry cooler consists of a series of coils in which the heat-transfer fluid flows, exchanging heat with an air flow forced by a propeller fan. The coils can be v-shaped or horizontally arranged, the former used for high heat rejection ratios. The coils are usually equipped with copper tubes and aluminium fins. Special devices are also present in order to reduce noise emission at the propeller.

2.1.1.1 Heat-transfer process in dry coolers

The heat-transfer process between air and heat-transfer fluid occurs in the coil, where the fluids flow in a combined counterflow and cross-flow arrangement. During the DC selection, and therefore in order to evaluate the heat exchange rate, the air entering the condenser coil is assumed to be at a temperature equal to the summer outdoor design dry-bulb temperature T_{db}. In a typical meteorological year (TMY), which is taken to be a year of average weather, T_{db} is the temperature evaluated under the worst conditions (e.g. highest temperature encountered with a certain frequency). In



(a)

(b)

FIGURE 2.2: (a) A generic schematic of a water cooled shell tube condenser; (b) A water cooled condenser with a cut-out on the shell and one head removed for educational purposes. Power rating of 31 kW.

case the air-cooled condenser is installed on the roof, an additional 1.5 to 3 °C should be added to the summer outdoor design dry-bulb temperature to account for the temperature increase of the roof due to solar heat [50].

Given the total heat refrigeration load it is possible to evaluate the air flow rate required and the power required by the motor used by the propeller fan. The air flow rate can be evaluated as:

$$V_a = \frac{\dot{Q}_{rej}}{\rho c_p (t_1 - t_2)} \tag{2.1}$$

where:

- V_a is the volumetric flow rate in m^3/s
- $\dot{Q}_{rej}$ is the refrigeration load at the evaporator
- c_p is the specific heat capacity of the air at average temperature of 30 °C
- ρ is the air density at 30 °C: 1.165kg/m^3

The inlet air temperature t_1 is considered to be the maximum value reachable by the air in a given region, while the temperature t_2 is chosen between 3.0 and 5.5 °C under the condensation temperature. The air flow rate varies between 285 and 570 m^3/(h kW) [51].

2.1.1.2 Air-cooled condenser

Air-cooled condensers are similar in construction to a dry cooler except that the working fluid is a refrigerant. The condensation of the refrigerant takes place using the atmospheric air from outside. The latent heat of condensation of the refrigerant is extracted by a condenser coil where outside air is used as coolant, leading to an increase in the air temperature.

In the condenser coil it is possible to distinguish between the main condensing coil section and the subcooling coil section. These two sections are connected in series and are composed of several refrigerant circuits to condense the refrigerant into a liquid. Figure 2.3 shows the construction of a typical air-cooled condenser for air conditioning applications. The condenser coil is usually equipped with copper tubes and aluminium fins when halocarbon is used as refrigerant [50].

Cooling air is usually forced through the condensing coils by a fan which is usually located downstream in order to provide an even airstream in order to promote the heat exchange. A propeller fan has a lower fan total pressure and large volume flow rate, which makes it more suitable for air-cooled condensers. In large air-cooled condensers, condensing and subcooling are usually located on two sides, and the propeller fans and dampers are not at the top of the unit [50], while for industrial dry coolers the propeller fan is usually located at the top of the unit.

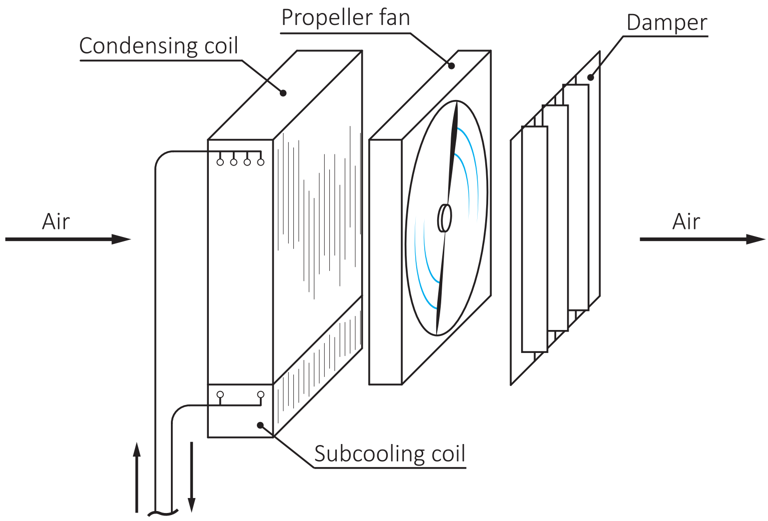


FIGURE 2.3: Air cooled condenser typical constriction.

The use of water-cooled condensers (Figure 2.2(b)) and dry coolers allows the placement of the heat rejection units far from the condensers (usually mounted near the chiller). From the plant engineering point of view this solution allows separation of the heat rejection zone from the refrigeration zone, connecting them through service lines (pipes).

2.1.1.3 Selection of air-cooler condenser or a dry cooler

The performance of heat rejection $\dot{Q}_{rej}$ varies depending on the cooling air intake temperatures. During the design of a cooling plant the selection of the heat rejection device should proceed with the following steps:

- Evaluate the total load at the condenser of the refrigerating system.
- Select the air-cooled condenser from the catalogue with the required $\dot{Q}_{rej}$ at a specific cooling air intake temperature with the appropriate T_{ev}.
- Take into account the problems related to the noise generated by the operation of fans. Dry coolers with reduced noise level are more expensive.
- Take into account the climate influence: in hot climates with a frequently high dry bulb temperature, the $\dot{Q}_{rej}$ could be reduced requiring an increased device size.

2.1.1.4 Electricity consumption

Beyond just a thermodynamic approach that explains how DC works, for the purposes of this work an analysis related to electricity consumption during operation is required.

The electricity consumption of DC is mainly related to the operation of the electric motor moving the fan. Catalogues, besides providing the heat rejection rate, air/fluid flow rates and test temperature, usually give the electrical power of the motor installed, which depends on the model.

The operation of a dry cooler is not always at the maximum power: electric motor speed control is controlled by an inverter, improving energy efficiency and economizeing the total cost related to energy consumption. Table 2.1 shows the correlation between the installed motor power and the heat rejection capacity.

2.1.2 Cooling towers

A cooling tower (CT) is a heat rejection device that rejects waste heat to the atmosphere may using either the evaporation of water to remove process heat and cool the working fluid to near the wet-bulb air temperature. There are two types of CTs:

- closed circuit cooling towers
- open circuit cooling towers

TABLE 2.1: Electric motor power and heat rejection capacity data, collected from LU-VE dry coolers catalogue, XDHL series [52].

$\dot{Q}_{rej}$ kW	P_{elec} kW
86	3.25
174	6.50
264	9.75
537	19.50
870	26.00

In closed circuit CTs, there is no contact between the air and the fluid being cooled. This tower has two separate fluid circuits, one in which the fluid is recirculated on the outside of the second circuit, which is a bundle of tubes through which the hot water is flowing. The air drawn through this cascading water provides evaporative cooling similar to that of an open cooling tower, except that the cooled water never makes direct contact with the air. These types are used for refrigeration purposes. An evaporative condenser (Figure 2.6) may be considered a closed circuit CT, since there are two separated circuits, one for the refrigerant, and one the recirculating cooling water. This arrangement is necessary to avoid refrigerant losses to the atmosphere.

An open circuit CT (Figure 2.4) is an enclosed structure that distributes warm water over a labyrinth-like packing or "fill", which provides an expanded air-water interface for heating of air and evaporation to take place. The water is cooled as it falls through the fill, and is then collected in a cold water basin below. The heated moisture-laden air leaving the fill is discharged into the atmosphere. These types of cooling towers are used for industrial cooling purposes, applications could be found in steelmaking industry as shown in chapter 4.

2.1.2.1 Operating principle

In CTs heat transfer between water and air takes place not only through conduction and convection, but additionally and for the most part through

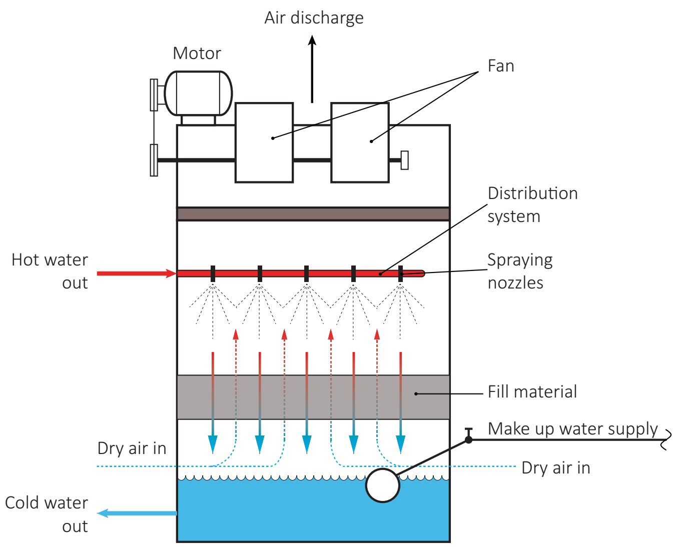


FIGURE 2.4: Cooling tower, open loop.

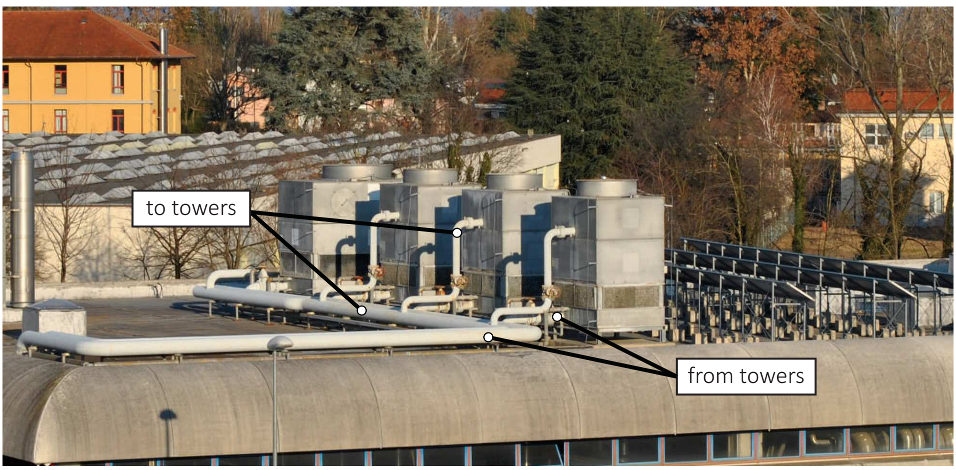


FIGURE 2.5: Cooling towers located on the roof of the air conditioning plant's building. Pipes carrying water being sent to the towers and pipes carrying water returning to the system are labelled.

a mass transfer process. The quantity of heat transferred from water to air is much higher with such a method than with conduction and convection.

CTs consist of a condensing coil, a water spray bank, a forced-draft or induced-draft fan, an eliminator, a water basin, and a casing. Figure 2.4 shows a fan-induced draft, counter-flow cooling tower.

In the evaporative condenser, other than the elements making up a CT, a condensing coil and a circulating pump are also present. Figure 2.6 shows an example. Water is sprayed over the outside surface of the condensing coil. Because of the evaporation of water, heat is extracted through the wet surface of the coil. The remaining spray falls and is collected in the water basin located at the bottom of the tower. Air enters from the inlet located just above the water basin. It moves up through the condensing coil, spray nozzles, and water eliminator used to separate water droplets from the stream, is extracted by the fan, and finally discharged at the top outlet in a counterflow arrangement. Other airflow and water flow arrangements have also been developed. The condensing coils are usually made of bare pipes of copper, galvanized steel, or stainless steel. Due to the high heat-transfer coefficient of the wet surface, fins are not required to increase the surface area.

2.1.2.2 Technical terms

The following terms are commonly used when referring to a cooling tower or an evaporative condenser:

- *Approach*: The temperature difference between cooled water leaving the tower and the wet-bulb temperature of the air used as coolant.
- *Blowdown*: Water discharged periodically to avoid buildup of dissolved solids.
- *Fill*: The heat-transfer surface within the tower. Hot water from the condenser or coil is sprayed along the fill, which increase the heat exchange between water and air, and is collected in the water basin.
- *Makeup*: Water added to the circulating water to compensate for the loss of water to evaporation, drift, and blowdown.
- *Range*: The temperature difference between the water entering and water through the cooling tower.

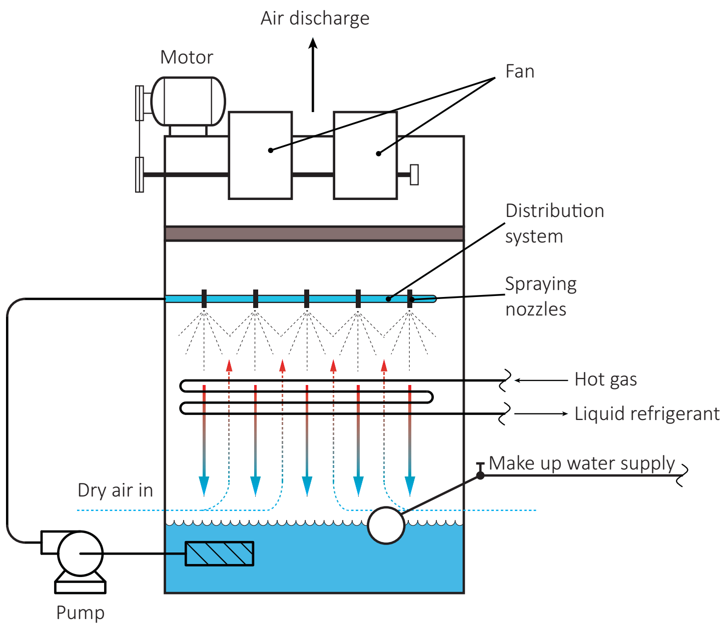


FIGURE 2.6: Evaporative condenser typical construction.

2.1.2.3 Tower coefficient

Thermal analysis of cooling towers can be performed with a model developed by *Baker and Shryock* [53]. The simplifications in this model are that the heat energy difference between the make-up water and the blowdown and the drift losses are ignored, and that the increase in enthalpy of the water from the addition of liquid water is ignored. The resulting energy balance between water and air can be evaluated as:

$$\dot{m}_w c_{pw} dT_w = \dot{m}_a dh_a \tag{2.2}$$

where:

- $\dot{m}_w$, $\dot{m}_a$ = the mass flow rate of water and air, (kg/s)
- c_{pw} = the specific heat of water
- T_w = the temperature of condenser water
- h_a = the enthalpy of air

Then, by introducing another simplification: ignoring the thermal resistance of the saturated air film that separates the condenser water and the air-stream, the combined heat and mass transfer from the air-water interface (the saturated air film that surrounds the condenser water droplets) to the bulk air-stream can be evaluated as:

$$\dot{m}_a dh_a = K_m (h_s - h_a) dA \tag{2.3}$$

where:

- K_m = the mass-transfer coefficient
- h_s = the enthalpy of saturated air film
- A = the surface area at air-water interface

In Equation 2.3, the change of enthalpy, or total heat of air, consists of changes in sensible heat and latent heat. Consider a cooling tower with a fill volume V and a contact surface area $A = aV$, (m^2). Here a is the surface area of fill per unit volume. Also let $K = K_m / c_{pw}$. Then combining Equations 2.2 and 2.3, we get:

$$\frac{KaV}{\dot{m}_w} = \int_{T_{w1}}^{T_{w2}} \frac{dT_w}{h_s - h_a} \tag{2.4}$$

The CT's tower coefficient, generally known as the number of transfer units - NTU, is the integrated value of Equation 2.4.

2.1.2.4 Factors that affect cooling tower performance

During the selection of a cooling tower, technical and economic aspects are connected. From the technical point of view, the selection of a water-cooled condenser gives preference to a high efficiency and heat rejection rate, as well as economic factors. This results in a minimization of the total power consumption of the compressors, condenser fans, and condenser water pumps in the device. Proper selection of tower parameters like: range, water-air ratio, approach, fill configuration, and the water distribution system directly affects the performance of a cooling tower and therefore indirectly affects the performance of the refrigeration system. A cooling tower is rated with the following indicators (typical values shown) [50]:

- water circulation rate: 0.014 L/s per kW of heat rejected
- temperature of water entering the condenser: 35 °C
- temperature of water leaving the condenser: 30 °C
- outdoor wet-bulb temperature T_o':26 °C
- range: 5 °C
- approach: 4 °C

2.1.2.5 Energy consumption

Electricity in a cooling tower is mainly consumed by the motor fan and to a much lesser extent by the control system. As was investigated for dry coolers, electric motor power relative to the heat rejection rate is now presented and compared. As for cooling towers, electric motor speed control is usually controlled by an inverter in order to improve energy efficiency and economize the total cost related to energy consumption. Table 2.2 shows the electric motor power based on the heat rejection rate.

TABLE 2.2: Electric motor power related to heat rejection rate, YWCT® catalogue for cooling towers, P Series [54].

$\dot{Q}_{rej}$ kW	P_{elec} kW
104	1.49
140	2.24
256	2.98
314	4.10
465	5.59

Comparing Tables 2.2 and 2.1 shows the energy impact when using selected condensers. A graphical representation is shown in Figure 2.7, where it can be seen that dry coolers are more energy intensive compared to cooling towers due to the higher power requirements of the propeller fan motor. Results shown in Figure 2.7 are based on a qualitative assessment, obtained by the comparison of two manufacturer catalogues. Even if this analysis is limited to two catalogues/series, it is helpful in understanding the difference in magnitude between the maximum power required.

2.1.2.6 Water consumption

Although cooling towers require less power than dry coolers, CTs still require water during operation. To dissipate residual heat, the consumed water $\dot{m}_{ev}$ can be estimated as a function of evaporated water (equation 2.5):

$$\dot{m}_{ev} = k\frac{\dot{Q}_l}{L} \tag{2.5}$$

L being the latent vaporization heat of water (here taken to be 2.200 kJ/kg), $\dot{Q}_l$ the thermal load in kW and resulting W being expressed in kg/s. Coefficient k accounts for additional water losses due to bleed off and drift. Since the water is recycled and there is an opportunity for water constituents to be concentrated in the evaporative step, bleed off of high mineral water

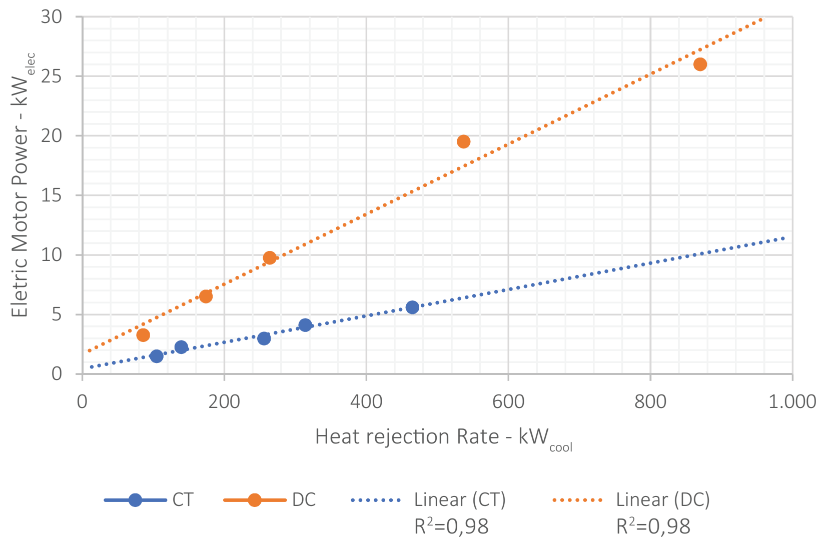


FIGURE 2.7: Electric motor power comparison CT vs DC.

and makeup with freshwater of acceptable quality is required to keep solid concentration in water circuits below an acceptable threshold [55].

2.2 Technologies for low grade heat recovery

In the second branch of Figure 1.1, *low grade* waste heat recovery opportunities are introduced. This section provides an introduction to the waste stream composition and the technologies used for energy conversion.

2.2.1 Waste stream composition

Heat recovery involves many types of streams, ranging from electric arc furnace off-gases, to liquids such as hot water, to more "noble" internal combustion engine off-gases. Although from the theoretical point of view, the chemical compositions do not directly influence the reusability of the available energy, the stream composition could affect the effectiveness in the recovery process and material selection. Thermal conductivity and heat capacity affect the heat exchanger effectiveness, and can vary substantially depending on the composition and phase of waste heat streams. The process-specific chemical make-up of off-gases will have a significant impact on heat exchanger designs, material constraints, and costs [27].

The deposition of any fouling substances on the heat exchanger surfaces can narrow the flow channels, increase the pressure loss, and reduce the heat transfer rate. As shown in Table 2.3, denser fluids have higher heat transfer coefficients enabling higher heat transfer rates per unit area for a given temperature difference.

TABLE 2.3: General range of heat transfer coefficients for sensible heat transfer in tubular exchangers [56].

Fluid Conditions	Heat Transfer Coefficient $W/(m^2\,K)$
Water Liquid	$5.0 \cdot 10^3$ to $1.0 \cdot 10^4$
Light organics, liquid	$1.3 \cdot 10^3$ to $2.0 \cdot 10^3$
Gas (P=1.000 kPa)	$2.5 \cdot 10^2$ to $4.0 \cdot 10^3$
Gas (P=100-200 kPa)	$8.0 \cdot 10^1$ to $1.2 \cdot 10^2$

Waste heat streams can vary in mass flow rate and temperature, these two indicators can be used to assess the heat availability and *quality*. Heat quantity can be expressed as:

$$\dot{Q} = \dot{m} \cdot \Delta h(T) \tag{2.6}$$

where $\dot{Q}$ is the waste heat loss, $\dot{m}$ is the mass flow rate, h is the enthalpy expressed as a temperature function.

The concept of waste heat varies among different industries depending on the processes involved and the stream quality. A variety of heat exchangers specifically designed for heat reuse can be found in industrial applications, examples are: recuperators, regenerators, rotary regenerators/heat wheels, passive air pre-heaters, regenerative/recuperative burners, economizers, and waste heat boilers.

The main candidates are metalworking industry (particularly high temperature furnace gases), hydrogen plants, and the glass industry. In these cases heat streams are consistent and at a high temperature and flow rate, but there is also a significant amount of energy associated with low temperature waste heat, at temperatures below 232 °C. Existing systems are mostly used with high and medium temperature waste heat. This work instead focuses on the heat recovery of low temperature heat streams.

Most of waste heat is in the low-temperature range [27], and the challenges related to recovering heat in these conditions faces problems such as heat exchanger size, since large surfaces are required for heat transfer. The correlation can be expressed by the following formula:

$$\dot{Q} = UA\Delta T \tag{2.7}$$

where $\dot{Q}$ is the heat transfer rate in W, U is the heat transfer coefficient in $W/(m^2\,K)$, A is the surface area for heat exchange in m and ΔT is the temperature difference in °C or K.

Equation 2.6 alone does not provide a full indicator of heat reusability, in fact, the temperature is used as another indicator. Due to the second law of thermodynamics, in order to achieve high efficiency in thermal cycles waste heat temperature must be higher than the temperature of the heat sink. The Second Law of thermodynamics can be expressed by the following formula:

$$\eta_{Carnot} = 1 - \frac{T_c}{T_h} \tag{2.8}$$

where T_h is the temperature of the waste heat flow, and T_c is the heat sink temperature. It can be noted that theoretical efficiency can be improved by increasing the waste heat temperature or by reducing the heat sink temperature, though this is limited by material resistance at high temperatures and the external environmental temperature T_c. Equation 2.8 represents the so called *Carnot efficiency*, and is the maximum possible efficiency of a cycle at a given temperature.

2.2.2 Low-temperature energy recovery options

When recovered heat is not used for internal processes e.g. pre-heating, or the temperature does not allow reuse, alternative configurations can be summarized as:

a) Power generation through Organic Rankine Cycle [30, 31, 38, 44, 45] presented in section 2.3, Kalina cycle is not studied in this work.

b) Direct heat reuse for hot water, space heating, and low temperature process heating as well as district heating or cooling [25, 47–49], presented in section 2.5.

c) Cooling applications using absorption chillers [40], presented in section 2.4.

While heat and power generation configurations are a well-established opportunity for heat recovery, absorption cooling is still under evaluation and within the scope of this work a techno-economic feasibility analysis will be performed.

2.3 Organic Rankine cycle

The organic Rankine cycle (ORC) involves the same elements used in a standard steam power cycle, except the working fluid is an organic fluid characterized by a lower boiling temperature than water. Organic fluids allow the production of electricity using low heat source temperatures e.g. biomass combustion, industrial waste heat, geothermal heat, or solar ponds.

As illustrated in Figure 1.1, ORC is the first heat recovery opportunity studied and used to exploit heat for electric energy production. Condensation is still required, and this is why ORC is connected with either with DC or CTs in the abovementioned figure.

2.3.1 Working principle of ORC

In an organic Rankine cycle (ORC) four main stages of the working fluid can be distinguished: boiling, expansion, condensation and pumping. The organic compounds involved are typically a refrigerant, a hydrocarbon (butane, pentane, hexane, etc.), a silicon oil, and a perfluorocarbon. Organic working fluid is pumped from a low to high pressure, entering the heat exchanger (steam generator in the standard Rankine cycle) where it is heated at a constant pressure by an external heat source. After heating, the working fluid at vapor state expands through the turbine generating power, then for the cycle repetition the organic fluid is condensed. Figure 2.8(a) gives the scheme of an ORC plant.

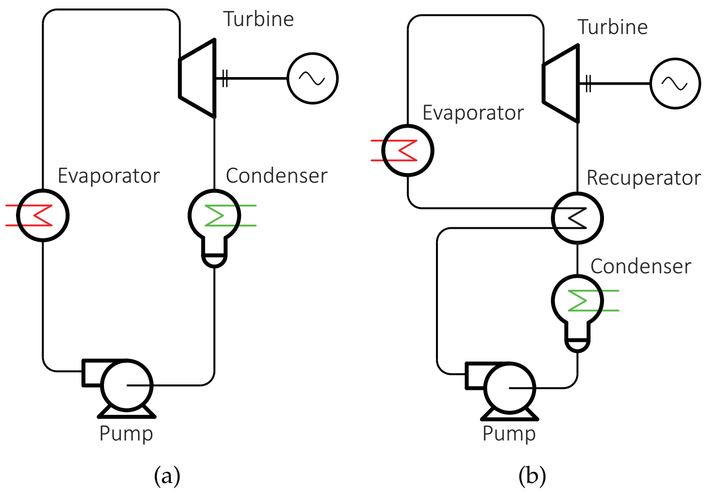


FIGURE 2.8: ORC configurations: (a) simple ORC scheme; (b) ORC with recuperator.

For some organic fluids some residual heat is available after expansion. With such types of fluids a recuperator can be useful fot recovering heat in order to pre-heat the working fluid before the evaporator, reducing the heat required during evaporation. Figure 2.8(b) shows the configuration described.

The layout of the organic Rankine cycle is somewhat simpler than the steam Rankine cycle: there is no water-steam drum connected to the boiler, and one single heat exchanger can be used to perform the three evaporation phases: preheating, vaporization and superheating. Variations on the cycle architecture are also more limited: reheating and turbine bleeding are generally not suitable for the ORC cycle [57].

The organic fluids used have a lower boiling temperature than water. Refrigerants and hydrocarbons are the two most commonly used, and some examples are now given:

- Hydrofluorocarbons (HFCs), e.g. R134a, R245fa, which contain fluorine and hydrogen atoms. These types of refrigerants do not harm the ozone layer, but contribute to global warming.

- Hydrocarbons (HCs), organic compounds consisting entirely of hydrogen and carbon. These types are flammable, common by-products of gas processing facilities e.g. isobutane, pentane, propane.
- Fluorocarbon or sometimes referred as perfluorocarbons (PFCs) containing only carbon and fluorine. These types are not flammable.
- Chlorofluorocarbon (CFCs and HCFCs), also commercially known as Freon© Dupont e.g. R-11, R-12 and HCFCs e.g. R-22, R-123. All these refrigerants are now banned.

2.3.2 ORC applications

The efficiency of ORC, η_{ORC}, is quite low when compared with steam cycles, and it ranges from 3.5 to 25% [57–60] depending on size, organic fluid used, and heat source temperature. The efficiency is evaluated as:

$$\eta_{ORC} = \frac{\dot{W}_{net}}{\dot{Q}_{evp}} \tag{2.9}$$

where:

- $\dot{W}_{net}$ is the net electric output,
- $\dot{Q}_{evp}$ is the heat power exchanged at the evaporator.

The main advantage of an organic working fluid is the ability to exploit low temperature heat sources. The boiling temperature is lower when compared with water, allowing electricity production with heat source temperatures slightly lower than 300 °C, while steam cycles need steam temperatures exiting the generator of 500 °C.

Waste heat recovery is one of the most important development fields for the Organic Rankine Cycle. Many manufacturing and process industries (e.g. the cement industry, steelmaking industry, metalworking, refineries or chemical industries) rejects heat at relatively low temperature. For medium to large plant size these flows are particularly abundant, allowing heat recovery [61].

ORC can be implemented on biomass plants where heat available from the flue gases can be used as heat source through a heat transfer fluid and a heat exchanger. The heat transfer fluid is usually a thermal oil which is directed to the evaporator heat exchanger at the ORC loop, at a temperature slightly lower than 300 °C. The condensation water is used for hot water production.

Other applications of ORC are: geothermal plants, solar power plants, heat recovery on internal combustion engines.

2.3.3 Comparison with the steam Rankine Cycle

T-s diagrams of saturation curves can be used to understand the different implications of steam cycles and organic cycles. Figure 2.9 shows the saturated vapor curve for come organic fluids and water. It can be observed that:

- With organic fluids, superheating before the turbine inlet is not required. The slope of the right part of the curve is completely different from water, in particular after the critical point the slope of the curve tend to vertical. As a consequence, the limitation of the vapor quality at the end of the expansion process disappears in an ORC cycle.
- The entropy difference between saturated liquid and saturated vapor is much smaller for organic fluids. This also means that the enthalpy of vaporization is smaller. Therefore, for the same thermal power through the evaporator, the organic working fluid mass flow rate must be much higher than that of water, leading to higher pump consumption[57].

While for steam cycles, turbines are already standardized with a solid existing body of knowledge, ORC turbines are under development, and there are some problems with design and implementation. On the other hand, ORC turbine are single or two-stage turbines leading to a cost reduction when compared with steam turbines that usually have several stages due to the pressure ratio and enthalpy drop. Turbines used in ORC applications are: axial turbines, radial inflow turbines and, scroll expanders.

Using water as working fluid is convenient since it is economic, widely available, non-toxic, non-flammable and ozone-friendly. On the contrary, organic fluids are not safe for the environment due to a high Global Warming Potential (GWP) and Ozone Deplenting Potential (ODP).

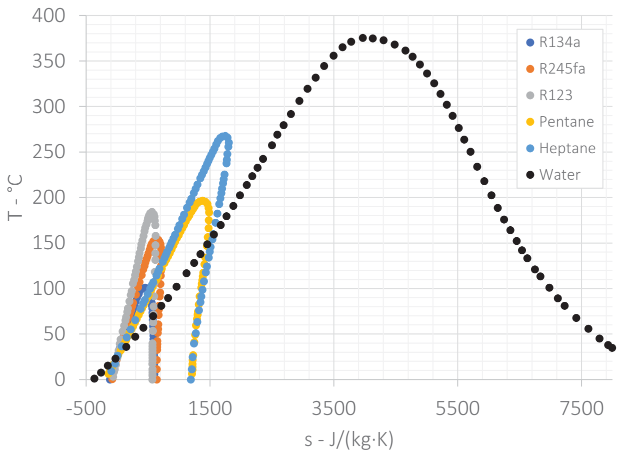


FIGURE 2.9: T–s diagram of water and various typical ORC fluids [62].

2.4 Absorption refrigeration

Absorption refrigeration is the second technology used to exploit waste heat analyzed. It is shown in Figure 1.1 between ORC and district heating. There is little interest in the literature for this technology, though Sarah Brückner et. al [40] have studied this it in an industrial application. This section provides an introduction to refrigeration cycles, in order to understand the difference between absorption refrigeration and a conventional mechanical vapor compression refrigerator. Additionally, the thermodynamic performance and technical characteristics of absorption refrigeration are given.

2.4.1 Introduction to refrigeration cycle

Refrigeration cycles are cyclical thermodynamic transformations that occur in machines called "inverse machines", which are used to extract heat from an environment and discharging to another. The term "inverse" comes from the fact that in these machines, usually called refrigerating machines, work is exploited instead of being produced. An inverse machine can be described as shown in Figure 2.10.

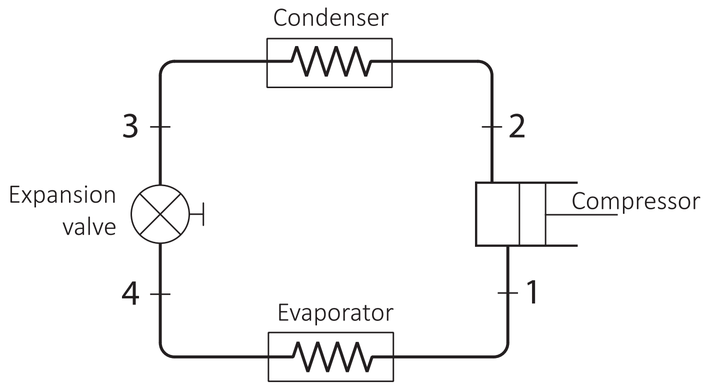


FIGURE 2.10: Refrigerator main components.

The most significant phase in a refrigeration cycle is the compression of the refrigerant. There are two methods used to increase the pressure of the refrigerant. One consists of the compression through a mechanical device (compressor), while the other method used is the compression through a cyclical series of chemical reactions (absorption). The machines used are:

- *Mechanical Vapor Compression Chillers* (MVC), which provide the compression of the refrigerant through a mechanical device, i.e. a compressor using electric or mechanical power.
- *Absorption Chillers* (ABS), which use the dissolution heat of a solute in a solvent which is cyclically concentrated and diluted to drive the cooling process. This type of refrigerator requires a heat source instead of electric or mechanical power.

The thermodynamic cycle is the same for both methods analyzed. Figures 2.11(a) and 2.11(b) show the thermodynamic cycle on T-s and p-h diagrams. The refrigerating effect derives from the evaporation of the refrigerant in the evaporator shown in Figure 2.11 between state point 4 and 1. Refrigerant at gas state is then extracted by the compressor at point 1 and is compressed isentropically until state point 2. At the condenser, between state point 2 and 3, refrigerant is cooled and then condenses and the latent heat of condensation is rejected into the external environment (heat sink). The liquid refrigerant is then expanded, between state point 3 and 4 by expansion valve, which reduces it to the evaporating pressure. Dotted line between point 3 and 4 indicates that the throttling process at the expansion valve is an irreversible process [50].

2.4.2 Absorption chillers

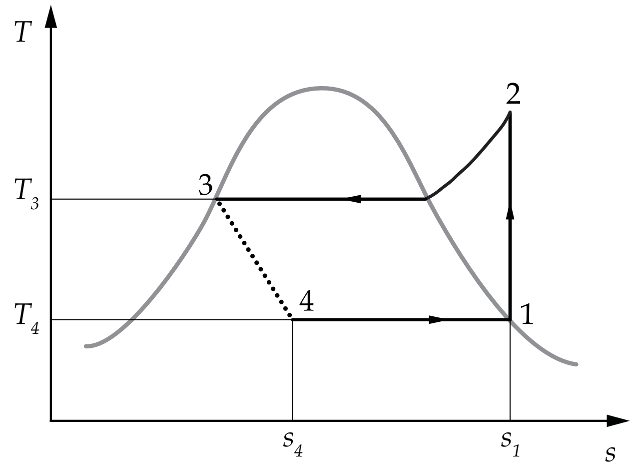
In the absorption cycle the compressor is replaced by an absorber, a generator, and a small pump, giving what is represented in Figure 2.12.

The advantage of an absorption chiller is the type of energy required to drive the cooling process. A source of heat is required instead of the electric energy used at the compressor, making this machine applicable for heat recovery purposes.

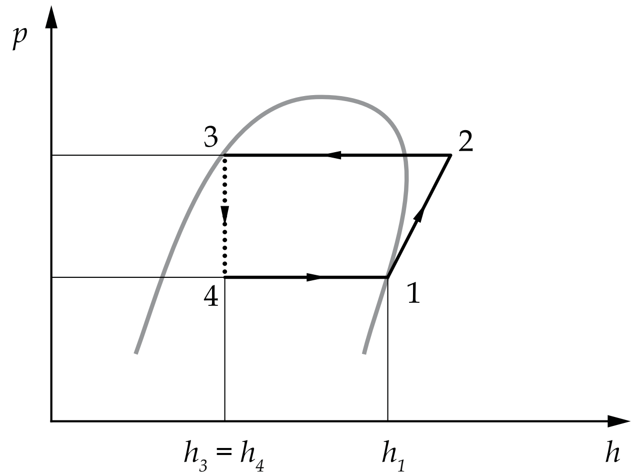
An absorber dissolves the refrigerant in a suitable liquid, the pressure of which is increased and moved by pump to a generator, where as a result of heat addition the refrigerant evaporates in the high-pressure side. Some work is required by the liquid pump but, for a given quantity of refrigerant, it is much smaller than what is needed by the compressor in the vapor compression cycle..

In an absorption refrigerator, a suitable combination of refrigerant and absorbent is used. One combination is ammonia (NH_3), used as refrigerant, and water, used as absorbent. This working fluid is used in a wide range of applications because of the low freezing temperature of the refrigerant and the absence of crystallization. For HVAC applications a working fluid of water and lithium-bromide is preferred, which has higher freezing temperature when compared with the previous combination.

The relevant physical phenomena are that when the Li-Br solution evaporates (or boils) heat is absorbed, and vice versa, when it condenses heat is released. Moreover the evaporating temperature of a liquid is a function of the pressure, i.e. as the pressure decreases so does the boiling point. A water and Li-Br mixture is used since some chemicals can absorb others, in this case water is used as refrigerant while Li-Br is used as an absorbent. The transformations that would occur while operating with this fluid are now presented, based on a machine working with an input water temperature of 90 °C:



(a)



(b)

FIGURE 2.11: T-s and p-h diagrams for refrigeration cycles. In (a) Temperature T - Entropy s diagram for refrigeration cycle; (b) Pressure p - Enthalpy h diagram for refrigeration cycle.

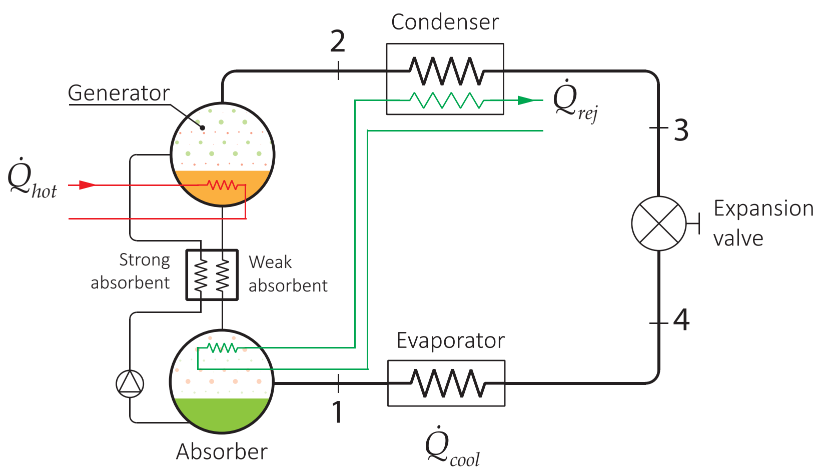


FIGURE 2.12: Absorption chiller scheme.

- The solution water/lithium-bromide is heated in the *generator* (at a pressure of about 66 mmHg - 0.087 atm), causing the water to evaporate.
- The water vapor enters the *condenser*, where it is condensed by cooling water (usually from DCs or CTs) and brought to the saturated liquid state.
- The water passes through an *expansion valve*, where it is expanded to a pressure of 6 mmHg (0.008 atm). Under these conditions, water evaporation occurs at temperatures close to 4 °C.
- Water is sprayed on the *evaporator* tubes in a vacuum condition chamber. In these conditions water evaporates at a low temperature, taking heat from the environment or from the fluid to be chilled. In HVAC applications chilled water enters the absorption chiller at 12 °C and exits at 7 °C.
- The steam, in the dry saturated steam condition, enters the *absorber*, where it is absorbed by the concentrated solution of Li-Br coming from the generator. The concentrated solution of Li-Br obtained from the generator is then returned to the absorber, where, thanks to its

low refrigerant content, it is able to absorb the refrigerant in the vapor phase coming from the evaporator (while maintaining the low pressure) and to recreate the initial concentration conditions, making it possible to cyclically operate the system. The absorption process generates heat which is usually dissipated through the same system used for the condenser, as is shown with the green line in Figure 2.12.

Since the generator and absorber have different temperatures, it is advisable to insert a regenerative exchanger which cools the solution coming from the generator and vice versa.

2.4.3 Types of absorption chiller

Absorption chillers can be categorized based on the heat source, and consequently the heat exchanger used. When the heat source comes directly from the combustion of a fuel, ABS chillers are usually called "direct fired". *Steam fired* and *hot water fired* types use water as heat transfer fluid in vapor or liquid state respectively, this types are a common solution in heat recovery plants.

Other variations of ABS chillers are the *single* and *double* effect types. Double effect chillers are characterized by the adoption of two single effect cycles in cascade, so that the condensation heat of the first single cycle constitutes the energy input to the generator of the second cycle, thus creating a double-acting cycle.

Double effect ABS are usually more expensive than single effect machines, additionally requiring higher heat source temperatures of at least 150 °C.

2.4.4 ABS chillers thermodynamic performance indexes

If the energy required for the solution and refrigerant pumping is excluded, the coefficient of performance of a direct-fired absorption chiller COP_c can be calculated as:

$$COP_c = \frac{\dot{Q}_c}{\dot{Q}_h} \tag{2.10}$$

where $\dot{Q}_h$ is equal to the heat input to high-temperature generator measured in kW, and $\dot{Q}_c$ is the net cooling effect. The coefficient of performance of absorption chillers is usually a function of three variables:

- Exiting chilled fluid temperature, used for cooling applications.
- Entering hot fluid temperature, used as heat source.
- Cooling fluid temperature, used for condensation.

As an illustrative case, an evaluation of the COP of a hot water fired absorption chiller is reported in figure 2.13. It can be noted that the higher is the temperature of the hot water and cooling water, the higher the COP, resulting in increased efficiency of chilled water generation.

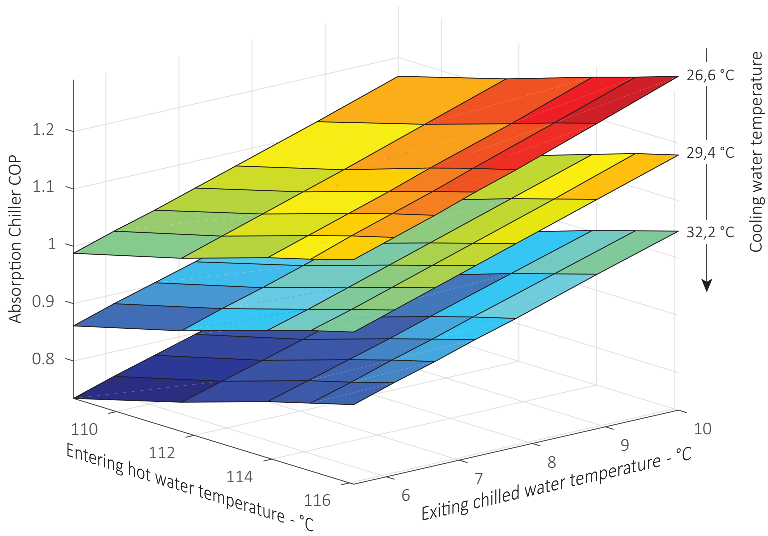


FIGURE 2.13: Absorption chiller COP evaluation based on entering hot water temperature, exiting chilled water temperature, and condenser temperature.

Usually in a single-effect machine, the COP is about 0.6 - 0.75 which rises to around 1.1 - 1.35 in double-effect machines.

2.5 District heating and cooling

District heating is the last technology analyzed used to exploit low grade waste heat. It is shown in Figure 1.1 at the bottom of the “recovery” branch, under absorption cooling. A number of studies have recently been performed to estimate industrial waste heat potentials at regional, international and global level [29, 33, 63, 64], and a growing body of literature has examined recovery technologies for specific processes and industry sectors e.g. [25, 30, 31, 37]. In particular, the use of industrial waste heat in district heating systems is an attractive option [25], and several successful experiences are reported in professional practice [47] and in scientific literature [48, 49]. Opportunities to expand existing networks or to build new ones are thus increasingly offered to companies and public administration as a means to achieve energy efficiency and environmental benefits.

2.5.1 Introduction

District heating (DH) refers to a system for heat distribution generated in a thermal power plant through a system of insulated pipes for HVAC purposes like domestic and commercial heating, and domestic or commercial water heating.

The heat source can derive from combined heat and power plants (CHP) fuelled with different types of fuel, heat-only boiler stations, geothermal heating, or heat recovery heat exchangers located in energy intensive industries. Due to the centralization of heat generation, which allows for economies of scales and better pollution control, DH can provide higher efficiencies and increased pollution control with lower end-user tariffs than domestic boilers. The main components are:

- Heat power plant, like the CHP plant already mentioned, heat-only boiler, geothermal stations, heat recovery boilers etc.
- Pipeline network, used for the heat distribution. Pipes are usually insulated and installed underground.
- Pumping station(s), used to move the fluid used for heat transfer, for long pipe networks more pumping stations are distributed along the network.
- Heat transfer fluid, usually water or in some cases steam.
- District heating substations, which comprise a heat exchanger coupled with the DH network and the end-user heating plant with a heat-meter counting the heat demand.

Figure 2.14 shows a DH system example where network pipes (send and return), an end-user substation, domestic heating plant, and the heating generator as well as some fuel types can be observed.

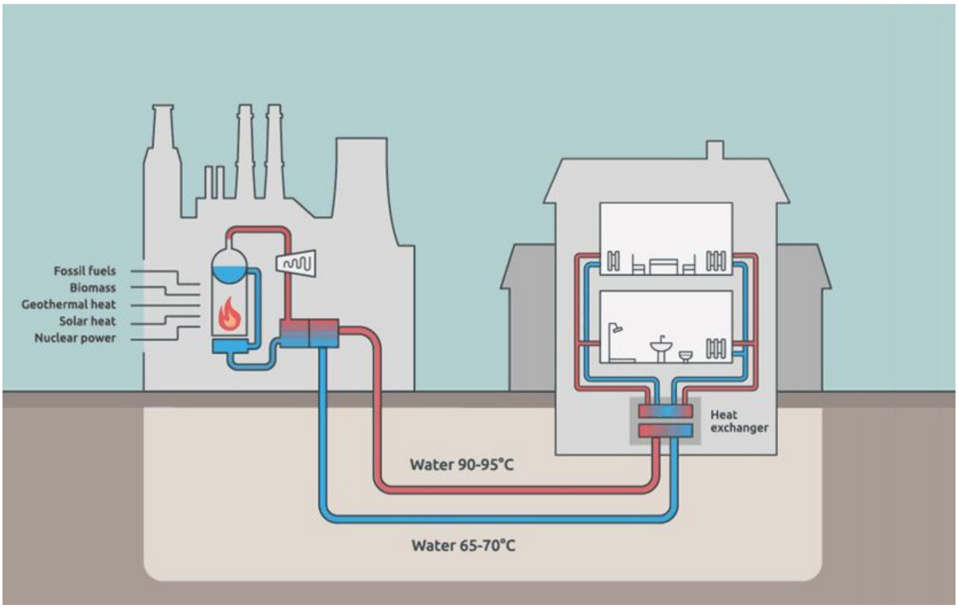


FIGURE 2.14: District heating scheme, courtesy of Laura Toffetti, DensityDesign Research Lab under the Creative Commons Attribution-Share Alike 4.0 International license `https://commons.wikimedia.org/wiki/File:District_heating.gif`.

Similar to DH, district cooling is a system used for chilled water distribution for cooling purposes like air conditioning. Chilled water is produced through trigeneration plants (CHCP - combined heating cooling and power), absorption chillers or MVC chillers. The diffusion is smaller when compared with DH and usually when district cooling network is available so is district heating. From the distribution network point of view the components are the same with special concern for condensation problems along the insulated pipes.

2.5.2 Heat generation

Heat sources in use for various district heating systems besides CHP plants include: nuclear power plants, simple combustion of fossil fuel or biomass, geothermal heat, solar heat, industrial heat pumps which extract heat from seawater, river or lake water, sewage, or waste heat from industrial processes.

The core element is a heat-only boiler station. Additionally a cogeneration plant (CHP) is often added in parallel with boilers. Both have in common that they are typically based on combustion of primary energy carriers. CHP plants coupled with DH offers the production of heat and electricity simultaneously, in this case the heat output is typically sized to meet half of the peak demand, even though the total energy supplied during the year can exceed 90%. Boiler capacity is selected in order to supply the entire heat demand (peak included), this is to allow maintenance or breakdown management.

2.5.3 Heat distribution

The heat transport network consists of two underground insulated pipes (send and return). In the delivery circuit the heat transfer fluid flows at its maximum temperature (usually 90 °C for hot water) until the end-user heat exchanger where enthalpy exchange occur between DH circuit and the end-user heating circuit. Once the heat has been transferred, return water temperature is usually about 70 °C. The technical characteristics of the pipes adopted depend on the installation type, the heat demand supplied and the territory morphology.

The common medium used for distribution is pressurized hot water, in some cases steam is also used. The advantage of steam medium pipelines is related to the final use of heat, in addition to heating purposes steam can be used in industrial processes. Disadvantages of this configuration include higher heat losses along the network.

Usually, DH networks use a derivation connection, where every user is connected directly to the main pipeline (send) and returning the "cold" water in another line (return). In this way the water delivery temperature is only influenced by the thermal losses along the pipe and not by the upstream user. This type of connection is preferable when flexibility is required since the realization is simple and it allows the connection of other

FIGURE 2.15: District heating biomass boiler, thermal power of 800 kW. Here shown during summer period maintenance.

users at a later time. Other types of connection are ring network and mesh network.

2.5.4 Pipeline characteristics

The pipes used for fluid transport in the district heating networks are pre-insulated. The distinctive technical characteristics of the pipes are the:

- construction material
- insulating material
- protective coating
- junction type
- pipe flexibility
- laying system

There are three main materials used for DH pipelines: steel, polyvinyl chloride (PVC) and high-density polyethylene (HDPE).

Steel : Steel pipes are certainly the most widespread, and are suitable for all situations. UNI 1281/67 UNI 1283/67 and UNI 1284/71 standards defines nominal performances and characteristics. Due to the resistance and durability, steel pipes are usually expensive. Moreover, the joint requires a head-to-heat welding. This and the high weight of the pipelines increases the realization costs.

PVC : Polyvinyl-chloride has a lower resistance compared to steel, however it offers other advantages such as a light weight, easy processing, and very high corrosion resistance. PVC pipes are cheaper than steel.

HDPE : High Density Poly-Ethylene is the intermediate solution between steel and PVC since it is more resistant than PVC and more economic than steel.

DH pipes are coated with insulating material, insulation can be directly applied during the installation (field fabricated insulation) or pre-installed by the manufacturer (prefabricated insulation). The materials used for insulation include glass fibre, PVC and HDPE, while the external protective coating is PVC or HDPE.

Depending on the pipe stiffness, DH pipes can be differentiated into flexible or rigid. Rigid pipes are the most widespread, usually supplied in rods standardized from 6 m which are laid and connected to each other through head-to-head welded junctions. Flexible pipes are made of PVC or wrinkled steel usually stored wrapped on a reel.

2.5.5 District heating substation

A DH substation is a connection unit between building's own heating system and the DH network used to exchange heat. The connection can be direct or indirect. The direct method consists of connecting the main DH pipeline directly to the secondary circuit is inside the end-user building. The heat transfer fluid flowing is the same in both primary and secondary circuit. The problems related to this configuration are the pressure difference between the primary and the secondary systems, as well as flow imbalances. The indirect connection interposes a heat exchanger between main and secondary circuit solving the problems related to pressure drops and flow imbalances previously explained. Figure 2.16 shows an example of district heating substation.

A DH substation has normally one or more of the following parts:

- A heat exchanger, used to split primary and secondary circuit, usually a plate heat exchanger.
- Flow control valves, used to regulate the flow through the heat exchanger,
- A shut off valve, used during maintenance or breakdowns to stop the flow on primary side.
- A heat meter, used to measure the heat exchanged for cost charging.
- Secondary side pump circulation, used in the building's own heating system.

FIGURE 2.16: District heating substation. The plate heat exchanger is placed inside the silver box with red exiting rods.

- A temperature control system: composed of thermostats, temperature sensors and valves. Used to control temperature on secondary side by regulating the flow on primary side.
- Filters: used in order to avoid extraneous matter from reaching critical components e.g. controlling valves and the heat exchanger itself.

2.5.6 The design of district heating networks

2.5.6.1 Heat loss evaluation

During the design of a district heating system, other than hydraulic and mechanical verifications, heat loss evaluation and temperature drops should be considered. Heat losses are a function of many factors such as insulation thickness, installation depth, and pipe distances. In Figure 2.17 a district heating send/return pipes are shown with all the distances used for the heat loss evaluation.

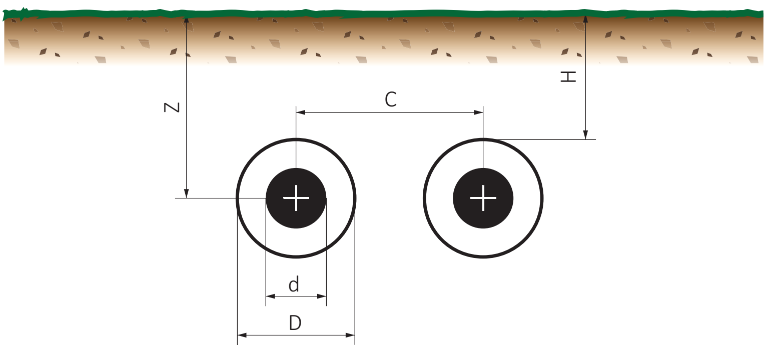


FIGURE 2.17: District heating model for heating loss evaluation.

The following procedure is proposed by Socologstor [65]. Heat losses Φ per unit length of a couple of pipes could be evaluated as follows:

$$\begin{aligned} \Phi &= U(T_m - T_t) + U(T_r - T_t) \\ \Phi &= U\left[(T_m + T_r) - 2T_t\right] \end{aligned} \tag{2.11}$$

where:

- Φ is the heat loss per unit length of a couple of pipes W/m
- U is the thermal transmission coefficient in W/(m °C)
- T_m is the send temperature in °C
- T_t is the soil temperature in °C
- T_r is the return temperature in °C

While temperatures are known, the evaluation of the thermal transmission coefficient requires knowledge of the thermal resistances for the evaluation of U, calculated as:

$$U = \frac{1}{R_{Tts} + R_{Ti} + R_{Ttg} + R_{Tt} + R_{Ts}} \tag{2.12}$$

where:

- R_{Tts} is the thermal resistance of the pipe - m °C/W
- R_{Ti} is the thermal resistance of insulation layer - m °C/W
- R_{Ttg} is the thermal resistance of the coating layer - m °C/W
- R_{Tt} is the thermal resistance of the soil - m °C/W
- R_{Ts} is the "exchanging" thermal resistance between the two pipes - m °C/W

The thermal resistance of cylindrical layer can be evaluated as:

$$R_l = \frac{1}{2\pi\lambda_l} \ln\left[\frac{d_e}{d_i}\right] \tag{2.13}$$

where:

- λ_l is the thermal conductivity layer analyzed - W/(m °C)
- d_i is the internal diameter of the cylinder - m
- d_e is the external diameter of the cylinder - m

Equation 2.13 can be easily applied to calculate R_{Tts}, R_{Ti}, and R_{Ttg}.

For the thermal resistance of the soil the following equation should be used:

$$R_{Tt} = \frac{1}{2\pi\lambda_t} \ln\left[\frac{4Z_c}{D}\right] \tag{2.14}$$

where:

- λ_t is the thermal conductivity of the soil - W/(m °C)
- D is the external diameter of coating pipe - m
- Z is the installation depth evaluated at the internal axis - m
- R_0 is the surface transition resistance equal to 0.0685 m^2/(°C W)
- Z_c is the virtual depth of installation evaluated as (in m)

$$Z_c = Z + R_0 \cdot \lambda_t \tag{2.15}$$

The thermal resistance between send and return pipes:

$$R_{Ts} = \frac{1}{4\pi\lambda_t} \ln\left[1 + \left(\frac{2Z_c}{D}\right)^2\right] \tag{2.16}$$

where:

- λ_t is the thermal conductivity of the soil W/(m °C)
- C axis distance m, evaluated as $Z_c = Z + R_0 \cdot \lambda_t$
- Z is the installation depth evaluated at the internal axis m
- R_0 is the surface transition resistance, equal to 0.0685 m^2/(°C W)
- Z_c is the virtual depth of installation in m evaluated as the previous equation 2.15

The thermal conductivity of steel, polyurethane (PU), polyethylene (PE) and the soil are shown in Table 2.4. For instance, we can consider the real case of a DH pipe with nominal diameter of 300 mm, that has an internal diameter d of 323.9 mm, an external diameter D of 450 mm, a steel thickness s_{steel} of 5.6 mm, and a polyurethane thickness s_{PU} of 7 mm. Then assuming a distance between pipes of 150 mm and an installation depth of 0.5 m the transmission coefficient is $U = 0.522\ \mathrm{W/(m\,°C)}$.

TABLE 2.4: Material conductivity for some elements [65].

Material	Conductivity W/(m °C)
Steel	55
PU	0.03
PE	0.35
Soil	1.5

In order to understand the heat loss per unit length of a pipe we assume the temperature of the soil T_t is 10 °C and the temperature of the water being sent is 80 °C. Finally the heat loss per unit length can be evaluated as:

$$
\begin{aligned}
\Phi &= U \cdot \left(T_{fc} - T_t\right) \\
\Phi &= 0,522 \cdot (80 - 10) = 36.54\, W/m
\end{aligned}
\tag{2.17}
$$

2.5.6.2 Pressure drop evaluation

Pressure drops can be divided in two types: localized and distributed.

- *Localized* pressure drops are related to section reduction, strong curves and valves or whatever generates a local perturbation on the flux and consequently causes a local pressure drop. Often due to difficulties in finding all the localized discontinuities, they are taken into account as a percentage of the distributed losses.
- *Distributed* pressure drops are related to flux into the pipes and in particular to viscous friction originating at the internal wall of the pipe.

The evaluation of distributed losses can be assessed with the following equation:

$$\Delta p = \frac{1}{2}\rho v^2 f \frac{L}{D} \tag{2.18}$$

where:

- ρ is the fluid density, kg/m^3
- v is the fluid velocity, m/s
- f is the friction factor, dimensionless
- L is the length of the pipe m
- D is the pipe diameter m

The friction factor can be evaluated with several analytical or empiric methods such as: Colebrook, Altshul, Blasius or Nicuradse. The first two methods are analytical and therefore require the roughness ε as a known value. Roughness is a function of the material of the pipe. Since not all pipes are made with the same roughness, an empirical approach is used in this thesis, thus we consider the following relations:

- Blasius' formula: used only if Reynolds' number is between 10^4 and 10^5

$$f = \frac{0.0316}{\sqrt[4]{Re}} \tag{2.19}$$

- Nicuradse's Formula: used if Reynolds' is higher than 10^5

$$f = 0.00032 + \frac{0.221}{Re^{0.237}} \tag{2.20}$$

2.5.6.3 Pumping systems

Since fluid is used as a heat transfer medium for district heating systems a pumping station is required during operation. A pressure of 1.6-2.5 MPa is required during operation. The design of pumping station is related to the heat losses factors mentioned before: the pressure required and the length of the system. For district heating networks, particularly those that are long or with strong pressure drops (e.g. skyscrapers, hills or etc.), it is possible to add more pumping stations. The choice between the various plant solutions is a function of techno-economic cost analysis. The electrical power required by the pump can be evaluated as:

$$P = \frac{\Delta p \cdot \dot{V}}{\eta_p} \tag{2.21}$$

where:

- Δp: head loss (Pa)
- $\dot{V}$: volumetric flow rate (m^3/s)
- η_p: electric efficiency of the pump

2.6 Conventional systems for energy conversion

To allow a fair economic and environmental evaluation of the Water-Energy nexus indicators a comparison with conventional energy conversion technologies is also needed. As illustrated in Figure 1.1 faced with the low-grade waste heat technologies, the conventional technologies are shown. In particular ORC will be compared with the national energy mix; absorption cooling will be compared with mechanical vapor compression chiller; and district heating will be compared with heating produced with a boiler fuelled with natural gas.

The conventional systems, used as a reference technology are now introduced: the national energy mix is presented in section 2.6.1, mechanical vapor compression refrigeration in section 2.6.2, and boilers in section 2.6.3.

2.6.1 The national energy mix

The national energy mix (NEM) in this work describes the production of electrical power in a nation of concern. More precisely, the focus is on the primary sources used and the condensation devices required, based on the technologies used for power generation.

The national energy mix is used as a reference system when a comparison with ORC is required.

2.6.1.1 Energy Sources

Energy sources can usually be divided into non-renewable and renewable resources. Non-renewable means that the resource that does not renew itself at a sufficient rate for sustainable economic extraction in meaningful human time-frames. Examples of non-renewable resources are fossil fuels (coal, petroleum, natural gas) and nuclear. In contrast, examples of renewable resources are:

- wind power
- hydropower
- solar energy
- geothermal energy
- bio energy (biomass, biogas, and biofuel)

Non-renewable energy sources are exploited in steam turbine generators or in combustion turbines when natural gas is used. Steam turbines are also used for power generation using nuclear and geothermal sources. Combined heat and power (CHP) plants usually exploit fuels like natural gas, biomass, and biogas. Figure 2.18 shows the thermal efficiency of different power plant types, as well as the maximum single unit output.

Based on the plant type and the geographical position, different cooling systems are used for condensation or cooling purposes (e.g. CHP engine cooling). Water consumption based on the energy source will be discussed in sections 2.6.1.2 and 2.6.1.3. Table 2.5 provide the CO_2 emitted and the primary energy consumption based on the source used.

2.6.1.2 Cooling systems of power plants

Natural water sources (typically the sea) are used for cooling purposes in power generation. Cooling is performed through once-through cooling systems [68]. These systems, which take water from the sea, circulate it through the plant's heat exchangers and return it to the local source, are characterized by high water withdrawals, but relatively low water consumption.

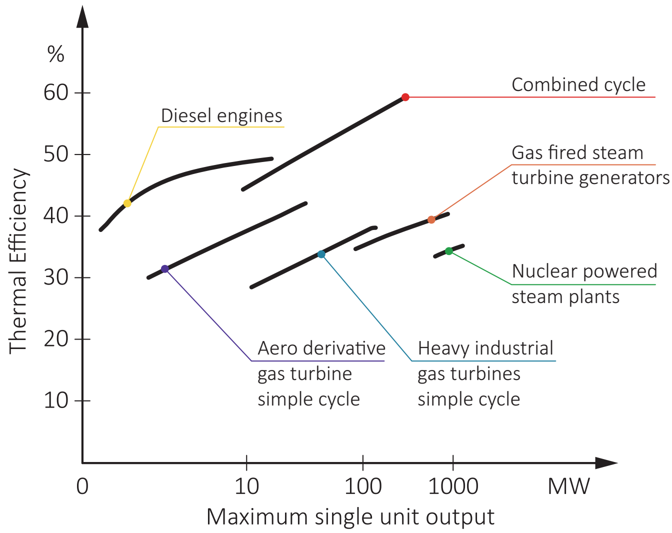


FIGURE 2.18: Comparison of efficiency and power output of various power plant types, extrapolated from Refs. [66, 67].

TABLE 2.5: Carbon emissions and primary energy consumption based on power plant source [58].

Source	CO_2	PE
	$tCO_2/GW{\cdot}h_e$	$TOE/kW{\cdot}h_e$
Biomass & Waste	35	2.03E-04
Solid Fuels	1001	2.68E-04
Natural Gas	481	2.00E-04
Geothermal Energy	45	3.65E-04
Hydropower	18	5.20E-05
Nuclear Energy	23	2.82E-04
Crude Oil	742	2.65E-04
Solar Energy	50	6.45E-05
Wind Energy	17	4.43E-05

Closed-loop or wet recirculating cooling systems reuse cooling water in a second cycle rather than immediately returning it back to the original water source. Examples are cooling towers and dry coolers.

Based on the fuel/source used for power generation CO_2 emissions and water consumption as well as primary energy used can vary substantially. Table 2.6 shows a summary of the water consumption depending on the power plant source. It can be noted that hydropower is the most intensive in terms of water consumption, a brief explanation is given in the next section.

TABLE 2.6: Water consumption based on power plant source [58].

Source	Water
	$l/MW{\cdot}h_e$
Biomass & Waste	1499
Solid Fuels	1968
Natural Gas	1044
Geothermal Energy	7429
Hydropower	36766
Nuclear Energy	2289
Crude Oil	1334
Solar Energy	2354
Wind Energy	2

2.6.1.3 Water consumption in hydropower

Storage of water behind large hydropower dams leads to consumptive water use through evaporation from the open water surface of the artificial lake. Many studies can be found in literature evaluating the evaporated water per GJ produced in hydroelectric facilities for instance, it was shown that, on average 1.5 m^3 of water per GJ of electricity produced is evaporated from hydroelectric facilities in California [69].

Gerbens-Leenes et al. [70] estimated the global average blue water footprint of electricity from hydropower is 22 m^3/GJ By combining the estimate of global evaporation from artificial water reservoirs in the world from [71] with data on global hydroelectric generation from [69].

Hydropower in energy systems is associated with low life cycle carbon equivalent emission factors and primary energy consumption, but has the highest freshwater consumption footprints mainly due to evaporation from hydropower reservoirs [15].

2.6.2 Vapor-compression refrigeration

Vapor-compression refrigeration is widely used for HVAC applications, domestic and commercial refrigerators, food storehouse for chilled or frozen storage of foods and meats, air conditioning of automobiles, trucks and railroad cars. Mechanical vapor compression systems are also used for refrigeration in oil refineries, petrochemical and chemical processing plants, and natural gas processing.

2.6.2.1 Types of refrigeration compressors

The key element in mechanical vapor compression chillers is the compressor. On the market a variety of types are present, depending on size, noise, efficiency and pressure, which are selected based on the application and the construction. Common compressors can be categorized based on the compression system, which can be reciprocating, screw, centrifugal, and scroll.

Another way to categorize compressors is based on the arrangement between the compressor and the electric motor. Compressors are often described as being either open, hermetic, or semi-hermetic. Different arrangements can lead to the following configurations:

- *Hermetic*: the motor and the compressor are "sealed" or "welded" in the same housing. This type of compressor has reduced leakage of refrigerant, and is cheap and small sized.
- *Semi-hermetic*: allows access to the motor for maintenance or failures. This type is common for medium cooling capacity applications.

- *Open motor* (belt driven or direct drive): where motor and compressor are separate, this configuration is used for high cooling capacity chillers. The problems are related to refrigerant leakage.

A chiller designed for high cooling loads and performances is the so called "magnetic" centrifugal chiller where enhanced efficiency is achieved through the application of active magnetic bearings with a variable speed drive.

The energy required during the compression is given by an electric motor. To understand the energy consumption for different chillers the thermodynamic indexes are now introduced.

2.6.2.2 MVC chillers thermodynamic performance indexes

The coefficient of performance (COP) is a performance index used in thermodynamic cycles, in particular for heat pumps, refrigerators or air conditioning systems. Because the COP can be greater than 1, COP is used instead of thermal efficiency. The COP can be used for the analysis of the following thermal machines:

- Refrigerators used to produce a refrigeration effect only, whereby COP_f is used as abbreviation.
- Heat pumps in which the heating effect is produced by rejected heat, whereby COP_{hp} is used as abbreviation.

For a refrigerator, COP is defined as the ratio of the refrigeration effect $\dot{Q}_1$ to the work input $\dot{W}_{in}$, that is,

$$COP_f = \frac{\dot{Q}_1}{\dot{W}_{in}} \tag{2.22}$$

for the Carnot refrigeration cycle:

$$COP_f = \frac{\dot{Q}_1}{\dot{Q}_2 - \dot{Q}_1} = \frac{h_1 - h_4}{h_2 - h_1} \tag{2.23}$$

where h_1, h_2, and h_4 can be found in Figure 2.11(b).

The COP evaluated in 2.23 is commonly called the energy efficiency ratio (EER) and is used as an indicator of a chiller's performance. To evaluate the system behavior and energy consumption over an entire season in order to understand the electrical energy input required, EER can be evaluated over an entire season. This new index is called SEER. As an example, the EER of the magnetic centrifugal chiller discussed above is 6.4, while SEER is higher than 8.32.

The evaluation of the other types of COPs, such as COP_{hp}, is not of interest in this thesis.

2.6.3 Boiler

A boiler is used in heating systems to transfer thermal energy from combustion products to a heat transfer fluid which, through the system itself, is used to heat the end users e.g. domestic radiators, fan coils, district heating substations and all the designated devices. The heat transfer means is usually water, and if the element is designed to change the state of water from liquid to steam, the element is typically called "steam generator", which is used in thermoelectric plants for power generation through turbines.

Boilers are used in various heating or process applications, including domestic, commercial, and industrial hot water generation. Boilers use several fuels such as wood, coal, oil, or natural gas. Electric boilers use resistance or immersion type heating elements.

In this work, due to the relevance for the cases analyzed, boilers fuelled with natural gas (NG) or heating oil (HO) are considered.

2.6.3.1 Technical characteristics

The essential components of a gas boilers are a: burner, combustion chamber, internal heat exchanger, and a chimney.

A burner is a mechanical device that supplies required amount of fuel and air and creates a condition of rapid mixing to produce a flame.

- Burners vary based on the fuel used. Gas burners consists of an air shutter, a gas orifice, and outlet ports and are usually made of aluminum painted, heavy-gauge steel or aluminized steel, or sometimes stainless steel. When burners are placed vertically they require only minimal drafts of air and are suitable for vertical fire-tube boilers. Some boilers have a burner coupled with a fan to supply and control combustion air in the combustion chamber. This configuration is often employed for industrial boilers [50].

FIGURE 2.19: Industrial boilers, courtesy of Zantingh B.V., under the Creative Commons Attribution-Share Alike 4.0 International license `https://commons.wikimedia.org/wiki/File:Zantingh_Brander.JPG`.

- A combustion chamber is the place where the combustion and the heat transfer from combustion products to heat transfer fluid occur. When biomass is used as fuel, the ashes derived from combustion are accumulated before being extracted. The combustion chamber is usually slightly depressed in the case of solid fuel, in the presence of a burner the chamber is sometimes at a pressure higher than atmospheric.
- A heat exchanger is the element responsible of transmitting the heat produced by combustion to the water in the heating system. It is usually placed inside the combustion chamber where the heat transfer occurs by conduction, convection and thermal radiation.
- The chimney is the duct used for the expulsion of exhaust gases coming from the combustion chamber.

2.6.3.2 Boiler efficiency

The performance of a gas-fired boiler is usually indicated by the thermal efficiency η_b. One method to measure the thermal efficiency is expressed by Hassenstein's formula:

$$\eta_b = 100 - K \cdot \frac{T_f - T_a}{CO_2} \tag{2.24}$$

where:

η_b is the combustion efficiency (%)

K is a coefficient based on the type of fuel used:

- 0.60 liquid fuels
- 0.45 gas fuels
- 0.75 coal
- 0.85 biomass

T_f flue gases temperature, (exiting flue gases temperature form the boiler)

T_a heating cellar air temperature

CO_2 carbon dioxide percentage on the flue gases

The coefficient K depends on the percentage of CO_2 in the flue gases and on the type of the fuel used. In this work η_b is used to assess the fuel consumption required by a certain heat load, and is usually taken from manufacturer's catalogue.

2.7 Concluding section

The elements described in this chapter could be used for heat recovery purposes, increasing energy efficiency and reducing CO_2. In particular, while ORC and district heating are well established configurations for heat recovery, there is a less known about absorption chilling and therefore it is analyzed in detail to better understand thresholds for its use.

An introduction to condensers has been given because of their strong relevance for this thesis. Cooling towers and dry coolers are common devices used in refrigeration cycles and power cycles, as well as to cool down process fluids. Moreover CTs and DCs are used in industrial heat recovery configurations particularly when plants are relatively small systems located close to end markets, usually in inland areas and often use closed-loop cooling systems.

It can be observed that:

- CTs performance is related to the wet bulb temperature of the ambient air. CTs consume significant amounts of water, while electricity is required for ventilation fans (smaller than DCs). The cooled fluid exiting the CTs usually has a lower temperature when compared with a fluid cooled using DCs.

- DC performance depends both from the dry bulb temperature of the air and from the rotational speed of the fan, which is usually more energy intensive when compared with CTs. Therefore, DCs require more electricity than CTs during operation and the exiting fluid is usually warmer. On the other hand, the evaporation of water does not occur during the operation.

- Absorption chillers (ABS) do not require electricity during operation, except for small recirculating pumps and controls. ABS chillers require heat to work and the energy efficiency is affected by three parameters: the temperature of the heat source, the cooling water temperature and the chilled water temperature. In particular, for fixed chilled water conditions, higher hot source temperatures and lower cooling temperatures lead to higher COPs.

- MVC chillers are devices commonly used in HVAC applications, and during the operation electricity is required by the compressor. COP_f is only influenced by the temperature of the cooling water, in particular, the colder it is the better the COP_f is.

- District heating can be used as a configuration for heat recovery. The energy consumption is related to energy losses along the pipeline and pumping. It can be coupled with cogeneration plants (combined heat and power - CHP) and trigeneration plants (combined cooling, heat, and power CCHP), increasing the energy efficiency of entire urban areas. A boiler is the configuration to compare this with.

- ORC can be used for power generation from waste heat, and is the only commercially viable technology to generate power from low temperature waste heat flows at industrial sites.

- Depending on the climate conditions, a suitable choice between either CTs or DCs determines better performances of the cycles. However there is a substantial difference in terms of resource consumption.

Chapter 3

Methodology

In this chapter the methodological approach of this work is presented. A flow-chart outlining the methodology is used as reference, then further details are presented, such as configuration identification, system boundaries, and data uncertainty. A research analysis of the most important methodologies for assessing environmental impact of the systems is also conducted.

3.1 Introduction

A consistent methodology is important in order to to organize the work and keep the focus on the best way to answer to the research questions. In general, the term "methodology" refers to an outline of the way in which research should be done.

The pathway used in this thesis can be described as follows: defining the research questions, then defining the methodology and study structure in order to understand the methods and rules that should be applied. After the methodology definition, and in accordance with it, the selection of case studies is performed. Subsequent to the application of the case studies results a discussion is had, which leads to conclusions addressing the research questions. The pathway used in this work is summarized in Figure 3.1 where all these steps are presented.

The aim of this work is to understand the environmental impact of different heat recovery opportunities, as well as their economic feasibility. Hence as shown in the first box of Figure 3.2, which can be used as a

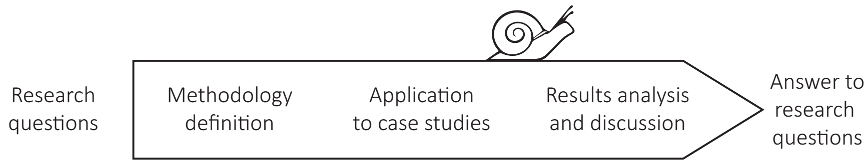


FIGURE 3.1: The research strategy used in this work.

guide to the elements described in the methodology definition, the water-energy nexus concept and the environmental indicators should first be introduced. A benefit and disadvantage estimation in relation to the configurations analyzed is fundamental for understanding the implications of various solutions. The environmental indicators are introduced and discussed in Section 3.2.

A review of the waste heat recovery technologies and configurations is performed in order to understand the best practicable solutions for heat recovery at energy intensive industrial sites. Heat recovery opportunities are discussed in Section 3.4 where the analyzed configurations are proposed.

As can be seen in Figure 3.2, after the introduction of the nexus indicators and practicable technologies, the system boundaries are defined. System boundaries are the key element in footprint evaluation; in fact, different system boundaries can lead to different results, even if the system elements under consideration are the same. One can consider the difference between a "cradle to grave" and a "cradle to gate" analysis: in the former, indicators are calculated from resource extraction ("cradle"), through the use phase and to the disposal phase ("grave"), and in the latter, indicators are calculated only on a partial product life, up until the "gate", when the final product exits the manufacturing company. System boundaries are discussed in Section 3.5.

Some of the parameters that industrial waste heat recovery plant performances can be affected by include waste heat temperature, heat availability, wet and dry bulb temperatures of the air, the heat sink temperature and so on. All these parameters constitute the input of the system, and depending on the depth of the analysis, parameters can be treated as fixed or variable values. *Stationary* and *dynamic* simulations of the configurations are discussed in Section 3.6.

Data uncertainty related the nexus indicators as well as the economic

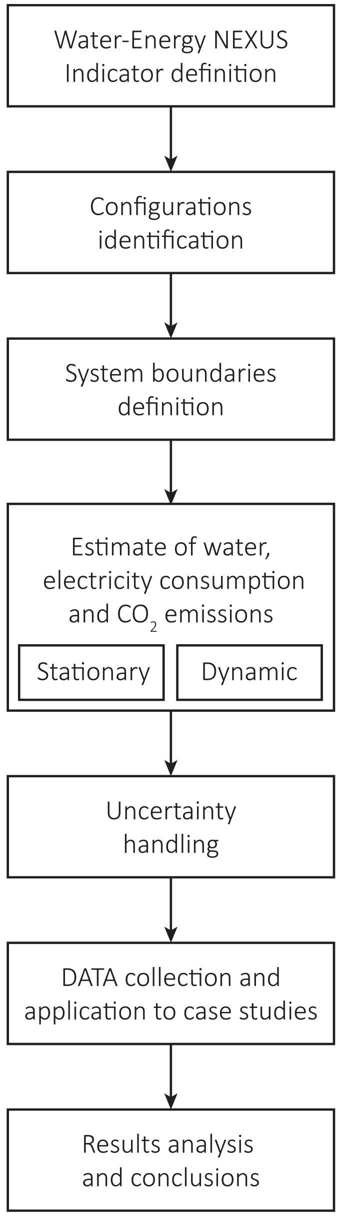


FIGURE 3.2: Methodology flow-chart.

parameters are presented in Section 3.7 where it is shown how uncertainty is handled.

Finally, an application to the case studies and an analysis of results are performed in order to assess the possibility of any numerical or conceptual errors.

3.2 Environmental indicators

The environmental indicators evaluated in this work are: carbon footprint, water footprint and primary energy consumption. The system boundaries chosen lead to the concept of *direct* and *indirect* consumption.

In reference [72], within the frame of an organizational life cycle assessment (O-LCA) and according to the UNEP[1] [73], the definitions of direct and indirect activities are based on the reporting organization. Specifically, direct activities are "activities from sites that are owned or controlled by the reporting organization [73]", while indirect activities are "activities that are a consequence of the operations of the reporting organization, but occur at sites owned or controlled by another organization (upstream or downstream) [73]".

Adapting the previous definitions to the framework proposed in this thesis, we consider *direct* emission/consumption all the emissions derived by the operation of a component owned or controlled by the configuration analyzed (e.g. CO_2 emitted by fuel combustion from an on-site boiler for heating purposes).

For *indirect* emissions (or consumption) we refer to the emissions that are a consequence of the operations of the configuration analyzed, but occur at sites owned or controlled by another system (upstream or downstream) (e.g. indirect CO_2 emissions by electricity use, indirect emissions of plant components manufacturing etc.).

3.2.1 Carbon dioxide footprint

Carbon footprint has been defined as "the quantity of GHGs expressed in terms of CO_2 equivalent mass emitted into the atmosphere by an individual, organization, process, product or event from within a specified

[1] United Nations Environment Programme.

boundary" [74]. As observed by [74], in spite of numerous standards there is a lack of uniformity in the use of direct and embodied emissions in the literature, and defining system boundaries is thus a fundamental step in carbon footprint evaluation.

The evaluation of direct and indirect CO_2 emissions based on the operation or the manufacturing plant's components is based on data available in the literature as well as the GEMIS database [75].

Particular attention must be paid to the national energy mix CO_2 footprint, where evaluation of the footprint can vary substantially based on the source and the technology used for power conversion.

Since every nation has its own energy mix, a method to evaluate the overall footprint of a nation of concern consists of a weighted sum taking into account the national energy mix sources and the relative footprint:

$$\Psi_{CO_2} = \sum_{i=1}^{n} \gamma_i \cdot \varphi_i \tag{3.1}$$

where Ψ_{CO_2} is the CO_2 footprint (tCO_2 per MW·h) of electricity based on the nation of concern; γ_i (%) is the widespread of the $i - th$ technology used for electricity production, and φ_i is the footprint of the $i - th$ technology used (tCO_2 per MW·h).

The CO_2 emissions related to components manufacturing are calculated based on the materials used. Emissions per kilogram of material are taken from the literature and GEMIS [75], then multiplied by the net quantity expressed in units of mass used for an element. Materials quantities are taken from manufacturer catalogues.

3.2.2 Water footprint

Water footprint is generally defined as the measure of "the volume of fresh water used to produce a product over the full supply chain, showing water consumption by source and polluted volumes by the type of pollution" [15]. In the configurations identified in this work however, water footprint is in relation to a process rather than to a product.

The present evaluation is also limited to blue water footprint, which measures the consumptive use of surface and ground water, rather than also encompassing grey water, i.e. measuring water pollution.

The evaluation of the water footprint is similar, in terms of the method and databases used, to the evaluation of CO_2 footprints. In this case, in reference to equation 3.1, φ_i is the footprint of the $i - th$ technology used (m^3 per MW·h) in the national energy mix, while γ_i remains unchanged. Monte Carlo methods are also used in evaluations of these indicators, as well as the water consumption based on manufacturing of components.

3.2.3 Primary energy footprint

Primary energy (PE) is defined as the energy measured at the natural resource level, specifically, PE is the energy used to produce the end-use energy, including extraction, transformation and distribution losses [76].

Primary energy can also be considered as the energy source found in nature that has not been subjected to any human engineered conversion process e.g. energy contained in raw fuels before the energy conversion process resulting in the production of electrical energy, refined fuels, or synthetic fuels. Primary energy can be categorized as non-renewable or renewable.

In this case datasets are taken from literature, [58] and [77], and using GEMIS [75]. Following from the definition of PE, the distinction between direct and indirect is no longer necessary. The evaluation of PE based on a nation of concern is similar as presented in equation 3.1, where φ_i is the PE of the $i - th$ conversion technology used (PE per kW·h_e) in the national energy mix, while γ_i remains unchanged.

3.3 Economic indicators

The definition of the economic indicators used are necessary in order to understand the economic sustainability of a configuration. Importantly, the main assumption of this work is that the rationales of companies and investors are purely economic, and it is within this framework that the economic feasibility of various configurations are studied hand-in-hand with environmental footprint analyses.

3.3.1 Life cycle cost

The life cycle cost analysis (LCC) is an economic evaluation method used to understand the economic feasibility of a project. In an LCC analysis the costs deriving from the installation, use, maintenance and final disposal are taken into account, allowing the determination of the global cost of a product over its entire life. This method is widely used in plant retrofitting economic evaluations. Cash flows over the useful life of the plant are considered to be operating costs, and it is reasonable to assume that the plant realization costs are all paid for at first moment of the plant operation. In particular, if we ignore the growth in operating costs related to obsolescence and inflation, we can assume that the share of operating costs is constant over the years. With these assumptions the formula used to calculate the LCC can be expressed as:

$$LCC = C_{op} \cdot \left(\frac{q^n - 1}{q^n \cdot i} \right) + C_{inv} \tag{3.2}$$

where:

- C_{es} is the operating cost.
- C_{inv} is the plant investment cost.
- i is the discount rate.
- n is the useful life of the plant.
- $q = 1 + i$ is the *anticipation* factor defined as:

$$\frac{1}{q^n} = \frac{1}{(1+i)^n} \tag{3.3}$$

3.3.2 Payback period

The payback period (PB) is the other economic indicator used in this work. It is used to evaluate the economic convenience of an investment. PB refers to the period of time required to recoup the funds expended. The PB corresponds to the year in which the cash flow transitions from a negative value to a positive value. The PB period can be divided in two types: simple and discounted. Simple PB is based only on cash flows, without considering discount rate and may be treated as a particular case of a discounted PB which is evaluated as:

$$PB = \min k | \sum_{k=0}^{n} \frac{\Phi_k}{(1+i)^k} \geq 0 \tag{3.4}$$

where:

k is the year

Φ_k is the net cash flow i.e. cash inflow – cash outflow in the $k - th$ year. It can be expressed as:

$$\Phi_k = (R_k - C_k) - E_k \tag{3.5}$$

where R_k is the annual revenue, C_k is the annual operating cost, and E_k is the initial disbursement, considered paid for at the initial moment (usually $E_k \neq 0$ when $k = 0$, otherwise $E_k = 0$).

i is the discount rate, i.e. the return that could be earned per unit of time on an investment with similar risk.

It should be noted that a simple PB can be calculated by setting $i = 0$, resulting in the equation 3.6, here shown only for the sake of completeness, since a discounted PB is preferable to a simple PB.

$$PB_s = \min k | \sum_{k=0}^{n} \Phi_k \geq 0 \tag{3.6}$$

The payback period is usually expressed in years and intuitively, shorter payback periods are preferable. PB is widely used in many investment engineering areas such as retrofitting, maintenance, upgrades, and in more general energy efficiency interventions.

In the economic assessment of an investment related to energy efficiency a problem arises in the evaluation of profits, since this investment type produces savings rather than profits. In this case, or in any case where no profits are produced e.g. machinery replacement after a breakdown where one has to decide between two options, it is useful to use the following definition of PB:

$$PB = \frac{C_{pl} - C_{pl,ref}}{C_{op,ref} - C_{op}} \quad (3.7)$$

where:

- C_{pl} is the plant cost of the new solution.
- $C_{pl,ref}$ is the reference cost (usually the cost of the comparable option).
- $C_{op,ref}$ is the reference operating cost.
- C_{op} is the operating cost of the new configuration.

The PB evaluation method described here does not taken into account the discount rate.

3.4 Configurations identification

At this point, low grade heat recovery configurations are identified and compared with the standard technologies used for the same purposes. The energy demands taken into account are: electricity, cooling and heating.

Electricity is assumed to be supplied by the national electricity mix, which uses different energy sources such as coal, natural gas, hydro power, nuclear fission, wind, or other renewable sources. Each has a different carbon and water footprint, as well as primary energy consumption.

The relevant technology used for electricity generation in low grade waste heat recovery is the organic Rankine cycle, a common solution for electricity production from waste heat, described in [31, 38, 44, 45] .

In this work the cooling energy demand can be supplied either with the conventional configuration using mechanical vapor compression (MVC) machines or using an absorption machine (ABS), which requires heat instead of electricity to drive the cooling process, and therefore the HR configuration is assumed to be an ABS. This opportunity is proposed by [30] and studied in this work.

Heating demand is assumed to be supplied using on-site boilers, fuelled with renewable and non-renewable fuels e.g. biomass, biogas, natural gas or heating oil. The alternative to the direct consumption of fuels is the recovery of waste heat coupled with district heating networks, which is the configuration used as a comparison. Successful experiences are reported in [47] and in the scientific literature [48, 49]. All the configurations presented here are summarized in Table 3.1, categorized by the energy type supplied.

TABLE 3.1: Standard Configurations (STD) *VS* Heat Recovery (HR) configuration analyzed for energy demand.

Demand	STD config.	HR config.
Electricity	National Energy Mix	Organic Rankine Cycle
Cooling	Mechanical Vapor Compression Chiller	Absorption Chiller
Heating	Boiler/Furnace	District Heating

A summary of the relationships between the nexus indicators and the configurations analyzed is presented in Figure 3.3.

The nexus indicators are located on the left side as colored squares and lines. CO_2, water, primary energy, low (grade) waste heat (LWH) and fuels are shown in the five blocks. Moving from left to right, the first conversion block encountered is the national energy mix (NEM), which requires water, primary energy, fuels and CO_2 (emitted to be precise, here shown as "input" required for power generation) for electricity production. Lines ending in a dot are used to signify consumption, while the triangles indicate outputs, as shown for example by the national energy mix box, which is connected to the yellow line with a triangle, indicating that electricity is produced as an output.

Moving further to the right of Figure 3.3 other conversion boxes are shown, which also correspond to the configurations selected for this study. The interdependencies of selected configurations can be seen, as previously shown for the national energy mix. All the heat recovery configurations are connected to the low (grade) waste heat source indicated in red, while standard configurations use other sources i.e. fuels and electricity.

Condensation is required for the ORC, MVC and ABS chiller. Waste heat rejection at the condenser is indicated by the orange line, where heat rejection if performed through two different devices: cooling towers and dry coolers. Based on the heat rejection device used, different sources are

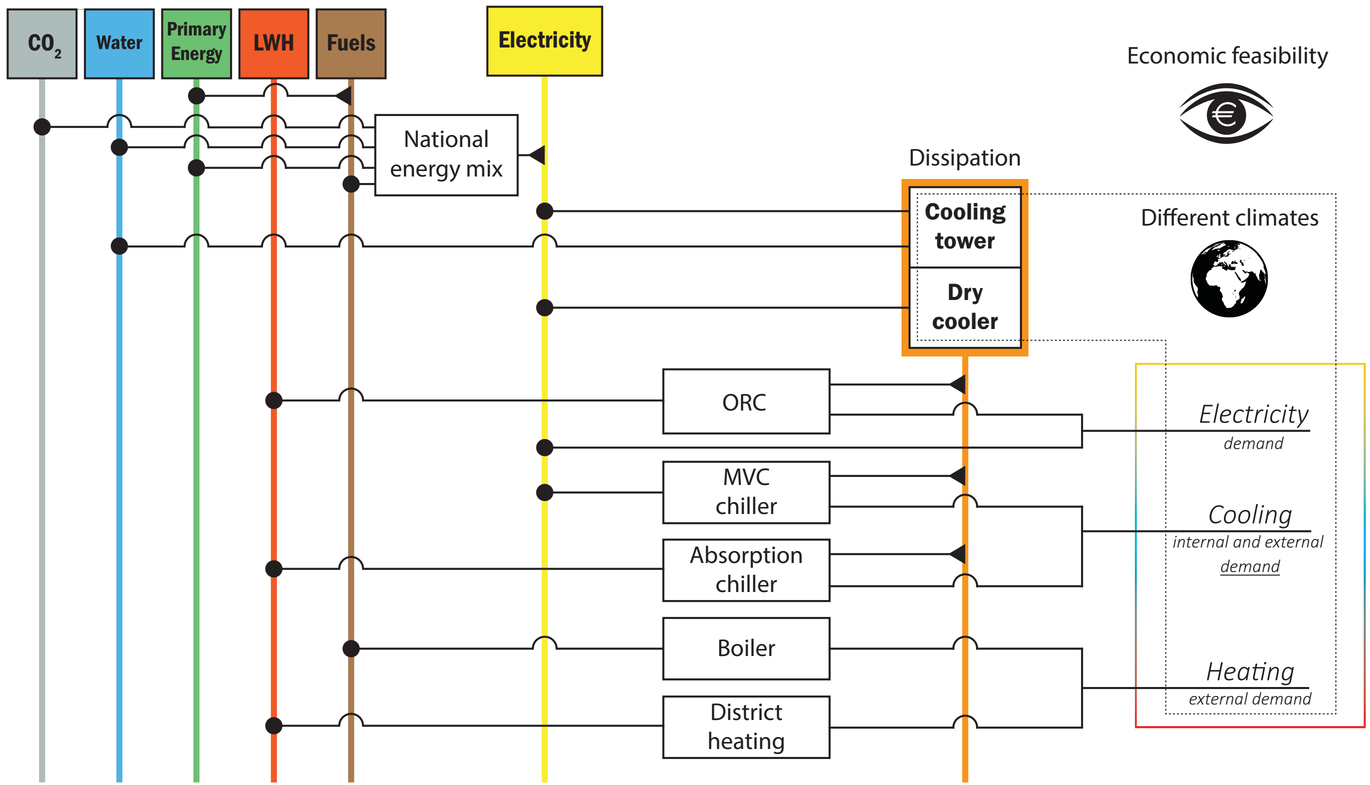


FIGURE 3.3: Footprint methodology scheme.

used e.g. dry coolers require only electricity as *direct* consumption while Cooling Towers require *direct* consumption of electricity and water.

This figure allows to a better understanding of the *indirect* use of sources related to the configurations analyzed. For instance, if dry cooling is used at the condensers, the *direct* water consumption on-site is zero, but since electricity is required during the operation the dry coolers impact on water consumption is *indirectly* based on the national energy mix studied.

The last part of the picture shows the connection between energy demand and the supply chain. Here another variable, shown as a globe, is introduced in order to answer the fourth research question, to understand the performances of LGWHR options depending on local conditions such as market prices, local energy mix and climate.

Dissipation and energy demand can be different in the different regions (and climates) in which the assessment of nexus indicators are performed. The assessment of climate influence on energy demand will be discussed in Section 3.6.

The last indicator analyzed is economic feasibility, indicated by a Euro symbol superimposed onto an eye, meaning that all the configurations presented will also be assessed in terms of economic profitability.

3.5 System boundaries definition

A system boundaries definition is necessary during the footprint calculation. Different boundaries can produce different results with the same case analysis. The boundaries used in this study are shown in Figure 3.4.

The rationale behind the boundaries definition focuses on the evaluation of LGWHR configurations as retrofitting or refurbishment options. This means that no new standard plants are built in order to provide the same energy produced by the LGWHR configurations.

For instance, we can consider the comparison between ORC and the national energy mix. In this case footprint evaluation takes into account the manufacturing processes related to all the ORC plant construction and other devices required for its running (condensers etc.), as well as the operation. The footprint calculations related to the standard configuration are related only to its operation, i.e. the construction of new STD power plants

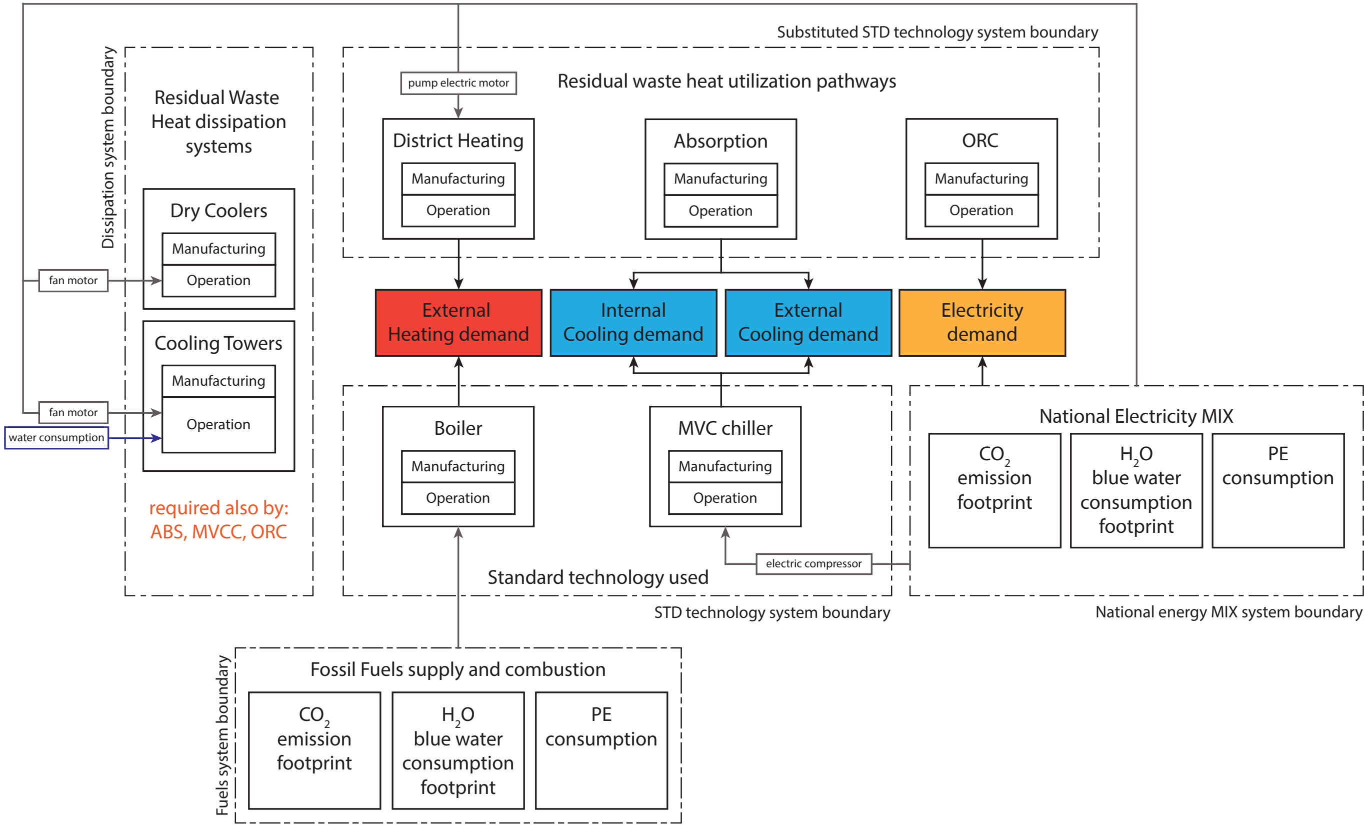


FIGURE 3.4: System boundaries chosen in this work.

is ignored. This because it is assumed that the plants used by the national energy mix already exist.

It can be noted that this reference system represents a refurbishment situation, which is more common than new constructions for industrial and commercial areas within the countries of concern. We assume that existing dissipation systems are maintained, and replaced when necessary. The impact of final disposal stages is out of the system boundaries for footprint calculation in this work.

Five zones can be distinguished in Figure 3.4:

- residual waste heat dissipation systems
- residual waste heat utilization pathways
- standard technology used
- national electricity mix
- fossil fuels supply and combustion

Each zone indicated is delimited with a dashed line. Colored boxes define the type of energy demand: the red box is for *external* heating demand, the blue boxes are for *internal* and *external* cooling demand, and the yellow box for electricity demand.

A brief digression about the type of demands needs to be presented. Energy demands can be treated as internal or external, by considering if the energy produced is used on-site or coupled with a transport network.

Electricity demand can be distinguished as internal or external, but in this work we consider only one type of demand since we assume that the ORC configuration is connected with the national electrical grid, or when it is not connected, we assume that the electricity produced is used internally and therefore not purchased from the national grid.

A different approach to cooling and heating demand is required. For these two demand types a distinction between internal and external takes on a stronger meaning. Internal demand of cooling energy is often produced on-site in industries where process fluids or dedicated components need to be cooled. An external cooling demand might be represented by the cooling energy demand required by a district cooling system.

The same distinction can be applied to heat demand, for example external heat demand might be represented by a district heating network, while internal heat demand might be represented by internal space heating or process pre-heating. The latter is not taken into account.

STD technology system boundary consider the evaluation of footprints for two configurations, i.e. Boiler and MVC chiller considering Manufacturing and Operation, in particular manufacturing is taken into account only if along the lifetime period analyzed the plant need to be substituted. Electricity or fuel consumption are accounted for in a way depending on the configuration. The same approach is also used for the “substituent” heat recovery configuration: manufacturing and operation are taken into account, as well as fuel and electricity consumption, based on lifetime and plant performances.

The last boundary system refers to the “dissipation system”. To avoid excessive connecting lines, which would make the figure less legible, all the links to MVC, ABS and ORC have been removed and replaced with the specification at the bottom, underlining that it has not been neglected. Manufacturing and operation are accounted for using the same rules for the previous cases.

3.6 Stationary and dynamic analysis

Energy production is the key element of the analysis proposed in this thesis. All the nexus indicators can be calculated based on the energy consumption. Efficiencies on heating, cooling, and electricity production are different due to the fundamental difference in terms of energy quality related both to the thermodynamic cycles as well as the parameters involved.

The complexity of the analysis differs remarkably depending on the accuracy of the energy production evaluation. The components involved in all the configurations considered are subjected to different efficiencies based on variables such climate, wet bulb temperature, dry bulb temperature, air humidity, heat source temperature, waste heat availability and used energy demand variability. There are two approaches concerning the energy demand evaluation and the efficiencies on the configurations analyzed:

- With average inputs (stationary). In this case the elements of the plant are supposed to work with constant inputs and variables, evaluated and averaged in a certain period of time (typically one year).

The system remains in the same state as time elapses, in every observable way. This is an approximation used when the analysis required has more of a focus on other aspects, such as economic concerns, rather than energy balances.

- Considering a dynamic demand and a dynamic system response. In this case the system is subjected to variable inputs. For this purpose commercial software or calculation codes are used to evaluate the system response.

Dynamic simulation of plants is necessary to have a clear understanding of the trends of energy demands and energy production efficiencies when taking into account waste heat availability and different climates. This is particularly true for condensers, and for refrigeration and electricity production systems in general.

3.6.1 Transient simulation tool

The simulations in dynamic conditions of heat recovery configurations as well as the reference configurations, were performed with TRNSYS©. TRNSYS (TRaNsient SYstem Simulation program) is an energy simulation program used for transient systems simulations. The structure is modular and the libraries consist of components such as HVAC, electronic, controls, hydronics, as well as routines to handle input considering of weather data or other time-dependent forcing functions.

The graphical user interface (GUI) is called Simulation Studio, and it has been developed to easily import libraries of components, called *types*.

The types of the studied configuration can be connected to each other in the workspace, allowing the plant systems to be visually constructed.

Dynamic simulation of systems is performed in TRNSYS according to an input – output logic (the outputs of one component are connected to the inputs of another), where each *type* is considered as a “black box”. Every type processes input data as a function of inbuilt algorithms with user-defined parameters, and produces output data. The task of each *type* is to solve, for every time step of the simulation, the equations associated with that type as functions of inputs derived from connected types. In this way, complex systems can be studied and analyzed in detail.

Main applications include: solar systems (solar thermal and photovoltaic systems), low energy buildings and HVAC systems, renewable energy systems, cogeneration, and fuel cells. The version used for the simulations performed in this thesis is 17.02.0005.

3.7 Uncertainty management

3.7.1 Sensitivity analysis

The aim of the uncertainty and sensitivity analyses is to study how the model response is affected by uncertainties in the inputs. Uncertainty analysis (UA) evaluates the output variability while sensitivity analysis (SA) describes the relative importance of each input in determining this variability [78]. The purpose of SA is to characterize how model outputs respond to changes in input, with attention given to finding the input parameters for which outputs are the most sensitive. Such approach can be used to address the following issues [79]:

- testing the robustness of the results of a model or system under uncertainty input
- increased understanding of the relationships between input and output variables
- uncertainty reduction: identifying model inputs that cause significant uncertainty in the output
- verify whether there are unexpected relationships between inputs and outputs, which can be considered errors

Sensitivity analysis methods can be generally classified into local and global methods [78]. Local sensitivity measures, often referred to as “one at a time” (OAT), assess how uncertainty in one factor affects the model output, keeping the other factors fixed at a nominal value. The main drawback of this approach is that interactions among factors cannot be detected, since they only become evident when the inputs are changed simultaneously. Global measures instead offer a comprehensive approach to model analysis, since they evaluate the effect of a factor while keeping all others are variable, efficiently exploring the multidimensional input space.

From the methodological point of view, environmental and economic indicators are required in order to take into account the benefits for the climate and the economic feasibility of projects. As one might expect, the more eco-friendly a project is the more expensive. The economic feasibility of industrial project can be changed if taxes or incentives are taken into account. When policy-makers are interested in reducing climate change they can influence decision-makers or promote energy efficiency with incentives or by increasing the taxes on emissions.

3.7.1.1 Local sensitivity analysis: OAT

One of the simplest and most common approaches to sensitivity analysis is that of changing "one-factor-at-a-time" (OFAT or OAT), in order to to understand what effect it produces on the output. OAT commonly involves moving one input variable, keeping others at their baseline (nominal) values, returning the variable to its nominal value, then repeating for each of the other inputs in the same way. Sensitivity may then be measured by monitoring changes in the output [79].

Furthermore, by changing one variable at a time, it is possible to keep all other variables fixed to their central or baseline values and thus increasing the comparability of the results (all "effects" are computed with reference to the same central point in space).

As a result of its simplicity, this approach does not fully explore the input space, since it does not take into account the simultaneous variation of input variables. This means that the OAT approach cannot detect the presence of interactions between input variables.

3.7.1.2 Scenario analysis

Scenario analysis may represent an useful approach when applied in energy policy modelling, since it provides a picture of future alternative states of an energy system in the absence of additional policies ("reference" or "baseline" scenarios). In this way scenarios are a device to assess the impacts of an energy system on the environment, and to point out the effectiveness of environmental policies at avoiding these impacts [80].

In addition, scenario analysis can illustrate how alternative policy pathways may or may not achieve an environmental target and they can identify the robustness of a particular environmental policy under different future conditions. This is important because "background" factors such as market conditions, change in consumption habits or other trends might affect the success of an environmental policy.

As clarified by Sullivan et al. [81] the simplest way to conduct a scenario analysis is to consider three estimates of the key parameters affecting the model results : a pessimistic, a most likely, and an optimistic estimate (O-ML-P estimating technique). Depending upon the nature of a parameter, the pessimistic estimate may be the lowest value (e.g. low value of carbon tax) or the largest value (such as high external cost value of pollutant emission).

3.7.2 Monte Carlo methods

Monte Carlo simulation has become an interesting tool for risk management and assessment, particularly when input data has high variability. In this thesis, Monte Carlo methods are used to account for the high variability of life-cycle parameters for different data sources, as well as data used for footprint evaluation based on the national energy mix.

As explained by [82] Monte Carlo methods are based on the analogy between probability and volume. The mathematics of measure formalizes the intuitive notion of probability, associating an event with a set of outcomes and defining the probability of the event to be its volume, or measure, relative to that of a universe of possible outcomes. Monte Carlo uses this identity in reverse, calculating the volume of a set by interpreting the volume as a probability. In the simplest case, this means sampling randomly from a universe of possible outcomes and taking the fraction of random draws that fall in a given set as an estimate of the set's volume. The law of large numbers ensures that this estimate converges to the correct value as the number of draws increases. The central limit theorem provides information about the likely magnitude of the error in the estimate after a finite number of draws.

Consider, for example, the problem of estimating the integral of a function f over the unit interval:

$$\alpha = \int_0^1 f(x)\,dx \tag{3.8}$$

as an expectation $E[f(U)]$, with U uniformly distributed between 0 and 1. Suppose we have a mechanism for drawing points $U_1, U_2, \ldots$ independently and uniformly from $[0,1]$. Evaluating the function f at n of these random points and averaging the results produces the Monte Carlo estimate:

$$\hat{\alpha} = \frac{1}{n}\sum_{i=1}^{n} f(U_i) \tag{3.9}$$

If f is indeed integrable over $[0,1]$ then by the strong law of large numbers,

$$\hat{\alpha}_n \to \alpha \text{ with probability 1 as } n \to \infty \tag{3.10}$$

If f is square integrable and we set:

$$\sigma_f^2 = \int_0^1 \left(f(x) - \alpha^2\right) dx \tag{3.11}$$

then the error $\hat{\alpha}_n$ in the Monte Carlo estimate is approximately normally distributed with mean 0 and standard deviation $\sigma_f/\sqrt{n}$, the quality of this approximation improving as n increases. The parameter σ_f would typically be unknown in a settling in which α is unknown, but it can be estimated using the sample standard deviation [82]:

$$\hat{s}_f = \sqrt{\frac{1}{n-1}\sum_{i=1}^{n}\left(f(U_i) - \hat{\alpha}_n\right)^2} \tag{3.12}$$

Thus, from the function values $f(U_1), \ldots, f(U_n)$ we obtain not only an estimate of the integral α but also a measure of the error in this estimate. The form of the standard deviation error $\sigma_f/\sqrt{n}$ is a central feature of the Monte Carlo method. Cutting this error in half requires increasing the number of points by a factor of four, and adding one decimal place of precision requires 100 times as many points [82]. These are a result of the square-root convergence rate implied by the $\sqrt{n}$ in the denominator of the standard error. In contrast, the error in the simple trapezoidal rule

$$\alpha \approx \frac{f(0) + f(1)}{2n} + \frac{1}{n} \sum_{i=1}^{n-1} f(i/n) \tag{3.13}$$

is $O(n^{-2})$, at least for a twice continuously differentiable function f. Monte Carlo is generally not a competitive method for calculating one-dimensional integrals. The value of Monte Carlo as computational tool lies in the fact that its $O(n^{-1/2})$ convergence rate is not restricted to integrals over the unit interval.

Indeed, the steps outlined above extent to estimating an integral over $[0,1]^d$ (and even $\Re^d$) for any dimension d. When we change the dimension we change f, and when we change f we change σ_f, but the standard error will still have the form $\sigma_f/\sqrt{n}$ for a Monte Carlo estimate based on n draws from the domain $[0,1]^d$. In particular, the $O(n^{-1/2})$ convergence rate holds for all d. In contrast the error in a product trapezoidal rule in d dimension is $O(n^{-1/2})$ for twice continuously differentiable integrands; this degradation in convergence rate with increasing dimension is characteristic of all deterministic integration methods. Thus, Monte Carlo methods are attractive for evaluating integrals in high dimensions.

A fundamental implication of Monte Carlo Methods is its application in financial engineering, in particular on investment risk analysis, which considers parameters or variables used in economic feasibility evaluation as stochastic.

In this work Monte Carlo methods are used to get a probability distribution of economic benefit indicators. As a consequence of this method, a measure of the risk could be operated also on the main inputs, showing which variables substantially affect economic indicators.

The uncertainty related to risks investment can be modelled with Monte Carlo methods, as shown in Figure 3.5. Investment cost estimates (in this case the output) results from a series of cost factors that can be modelled as statistical distributions.

Monte Carlo methods bypass the problem of a deterministic solution, that for a certain type of problems might be too expensive or even impossible, including phenomena with significant uncertainty in inputs such as

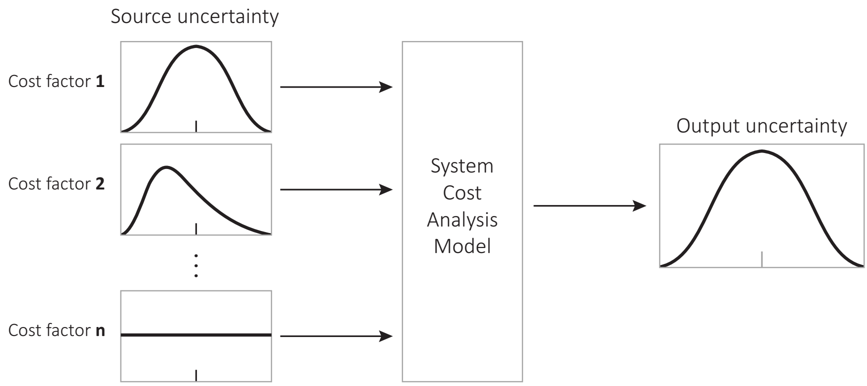


FIGURE 3.5: Risk analysis with Monte Carlo methods, the relation between input and output uncertainty.

the calculation of risk in business. The problem is solved numerically, producing a sufficient number n of possible combinations of input variables and calculating the output using the equations defined by the model.

The elements of the method can be briefly summarized as:

- *parameters*: user-specified input and thus controllable
- *exogenous input variables*: input variables of controlled events not under control that can be modelled with a probabilistic approach
- *output variables*: after a series of runs of the algorithm, using both deterministic and probabilistic inputs and the model equations, the outgoing data is analyzed providing a probabilistic distribution of the outputs
- *model equation*: the mathematical equations as a function of parameters and variables which describe the relations between the system elements and the links used for the output evaluation

To construct each of the n combinations, a random value for each input variable is generated (more precisely extracted) according to the specified probability distribution and respecting the correlations between variables.

Repeating this process, with n large enough to allow statistically reliable results, n independent values of the output variables are obtained, representing a probabilistic distribution of the problem solution.

The resulting statistical distribution can be analyzed, estimating descriptive parameters, reproducing histograms of frequencies, and obtaining the trends of output distribution functions.

The steps in Monte Carlo simulation here used can be summarized and described as:

1. Exogenous variables and parameter identification.
2. Model definition.
3. Probability distribution assessment and assignment.
4. Simulation runs.
5. Results validation and analysis.

Data collection and analysis is the first step in Monte Carlo simulation, the analysis of input data allows an understanding of critical elements that affect the resulting output. Even if a huge number of parameters and variables could be used, a trade-off between model complexity, implementation difficulties, and accuracy should be found, taking into account only relevant variables. To this end a pre-examination of data should be performed before the model building, highlighting relevant variables. Sensitivity analysis is a good tool to quickly understand which variables or parameters are the most influential.

Model definition refers to all the mathematical relations used to evaluate the output. It is also important that the model takes into account in an explicit way the correlation between variables.

For each variable a probability distribution must be specified. The best distribution that fits properly might be found in the literature, from data analysis, or from advising and training services. The definition of the best probability distribution for available datasets is a key element in this method and it is the subject of many studies and courses.

The results analysis is the last part of a Monte Carlo simulation, and it should be performed with the aim to verify whether there are inconsistencies. In such a case some parts need to be adjusted, calibrated, and re-evaluated. An input data re-analysis might be necessary.

3.7.2.1 The problem of the best distribution

One of the main problems with Monte Carlo Methods is related to the following question: "which distribution should I choose for a variable?".

Some datasets are based on historical series or based on the operator's experience. At a company level, data records are available based on the past events e.g. prices (buy and sell), machinery maintenance stops, working time, or break downs. In this situation statistical techniques could be used to perform a "best fit" of historical data series, finding the best distributions.

Another method of getting the best fit is to use a "re-sampling method". The base concept is to extract (with reintroduction) random values directly from the original series, which are used directly in the calculations. An advantage of this method is that no statistical distribution needs to be defined and the variability associated to the input data is not lost. Using historical data series works under the implicit assumption that a description of future events can be derived from the historical data, which is a limitation of such data.

Another simple method often used is to calculate three values, namely *pessimistic*, *realistic* and *optimistic*, for the worst case, the most probable and the best case respectively. These values are used to build a triangular distribution associated to the variable.

When no historical data is available a subjective analysis can be performed. Subjective analysis refers to a probabilistic distribution declared based on the analyst's experience. This occurs frequently in the economic field, particularly in investment evaluation.

3.7.2.2 Random number generation

At the core of all Monte Carlo simulations there is a sequence of apparently random numbers used to drive the simulation. In analyzing methods, we will treat this sequence as though it were genuinely random. This is convenient fiction that allows us to apply tools from probability and statistics to analyze Monte Carlo computations. This is convenient because modern *pseudorandom* number generators algorithms are sufficiently good enough at mimicking genuine randomness to make this analysis informative [82]. Nevertheless, we should be aware that apparently random numbers at the heart of a simulation are in fact produced completely by deterministic algorithms.

3.7.2.3 Correlation between input variables

A central issue in Monte Carlo model building is the correlation between inputs, in particular, the common assumption that all the inputs are independent of each other sometimes leads to a results that are not representative of reality.

A method to handle the dependence between two variables consists of treating one variable as independent from the other, while the other is considered dependent and calculated separately. The core is to divide the possible values of the independent variable in intervals, and for each interval a probabilistic distribution is associated for the dependent variable.

During the simulation, on every interaction the first step is to generate a value for the independent variable which consequently determines a distribution used to evaluate the dependent variable.

The implementation of this method as well as the way of generating the correlations between inputs are complex, and can lead to incorrect results and evaluations if not implemented properly.

3.7.2.4 Number of iterations required

The simulation output of a Monte Carlo algorithm is a random variable, consequently an analytic formulation of the solution is not possible. However it is possible to get a sample of values where the frequency and the probability distribution are an approximation of the analytic results.

It is possible to demonstrate, using the central limit theorem, that significant precision and accuracy can be reached by increasing the number of simulation runs, giving more samples. The greater the number of iterations the better the convergence to exact values. Figure 3.6 shows graphically the convergence of π as a function simulations run (random points generated).

The convergence of the output to exact value grows with the number of the simulations performed, figure 3.7 shows the error of the attempt to evaluate π. Since the power of the computer used was enough, a high number of simulations were able to be performed in order to evaluate the economic functions stud-ied in this work.

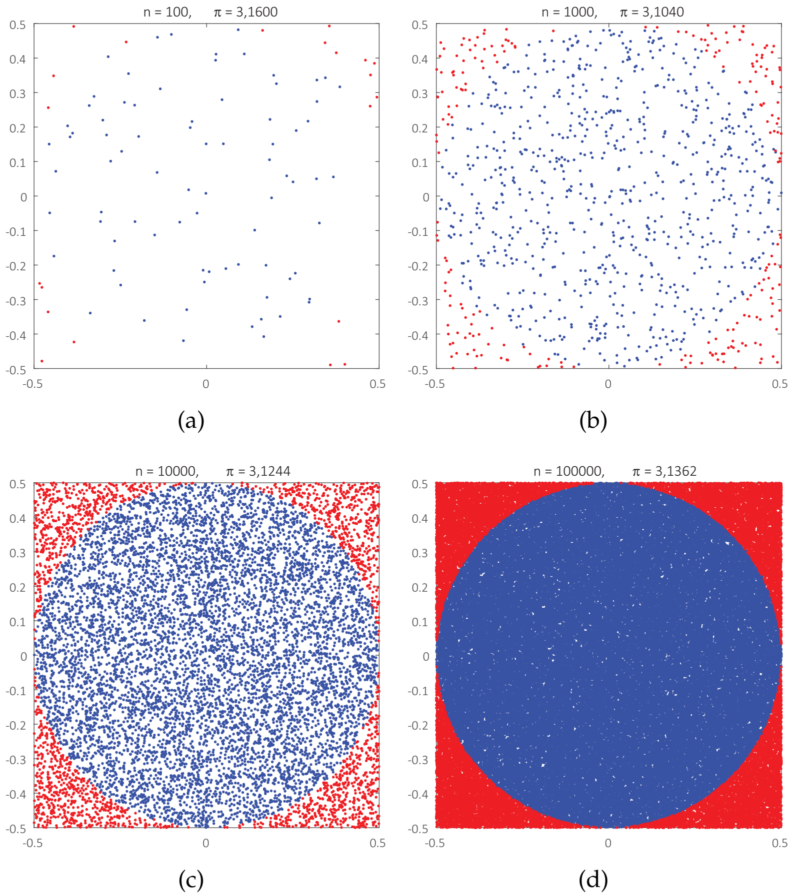


FIGURE 3.6: Estimate of π with Monte Carlo Method. Starting from disk equation $r^2 = x^2 + y^2$ and using a random point generation, an estimate of the area is evaluated based on the number of points, their area and the point's position inside or outside the disk. (a) $n = 100$ and $\pi_{apx} = 3.1600$; (b) $n = 1000$ and $\pi_{apx} = 3.1040$; (c) $n = 10000$ and $\pi_{apx} = 3.1244$; (d) $n = 100000$ and $\pi_{apx} = 3.1362$.

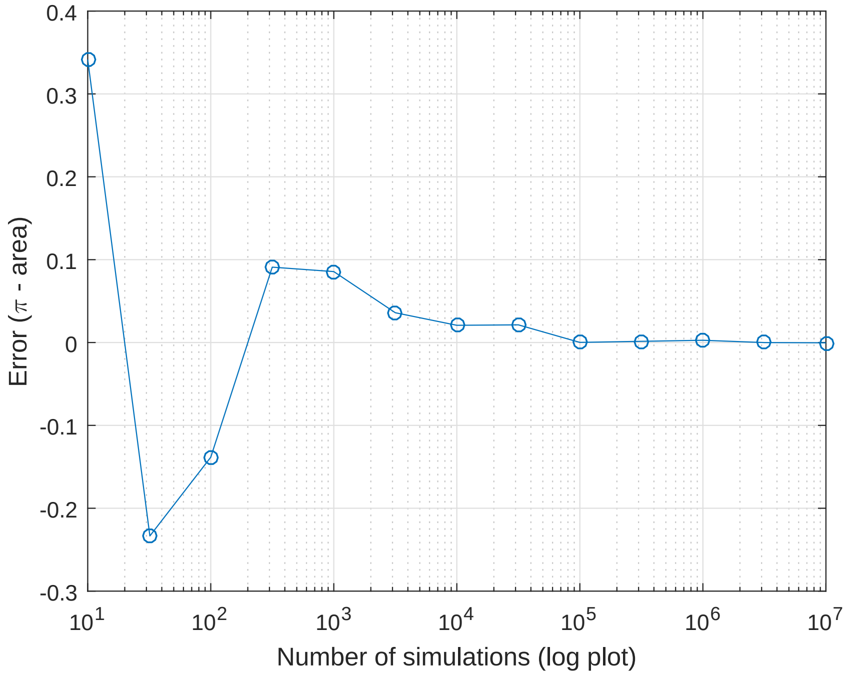


FIGURE 3.7: π error over number of simulations.

3.7.2.5 Outputs evaluated in this work

Monte Carlo methods are a powerful tool for assessing the economic risk of investments. The output generated will be a probabilistic distribution of LCC and PB. This method is used also to evaluate uncertainty of the footprints on the national energy mix.

In particular, an estimate of water, carbon footprints and primary energy consumption was performed, based on the technology used for electricity generation in different countries. Furthermore a cost analysis of electricity and water for industrial purposes was performed.

Limitations about the method are related mainly to the data collection, model building, and the correct distribution assigned to variables. As well as pre-processing problems, there are problems related to the analysis and interpretation of the output results. The decision maker should be aware of the limits of the model, and in particular accountability resides entirely with the analyst.

3.7.2.6 Monte Carlo simulation tool

In this work the data uncertainty handling and cost analysis were performed using @Risk® which is an add-in to MS Excel® designed for Monte Carlo simulations with a long-established reputation of computational accuracy, modelling flexibility, and ease of use. With this add-in it is possible to load raw data on Excel spreadsheets, define the distribution functions associated, and run the simulations. Libraries and functions can be called using the designated *ribbon* on MS-Excel® after installation.

Examples of probability distributions already available are: normal, lognormal, uniform, triangular, Rayleigh, PERT, and discrete. During the simulations, random values are taken from the probability distributions associated with the input of the model, and by repeating and recording this procedure hundreds or thousands of times a probability distribution of possible outcomes is found. The software allows quick plots and statistical reports on the simulations performed to be made.

The @Risk version used is 7.51, developed by PALISADE® under "The Decision Tools Suite 7.51".

Chapter 4

Case studies and model building

After a description of the case studies is analyzed, the model building for each case is described, focusing also on the datasets used. The case studies proposed in this thesis derive from real plants, selected based on the criteria shown in figure 1.1. Starting from a case study of low grade waste heat recovery for absorption cooling or for power generation through ORC at an Italian electric steelmaking site, a generalization was made by evaluating the same project but considering different climates, as well as different prices of electricity, water, and carbon taxes.

In order to generalize further, case study *C2* was is based on heat recovery from a generic industrial waste heat source, which has a different availability of waste heat and a different energy demand to that of previous cases. Additionally, the fact that the energy demand of a district heating network is always off-site necessitates the introduction of the distinction between internal and external energy demand.

Figure 4.1 shows the study cases allocation, and corresponds to figure 1.1 from chapter 1. The low grade waste heat recovery configurations are compared with conventional technologies used for the same purposes. The case studies delimited with dotted lines are now briefly introduced.

Case study *C1a* refers to an EAF steelmaking plant operating 7000 h/y, where flue gases coming from the furnace are cooled down in a special heat exchanger called a water cooled duct (WCD). The heat recovery opportunities are related to internal reuse of electricity and cooling energy using ORC and absorption cooling respectively. The base case involves no

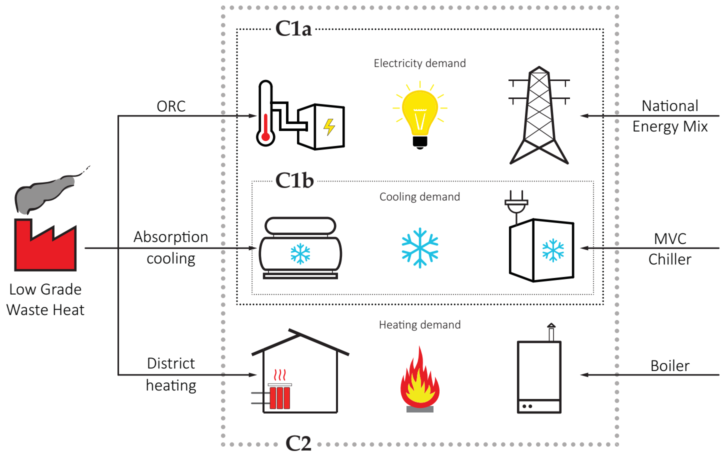


FIGURE 4.1: Case studies allocation based on figure 1.1 from chapter 1.

heat recovery, while a mechanical vapor compression chiller is used for internal cooling demand. Cooling demand is assumed as constant during operating hours, as are the performances of condensers, chillers, ORC and heat recovery unit. The sum of direct and indirect blue water consumption, carbon emissions and primary energy demand are evaluated.

The approach assumed in the previous case is balanced by case study *C1b*, where heat recovery configurations and standard cycles are modelled, accounting for the dynamic performance the plant. The simulations account for climate variations, performance of condensers, chillers, and waste heat availability at the heat exchanger recovery unit. The cooling demand derives from the electric transformation cabins, which is simulated in order to evaluate the influence of climate. A more precise evaluation of the nexus indicators is consequently given based on the simulation performed.

In case study *C2* the water footprint of district heating options for recovering low grade industrial waste heat are investigated. The sum of direct and indirect blue water consumption, carbon emissions and primary energy demand are all evaluated. A point-to-point model of a district heating system is simulated, recovering waste heat from an industrial site to heat an office building, which is considered to be an end-user.

The energy demand is simulated during the winter and summer period, additionally introducing the distinction between internal and external energy demand depending on whether the energy is used at the industrial site or at the remote end-user. This case is selected in order to get a generalization of the results evaluated in the previous cases, and also introduces a discontinuity in the waste heat available.

A parametric study of economic feasibility is performed in order to evaluate under which circumstances the additional water and carbon footprint, produced by district energy systems' construction and operation, is offset by the reduction in fossil fuel consumption, caused by the substitution of remote boilers. Differentiation between internal and external energy demands is introduced. A comparison is drawn between district heating-cooling technologies, and energy conversion via organic Rankine cycles, as well as absorption chilling for internal cooling demand.

The details about case study *C1a* are presented in section 4.1, *C1b* is presented in section 4.2, and C2 is presented in section 4.3.

4.1 Case study C1a

In the steelmaking industry, awareness of resource efficiency problems is high. In fact, iron and steelmaking is an energy intensive sector which currently accounts for about 18% of the primary energy consumption and 11% of the total electricity consumption of European industries [83].

As a consequence, steelmaking is also a carbon intensive sector, accounting for 5% of total CO_2 emission in the world [84]. The steelmaking industry is currently subjected to emission trading schemes (ETS) in several countries, in particular in the European Union (EU ETS), where a market of carbon emission allowances was introduced in 2005 to meet international commitments under the Kyoto protocol [85].

Steelmaking processes also require large water flows (about 28 m^3 per ton of steel) [68], mainly used for cooling purposes, and some studies are concerned with water footprint calculation for the sector [86]. A position paper published by the World Steel Association in 2015 [68] expresses a nexus view of the sector, fostering a holistic approach which should consider additional energy requirements and all environmental aspects when introducing water management policies and evaluating discharge reduction projects.

There are two main technology pathways for steel production: either iron extraction from iron ore and refining through a reduction process based on blast furnaces and basic oxygen furnaces (BF-BOF), or recycling steel scrap through a melting process performed in an electric arc furnace (EAF).

In this work the EAF route is considered, which currently generates about 30% of global steel production [87], because recycling is expected to increase in the next few years and because this route is usually characterized by higher water consumption. In fact, because of the magnitude of the materials and water flows involved, BF-BOF sites are usually located close to natural water sources (typically the sea) and cooling is performed through once-through cooling systems [68]. These systems, which take water from the sea, circulate it through the plant heat exchangers and return it to the local source, are characterized by high water withdrawals, but relatively low water consumption. Typical EAF plants, on the other hand, are relatively small systems located close to end markets of steel, usually in inland areas, which often use closed-loop cooling systems. These systems reuse cooling water in a second cycle rather than immediately returning it back to the original water source.

Most commonly, wet recirculating systems use cooling towers to expose water to ambient air. Some of the water evaporates; the rest is then sent back to heat exchangers for process cooling. These systems have much lower withdrawals than once-through systems, but tend to have appreciably higher water consumption [55].

To avoid excessive water consumption in EAF systems, closed circuit forced air-cooled systems are a widely used alternative to wet recirculating systems. In this case, the process medium itself or an intermediate coolant (typically water) is cooled down by conduction and convection through a forced air stream which flows past the outside of the tubes. Because the heat capacity of air is low and the coefficient of conduction and convection is also low, large air flows and heat exchanging surfaces are needed. Capital costs and energy consumption are consequently higher than in wet cooling systems with similar performances, but water make-up requirements are virtually null.

The company of concern is an EAF steel mill operating on a 24/7 basis, employing about 600 people, with a yearly production of about 1.5 Mt of steel, subject to EU-ETS obligations for the reduction of GHG emissions.

FIGURE 4.2: An electric arc furnace, note the elbow with a gap collecting the hot flue gases exiting the furnace (left side of the electrodes) source: `https://ru.wikipedia.org/wiki`.

The waste heat recovery opportunity of concern derives from the first part of the off-gas cooling system of the EAF, the so-called WCD, which is represented in Figure 4.3 and described in [58]. The structure of the off-gas cooling system is typical for EAF processes, and is described in detail in Ref. [88].

The off-gas enters the settling chamber, where larger particles are separated to reduce sediments in subsequent sections, flows through the WCD which cools it to about 600 °C, and is further cooled to 200-300 °C by a quenching tower (QT). The primary gas at 200-300 °C is then blended with secondary gas at 50-70 °C from the canopy hood situated over the furnace, so that the final mixture reaches a temperature which allows further particulate removal in a cyclonic separator and in the fabric filters of the baghouse collector.

For the heat recovery system of concern, we have considered the opportunity to derive a water flow from the cooling water circuit corresponding to a heat flow of about 1000 kW. Such heat flow is, however, only a fraction of the total heat flow available at the WCD. Hot water leaving the WCD currently enters a forced-air cooling device at temperature T11 and leaves it at temperature T12, 10 °C below T11.

T12 is based on the average EU-15 external dry bulb temperature, which was then increased by the exchanger temperature difference, such that the final value fell in the range suggested by Ref. [50].

The process is intermittent, as the EAF operates as a batch melting process based on the so-called tap-to-tap cycle, which includes furnace charging, melting, refining, de-slagging, tapping and furnace turn-around. The tap-to-tap time is about 40 min, which results in a typical pattern in flue gas temperatures described thoroughly in the literature [89]. Variations in flue gas temperatures correspond to oscillations in cooling water temperature at the heat recovery outlet (T1 in Figure 4.3). Because a smoother temperature profile is needed for most recovery options, a hot water tank is interposed as a storage system. Figure 4.4 shows temperature profiles of hot water leaving the tank (T2 in Figure 4.3) depending on storage size for the identified 1000 kW waste heat flow. Temperature oscillations within a range of ±5 °C, i.e. between 85 °C and 95 °C, were deemed acceptable, and a 100 m^3 hot water storage system was selected.

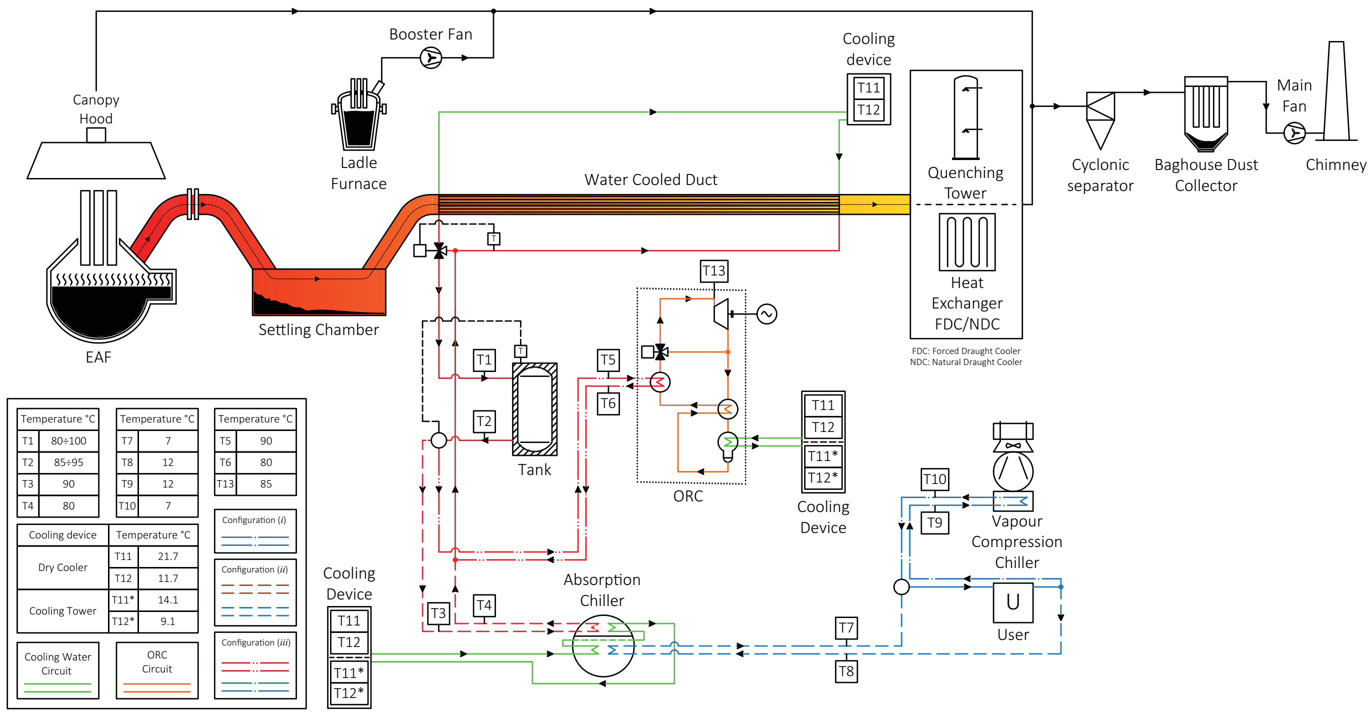


FIGURE 4.3: Process scheme and waste heat recovery options [58].

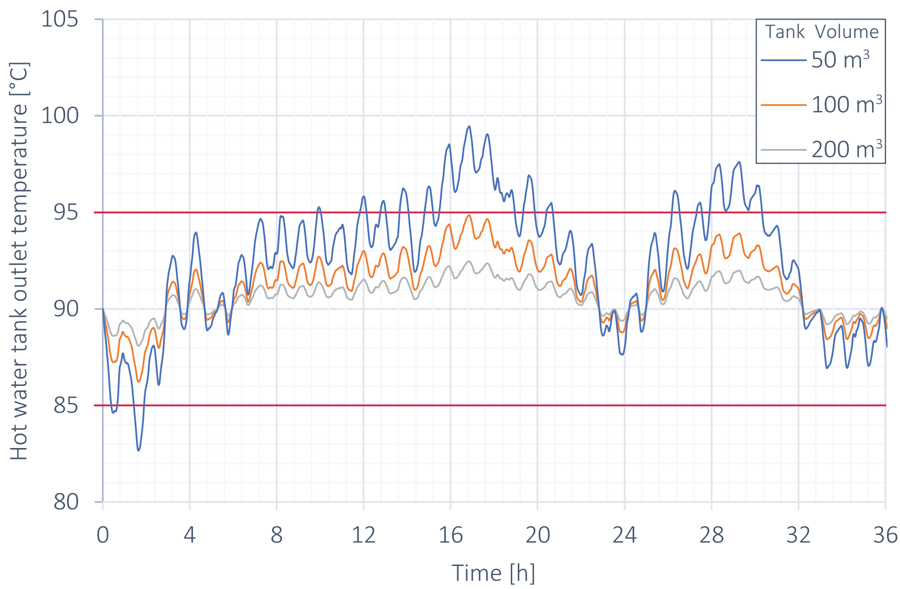


FIGURE 4.4: Temperature profiles of hot water at the tank outlet depending on tank size, valid for heat recovery configurations (*ii*) and (*iii*) [58].

4.1.1 Waste heat recovery options for case study C1a

Literature on heat recovery for steelmaking processes [45] suggests direct use of heat, e.g. via district heating systems, as a first option. For such projects to be economically feasible, suitable heat sinks within an economically feasible distance should exist, which is not the case for the system of concern and for many similar sites in Europe.

Even direct use of heat within the steel mill has not been considered in this case study, because low temperature internal heat demand is already met with other waste heat flows.

If the site has a suitable process or ambient cooling demand, requiring chilled water at about 10 °C, and waste heat at suitable temperature levels (typically above 80 °C, thermally driven cooling systems, particularly based on lithium bromide absorption cooling [90] can also be considered as an active waste heat conversion technology.

At electric steelmaking sites, chilled water is required for air conditioning of the electric transformer, generator and switch cabinets, mostly located within factory sheds.

At the steel mill in question, chilled water at 7 °C outlet temperature is currently obtained by several vapour compression chillers meeting an average cooling load of 500 kW with an average measured energy efficiency ratio (EER)[1] of 4. The cooling load is represented by user *U* in Figure 4.3 and its existing circuit, entering user at T9 = 7 °C and exiting at T10 = 12 °C, is represented with blue dashed line.

Every refrigeration cycle, both mechanical compression and absorption based, requires heat to be discarded to the environment to enable condensation of coolant fluid at the condenser. Refrigerators are thus usually coupled with heat dissipations systems, either dry (i.e. with forced-air) or wet. While domestic and small scale systems are air-cooled, for large scale refrigerators used in industrial contexts the choice between dry cooling and wet cooling is determined by the expected economic performance.

For the case study in question, proposed also in [58], condensation of

[1] In accordance with standard EN 14511, the characteristic parameter of a refrigerator is the energy efficiency ratio (EER), defined as the ratio of the total cooling capacity to the effective power input of the unit, expressed in Watt/Watt.

the refrigerant in the vapor compression units is currently performed exchanging heat to air, i.e. with dry cooling. Thus, the reference case, representing the current situation, is identified as configuration (*i*) in Figure 4.3 and in the following. Configuration (*i*) includes the independent vapor compression cooling system represented with blue dashed line in Figure 4.3, and no heat recovery from the WCD. In Figure 4.5, which summarizes the analyzed system configurations, boundaries and direct flows, it is shown that only electricity is consumed in configuration (*i*), because water circuits are closed and dry cooling is used for WCD water cooling and for the chiller.

When waste heat is recovered to drive absorption cooling machines, only single effect absorption chillers can be used, because hot water is available at an average temperature of T3 = 90 °C, associated with oscillations between 85 °C and 95 °C as detailed above. A reference EER of 0.7 is assumed for these machines, based on manufacturers' catalogues [91, 92] and the literature [90].

In this configuration, identified as (*ii*) in Figure 4.3, Figure 4.5, substitution of vapor compression units with single effect absorption chillers is associated with the centralization of heat dissipation systems, which makes it reasonable to consider cooling towers as an option for dissipating heat from refrigeration cycles.

Figure 4.5 shows that configuration (*ii*) requires direct electricity demand for auxiliaries and circuit pumps. If cooling towers are used, a direct water consumption is also required. In this case, water enters cooling towers at T11* and leaves them at T12*. T12* and T11* are based on a cooling tower approach and range, respectively, falling in the intervals suggested by Ref. [50], starting from the average EU-15 outdoor wet bulb temperature.

Brückner et al. [40] report that, assuming an operation time of 2500 h/year, absorption cooling is of little interest for industrial consumers requiring high returns on their investments. However, cooling of internal electric cabinets within a process plant working on a 24/7 basis is a basic process requirement, likely to be interrupted only during protracted production stops or for maintenance. A yearly operation time of 7000 h/year can thus be reasonably assumed for these auxiliaries in steel mills.

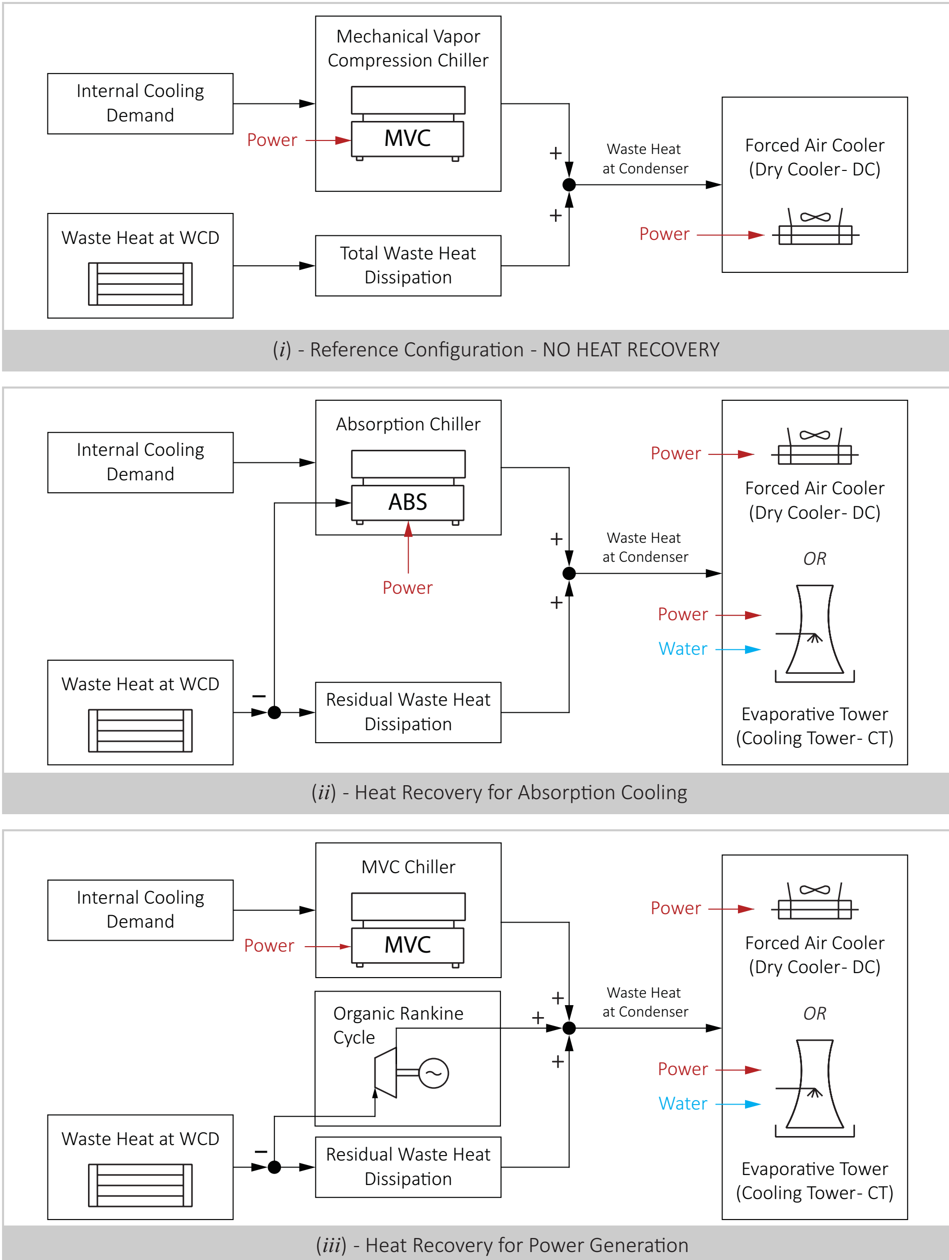


FIGURE 4.5: Summary of electricity and power flows for the reference configuration (*i*) and the alternative recovery options (*ii*) and (*iii*) [58].

If a direct use of waste heat is not feasible, power generation is considered as an energy conversion option to exploit waste heat. In particular, for low grade waste heat available from heat sources at temperatures higher than 80 °C, including cooling water from EAF and heating furnaces, organic Rankine cycles (ORC) are identified by Ref. [45] as the most economically attractive conversion technology, not least because of their commercial readiness [38]. Although the efficiency of ORCs at low temperature is necessarily low, “even technologies with low conversion efficiencies can be of interest if there is no other use for the excess heat” [45].

Heat recovery through an ORC is thus an option considered in this study, represented in orange and connected to the dotted-dashed lines in Figure 4.3, and identified as configuration (iii) in Figure 4.3 and Figure 4.5. ORC is introduced as an alternative option to allow economic and technical comparisons, as well as for the purposes of generalization to other process industries in Europe, as it is possible that industries with similar heat recovery opportunities do not have similar low temperature cooling demand.

To enable comparison, in configuration (*iii*) we assume that the same heat flow as in (*ii*) is recovered for power generation, i.e. about 700 kW. Considering an average inlet temperature (T5 in Figure 4.3) of 90 °C and an outlet temperature (T6) of 80 °C, a minimum temperature approach of 5 °C between the heat source and the working fluid is assumed. At these conditions, the estimated efficiency of the ORC is about 9%, which is in line with values reported by Ref. [31] and by Ref. [44] for heat source temperatures above 80 °C.

4.1.2 Energy and water flow balances for reference and heat recovery options

Waste heat flows Q_r to be dissipated for condensation in refrigeration cycles can be estimated according to equation 4.1 as a function of useful cooling effect Q_c.

$$Q_r = \left(1 + \frac{1}{EER}\right) \cdot Q_c \tag{4.1}$$

As absorption based refrigeration has a lower EER than MVC-based refrigeration, relevant waste heat flows are higher. In the case at hand however, one should consider that waste heat comes from a cooling water circuit, which would require a cooling system anyway (a dry cooler at the moment) to dissipate the waste heat flow Q_w to cool water down to 80 °C on average. Recovering a part of this flow to feed an absorption chiller leads to a reduction of total dissipated heat, which compensates for the relative increase in the cooling load at the condenser of refrigeration cycles due to its lower EER. The total cooling load at dry-coolers without heat recovery (configuration *i* in Figure 4.3) is thus given by equation 4.2:

$$Q_d = Q_w + \left(1 + \frac{1}{EER_c}\right) \cdot Q_c \tag{4.2}$$

whereas when heat is recovered for absorption cooling (configuration *ii*), the total load for the cooling system is given by equation 4.4:

$$Q_{HR,a} = Q_w - \frac{Q_c}{EER_a} + \left(1 + \frac{1}{EER_a}\right) \cdot Q_c = Q_w + Q_c \tag{4.3}$$

To obtain an electric power output P from the ORC in configuration (*iii*), the waste heat flow to be transferred from the process to the cycle evaporator equals P/η. Thus, the cycle energy balance, and particularly the heat to be dissipated at the ORC condenser, implies that the total load for the cooling system in this configuration is given by equation 4.4:

$$Q_{HR,p} = Q_w - \frac{P}{\eta} + (1-\eta) \cdot \frac{P}{\eta} + \left(1 + \frac{1}{EER_c}\right) \cdot Q_c \tag{4.4}$$

Having assumed that in configuration (*iii*) the same heat flow as in (*ii*) is recovered for power generation, the electric power output can be expressed as $P = \eta \cdot Q_c/EER_a$. Thus, the total load to be dissipated in configuration (*iii*) is (equation 4.5:

$$Q_{HR,p} = Q_w - \frac{\eta \cdot Q_c}{EER_a} + \left(1 + \frac{1}{EER_c}\right) \cdot Q_c = Q_{HR,a} + Q_c \left(\frac{1}{EERc} - \frac{\eta}{EERa}\right) \tag{4.5}$$

The heat load to be dissipated is larger in (*iii*) than in (*ii*). When evaluating energy and water consumption for alternative dissipation systems, for configuration (*iii*) it will be assumed that low temperature cooling systems are not modified and thus remain coupled with their current, dry cooling system. As a consequence, the heat load to be dissipated by the alternative cooling systems considered will equal to eq.4.6:

$$Q_{HR,iii} = Q_w - \frac{P}{\eta} + (1-\eta) \cdot \frac{P}{\eta} = Q - P \tag{4.6}$$

If wet cooling systems are used to dissipate residual heat, consumed water W can be estimated as a function of evaporated water (equation 4.7)

$$W = k\frac{Q_l}{L} \tag{4.7}$$

L being the latent vaporization heat of water (here set at 2.200 kJ/kg), Q_l the thermal load in kW and resulting W being expressed in kg/s.

Coefficient k accounts for additional water losses due to bleed off and drift. Since the water is recycled and there is an opportunity for water constituents to be concentrated in the evaporative step, the bleeding off of water with a high mineral content and making up with freshwater of acceptable quality is required to keep solid concentrations in water circuits below an acceptable threshold [55]. Losses due to drift are usually minimal, while based on empirical results reported by manufacturers [93], the effect of bleed off is comparable with evaporation, and $k = 2$ is thus a reasonable estimate.

If dry cooling systems are used, the direct consumption of water is zero. However, an indirect consumption of water is associated with the electrical energy consumption of these systems. In this work, the electrical consumption of dry cooling systems was empirically estimated by interpolating drive power demand data provided by manufacturers (e.g. Refs. [52], [94]) as a linear function of cooling load (equation 4.8):

$$P_D = a + b \cdot Q_l \tag{4.8}$$

Where Q_l is the thermal load in kW, $a = 4$ kW and b = 0.03 kWel/kW and the resulting power consumption P_D is expressed in kW. Both for dry and wet cooling systems, power consumption for water pumping should also be added, which is estimated as:

$$P_W = \frac{\Delta h \cdot Q_l}{\eta_p c_p \rho \cdot \Delta t} \tag{4.9}$$

with Δh being the circuit head loss in Pa (46 kPa for the system of concern), c_p the constant pressure specific heat in kJ/(kg °C), Δt the temperature difference, ρ the water density and η_p the pump electric efficiency.

4.1.3 Monte Carlo model building

Based on equations 4.1, 4.2, 4.3, 4.4, 4.5, 4.6, 4.8, 4.7, an energy system model has been built for the configuration of concern, which was integrated with economic data enabling the calculation of life cycle costs for each configuration. Configuration (*i*) is taken as reference, base case, and both the dry cooling (DC) and the cooling tower (CT) variant for each heat recovery configuration (*ii*), and (*iii*) are evaluated under the assumptions clarified with equation 4.6.

The economic feasibility of each alternative has been assessed for average EU-15 economic data, using sources detailed in Section 4, to generalize the evaluation of the case study of concern to the European context.

Variability in data and consequent uncertainty in estimates are high, so the Monte Carlo approach is taken, allowing synthesis with the various sources of uncertainty of the problem and to account for all possible values that can be assumed by uncertain parameters, weighted by their probability of occurrence [95]. The Monte Carlo approach is widely used in energy and environmental analysis [96], and has recently been applied for regional water-energy nexus evaluations by Refs. [97] and [98].

The modelling procedure through Monte Carlo simulation includes the following steps [99]:

1. Specification of uncertain model parameters.
2. Selection of a probability distribution describing the possible value range for each uncertain parameter.
3. Generation of the output variable by randomly selecting input values on the basis of the selected distribution for a large number of iterations.

In the present study, the technical parameters (i.e. conversion plant efficiency, EERs, cooling and power loads and parameters appearing in equations 4.1, 4.2, 4.3, 4.4, 4.5, 4.6, 4.8, 4.7, 4.9, have been assumed to be known with certainty. Uncertainty is associated with economic parameters including:

- capital costs of installed equipment;
- annual operation time;
- water and electricity prices;
- interest rate;
- investment duration, which in this sector is usually shorter than the technical lifetime of plants, this is related to economic obsolescence and payback constraints set by shareholders.

As in Ref. [99] this variation is representative of the unstable economic environment which is faced by investors making their medium term plans. Moreover, uncertainty is associated with emission, primary energy and blue water consumption factors for the purchased electricity, while the electricity generation mix of each country is assumed to be known with certainty, based on data derived from Refs. [100] and [101] for the year 2012.

Probability distribution types have been defined by fitting available data or by expert judgement in the case of limited data availability. In particular, a set of appropriate distribution shapes has been defined subjectively, based on minimum and maximum values of the possible range of uncertain parameters retrieved in literature. For instance, continuous distributions extending to infinity, such as the lognormal or the gamma distribution, have been excluded, assuming costs and environmental parameters are realistically bounded. A preliminary analysis of the extreme values reported in the literature was also performed to exclude outliers. After that, distribution fitting was performed with the commercial software @RISK [102], which uses the Akaike information criterion to select the best distribution type and maximum likelihood estimators to estimate distribution adjustable parameters [103]. Finally, the output was generated from repeated iterations.

The average of the repeated iterations is an unbiased estimator of the expected value, and the law of large numbers ensures that, for a large number of iterations, it converges to the expected value [103]. Establishing the number of iterations required to ensure convergence is a necessary step in obtaining estimates with an acceptable accuracy, i.e. within an acceptably tight confidence interval. The minimum number of iterations required to achieve a certain confidence interval can be calculated for each estimated parameter based on the central limit theorem as reported for example in Ref. [103]. In practice, as suggested by other authors who used similar software packages for a Monte Carlo analysis of investments in energy plants [103], in this study the number of iterations was automatically established by @RISK [102] to ensure that convergence was achieved for all simulation outputs within 3% of the actual value of the mean, at 95% confidence level.

4.1.4 Water, energy and carbon footprint evaluation

To evaluate water, energy and carbon footprints in this case study, the boundaries of the systems have been defined based on flows previously identified in Figure 4.5. In fact, these are the main relevant and differential flows for the examined configurations, because their level changes as a direct result of the decision between options (*i*), (*ii*) and (*iii*).

In a similar manner to [104], the assumption in this study is that the technology switch from vapor compression units to absorption cooling systems is not associated with changes in direct carbon equivalent emissions from refrigerant leaks. Also, carbon equivalent emissions from organic fluid leakages in the ORC system have been neglected: it is assumed that, among the numerous working fluids [44, 105, 106] which can be used within the temperature and efficiency ranges considered in this study, low global warming potential fluids [107] such as R152a, R600 and R601 are chosen. Thus, only emissions embodied in purchased electricity have been incorporated in carbon footprint evaluation. Similarly, primary energy consumption associated with purchased electricity was calculated based on site-to-source energy conversion factors [77].

4.1.5 Data collection and elaboration

A wide set of existing literature and data sources [15] [108] [109] [110] [111] [112] [75] [113] [114] [115] [116] [117] has been used to determine estimates for carbon emission, water consumption and energy consumption factors for each primary energy source; these estimates have been combined with power generation mix data to obtain coherent carbon, water and energy indicators for each country. For all the data sources used [15] [108] [109] [110] [111] [112] [75] [113] [114] [115] [116] [117] and for all indicators investigated, the estimates are based on a life cycle approach, i.e. all $CO_{2\ eq}$

emissions or water consumption from extraction to plant construction are considered. For this reason, emission factors are positive even for renewable sources which do not entail any combustion or direct use of water in their power generation cycle.

4.1.6 Economic input data

Investment cost distributions, which are the main sources of uncertainty for economic feasibility assessment, are based on literature [45], catalogues of manufacturers [52, 91–94, 118], and personal communications from their representatives in Italy, and are reported in Table A.1 in the form of size dependent cost functions, based on power and linear function shapes discussed e.g. in Ref. [56].

ORC is the most expensive technology [105], especially in the small capacity ranges associated with this application. Absorption chillers have a high proportion of size independent capital costs, which makes it advisable to avoid redundancies and load partitioning in order to minimize the number of units.

To assess the economic performance of generic plants, as in Ref. [99], triangular distributions are also used to estimate interest rates, investment duration and annual operation time. The expected values of these parameters, calculated under current conditions for the steelmaking sector, correspond to an interest rate of 7.3%, an investment duration of 7.3 years and about 6100 operation hours per year. The impact of different market conditions in various EU countries is analyzed by considering average prices of electricity and freshwater for industrial customers in each country. The difficulty in obtaining such data, particularly on recent costs of water for industrial customers, is recognized as a limitation of the present study, and was the main reason for using the Monte Carlo approach to deal with uncertainty.

For electricity, the Eurostat [119] and OECD [120] databases have been used. Uniform probability distributions were derived from values obtained for the years (2012–2014) from Ref. [119] and from the values reported by OECD [120] for the year 2013, bearing in mind that these values refer to industrial users with annual electricity consumption between 500 and 2000 MW·h.

For freshwater, the last comparative study on industrial prices in Europe dates back to the year 2003 [121] and, as for industry, reports data for seven countries only.

More abundant and recent literature concerns prices for households [120], [122]. Our approach is thus to extrapolate the ratio between industrial and residential consumer prices from Ref. [121], obtaining an expected value of 77.5%, and to apply this coefficient to the household price distributions derived from the data of several cities, based on [120] and on other data sources as reported in Table B.1. Table B.1 presents the expected values of the probability distributions of electricity and water price obtained for each country.

In particular, for Greece [123] and Luxembourg [124] direct data on industrial tariffs could be found. For all countries, uniform price distributions were assumed and their expected values are reported in Table B.1. A statistical correlation test has been performed for values reported in Table B.1, finding that correlation is not statistically significant. For this reason, it has not been incorporated in the Monte Carlo model.

4.2 Case study C1b

The footprint-evaluation oriented approach assumed in the previous case is balanced by this case study since dynamic performance of the elements involved in the various configurations are simulated.

The aim of case study *C1b* is to verify the dynamic behavior of cooling configurations studied in *C1a*, i.e. to assess the impact of cooling configurations for low grade waste heat recovery for steelmaking plants located in 16 climate zones defined by ASHRAE 90.1 [125], in terms of energy, carbon and water impact. Taking a nexus view, this case will analyze the implications of different climates, also considering the national energy mix, which can affect the consumed water depending on the different types of electricity generation.

The simulations account for climate variations (specifically dry bulb and wet bulb air temperature) of different climates, cooling load and the dynamic behavior of the heat recovery heat exchanger, the so called WCD, used for EAF's flue gases cooling.

The cooling load derives from the electric transformation cabins used at steelmaking plants, where air conditioning is required in order to avoid breakdowns or abnormal functioning. Cooling load is simulated accounting climate influence, and cabin's building characteristics. In particular three ways for cooling energy production are analyzed:

- standard mechanical vapor compression chiller
- absorption chiller using recovered heat from WCD
- standard mechanical vapor compression chiller coupled with free cooling mode.

The heat rejection required at condensers is provided by water condensers coupled either with either CTs or DCs based on the configuration analyzed. The free cooling mode consists of the direct use of the external air.

For the water-energy nexus indicators evaluation, data comes from the results elaborated in case study *C1a*.

4.2.1 Air conditioning system specification

At electric steel-making sites transformers are required to provide electricity to all the electrical equipment like motors, control rooms, robots and the EAF's electrodes. Transformers usually are located inside designated rooms called electrical cabins where due to Joule's first law heat is released, increasing the operating temperature, and therefore an air conditioning system is required to avoid abnormal functioning or breakdowns.

The cooling system consists of an air-cooling unit located inside the room, typically a fan coil unit, where the thermostat is set to keep the inside temperature under 40 °C, which is considered the safety operation threshold provided by electric equipment manufacturers. The cooling load required by the electrical cabin analyzed in this work is approximately of 1000 kW.

Compared with data centers [126], regulation requirements for electric cabinets at steelmaking sites are substantially less restrictive, as they house robust equipment designed for harsh working environments. Thus, in this study, it is assumed that the temperature control system operates with a set-point temperature of 35±2.5°C.

During the maintenance downtime, the heat load (and therefore the cooling load) is still present for 3 hours after the stop. This because machinery, control rooms, and the other electrical devices need to operate until the last part of the steel casting is solidified and rolled.

4.2.2 Building characteristics

The climate impact on the total cooling load required by the electrical cabins is evaluated based on a series of simulation using TRNSYS©, where the electric cabin building is modelled as well as the conditioning system.

In this analysis, the reference electric cabinet has a building surface area of 3700 m^2 and a volume of 17000 m^3. Electric cabins may be located inside or outside the steelworks depending on plant design. Outside installations are investigated in the present work in order to determine the extent to which the local climate affects cooling load and systems performance.

The thermal transmittance is evaluated based on data provided by cabinet manufacturers at 0.4 $W/m^2{\cdot}K$. The section showing the standard wall used for calculations is shown in Figure 4.6.

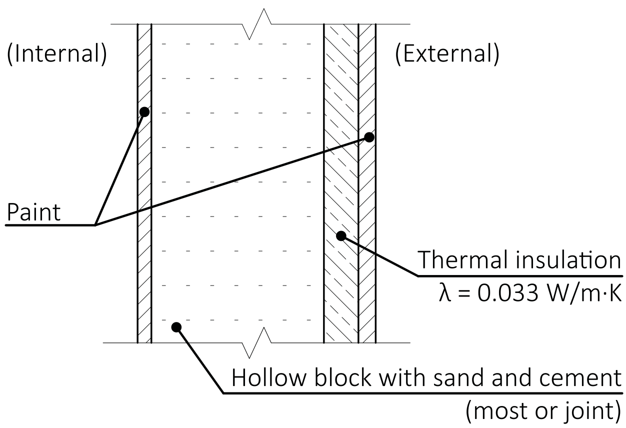


FIGURE 4.6: Electric cabinet wall section.

4.2.3 Configurations for case study C1b

Three alternative cabinet air conditioning configurations are modelled and compared in this study: the baseline mechanical vapor compression chiller (MVC), described in section 4.2.3.1, an energy saving mechanical vapor compression configuration based on air-side free cooling with outside air (FC), presented in section 4.2.3.3, and a waste heat recovery, absorption cooling-based configuration (ABS), as specified in section 4.2.3.2.

In particular, as in [127] and [58], it is proposed to recover waste heat from the hot gas line cooling system of conventional electric arc furnaces based on the plant layout and temperature profiles reported in [58]. In fact, in conventional EAFs, off-gases leaving the furnace and the following dropout box are cooled down to at least 600 °C, as required for the operation of subsequent plant components, by flowing through a modular gas-tight water cooled duct [88, 127], known in the industry as WCD.

In conventional configurations, the water used as a refrigerant in the WCD needs to be cooled down in heat rejection units (either DC or CT). Total removed heat loads vary over time, due to process intermittence, and depending on steelworks capacity, reaching values ranging e.g. between 10 MW and 20 MW for a 130 t nominal tap weight furnace [128]. For the heat recovery system of concern, we have considered the opportunity to derive a water flow from a module of the cooling water circuit corresponding to an average heat flow of about 3100 kW. To obtain a simple and homogenous assessment of the impact of heat rejection units depending on climate, it is assumed that the same technology, either DC or CT, is used both for heat rejection at the WCD and as a condenser for cabinet refrigeration cycles.

4.2.3.1 Air-cooled and water-cooled MVC chiller

Mechanical vapor compression chillers (MVC) are a common type of refrigerator used for air-conditioning purposes. In this study a water-cooled magnetic centrifugal chiller was used as a base case for electrical cabins air conditioning. The nominal capacity installed is 1300 kW and the performances are taken from the York Chiller catalogue [91]. The EER is 6.4 and the ESEER is 9.42 evaluated with an entering/leaving chilled water temperature of 12/7 °C and entering/leaving condenser water temperature of 30/35 °C.

The DCs and CTs are used for condensation purposes. Heat rejection performance depends on the air's dry and wet bulb temperature, as well as the design solution based on the device used. In this work a LU-VE catalogue [52] for DCs was consulted to extrapolate the power required by the fan's electric motor and the flow rate of water and air. The regression curves are evaluated as a function of the cooling power required. The CTs technical data is taken from YWCT's cooling towers catalogue [54].

Figure 4.7 shows the schematic diagram of this configuration. It can be noted the heat rejection of the WCD's cooling water is also required when heat recovery is not performed.

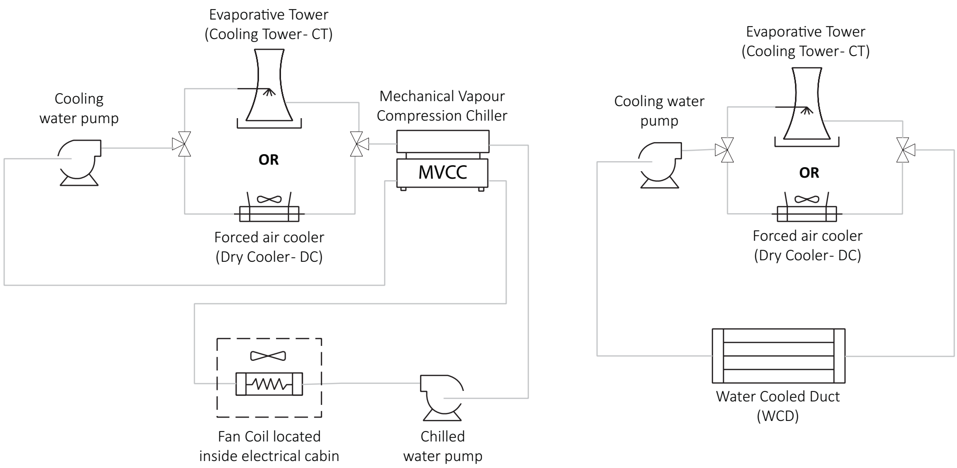


FIGURE 4.7: Mechanical Vapor Compression Chiller configuration schematic diagram.

4.2.3.2 Air-cooler and water-cooled ABS chiller

A waste heat recovery-based cooling system, represented in figure 4.8, relies on the plant configuration proposed in [58], which includes a hot water fed, single effect absorption chiller. In fact, as underlined in [127], in conventional WCDs at EAFs, because the emitted thermal energy is not further utilized, the cooling water outlet temperature is usually in the range of 50 °C [129].

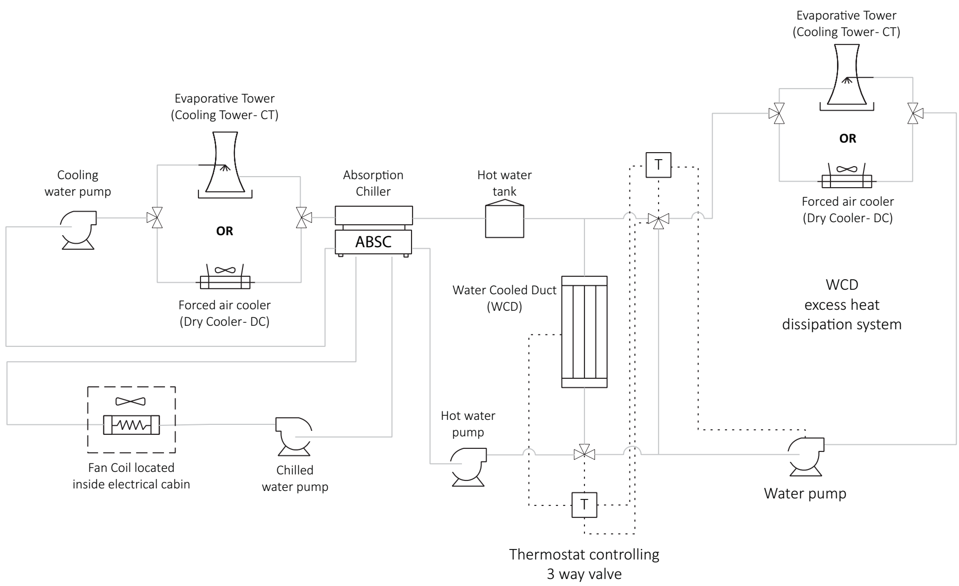


FIGURE 4.8: Absorption Chiller configuration schematic diagram.

If thermal energy recovery is considered, the design temperature of the cooling system has to be increased. While it is also feasible to increase it to 200 °C, as demonstrated in [127], for this absorption cooling application the choice was made to increase it only to the average value of 90 °C. In this way, the system is designed to operate with hot water, in order to avoid introducing additional complexities from steam operation, such as additional maintenance and safety requirements related to higher temperature, pressure and phase change, which would be an additional burden in EAF plants with or without minimal steam networks.

When waste heat is recovered in order to drive absorption cooling machines, only single effect absorption chillers can be used, because hot water is available at an average temperature of 95 °C, associated with oscillations between 85 °C and 105 °C. The reference EER is in the order of 0.7, in accordance with manufacturers' catalogues [91, 130] and literature [90].

A single effect, water-Li-Br, absorption cooling chiller with a nominal capacity of 1319 kW is assumed to be installed at the steelmaking plant. The technical data is collected based on the LG Absorption Chiller catalogue [130].

At EAF steelmaking sites where steam networks exist, an integrated development of heat recovery-based steam generation as in [127] and of absorption-based cooling could be considered in order to exploit more efficient double stage absorption cycles, whose EER is in the order of 1.25 [40]. This is, however, beyond the scope of the present case study.

Given the intermittence of the EAF melting process, based on the aforementioned tap-to-tap cycle, variations in flue gas temperatures correspond to oscillations in cooling water temperature at heat recovery outlet. Thus, as in [127] and [58] a water storage tank is used as a hot water reservoir to compensate for power-off phases by limiting the temperature variability, which for single effect absorption cooling purposes is deemed acceptable in the range of 85 °C to 95 °C. The hot storage size is also designed to meet safety design criteria for cabinet air conditioning systems, which imply that the cooling load to be removed from electric cabins is assumed to be constantly present during steelworks operations and to persist, during maintenance stops, for a period of three hours after the steelworks stop.

Flue gases flow exiting the EAF is intermittent, therefore the power available at the WCD for heat recovery is steelmaking process dependent.

As shown in figure 4.8 thermostats and valves are used to reduce the temperature variability derived from the process. A hot water tank is used as a reservoir when the EAF is not working, and vice versa, a dissipation unit (CT or DC) is supposed to operate when water temperature is greater than 95 °C.

4.2.3.3 Free cooling

The free cooling (FC) configuration analyzed in this study consists of an MVC air conditioning system coupled with an external air ventilation system which draws air from outside and, after filtering, directly introduces it into the cabinet, thereby reducing the cooling load for the conventional MVC chiller. Figure 4.9 shows the schematic diagram of the free cooling configuration.

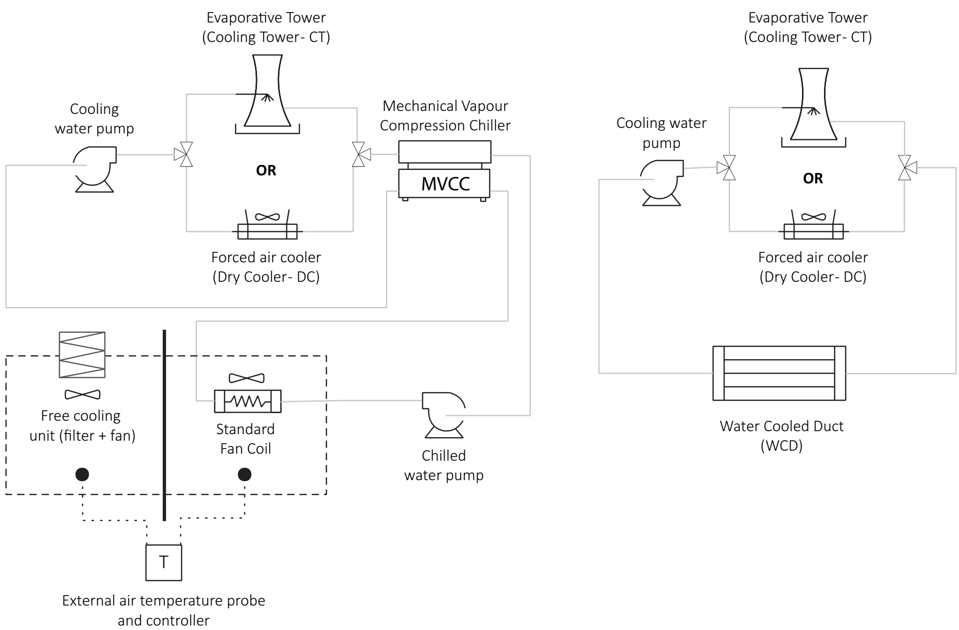


FIGURE 4.9: Mechanical vapor compression chiller with free cooling system schematic diagram. Dashed box indicates electric cabin building.

The control system is based on a thermostat set on a threshold temperature of 12 °C. The external air withdrawal, through the fan activation, is performed when the external air is under the limit temperature of 12 °C. Otherwise MVC is used for cooling purposes.

This solution is preferred in order to reduce the computational load without losing the significance in comparison.

In other applications, such as the aforementioned free cooling for data centers, FC consists of the production of chilled water without the use of a chiller. The plant is designed with a refrigerator bypass and the condenser is still used (and consequently the pumps) as heat sink.

FC configurations are strongly linked to climate temperatures, and as one might expect, cold climates are the major candidates. The use of FC mode is interesting since the energy savings derived could be considerable. MVCs requires electricity during operation, as well as condensers and recirculation pumps. When FC mode is working all the elements of the MVC are switched off, therefore only the fan is consuming energy.

4.2.4 Modelling of the cooling system in TRNSYS

For the dynamic simulation of the configurations presented in this case study a commercial software package, namely TRNSYS, developed at the University of Wisconsin is used.

TRNSYS libraries consist of various components, called *types*, which simulate a huge range of elements used in industrial plants such as chillers, condensers, boilers, turbines and more generally HVAC, electronic, controls and hydronics elements. Types can be linked together to simulate complex systems.

Dynamic simulation of systems is possible by including performance data and simulation parameters for individual elements. The configurations defined in section 4.2.3 are modelled in TRNSYS based on the schematic diagrams shown in Figures 4.7, 4.8 and 4.9 obtaining TRNSYS input files (usually referred to as *decks*). TRNSYS decks for the configuration studied are shown in Appendix C, where:

- The decks shown in Figure C.1 and C.2 are referred to as MVC configurations shown in 4.7 depending on the condenser type.

- The decks shown in Figure C.5 and C.6 are referred to as ABS configurations shown in 4.8 depending on the condenser type.
- The decks shown in Figure C.3 and C.4 are referred to as FC configurations shown in 4.8 depending on the condenser type.

The mass flow rates of chilled water and condenser water required by chillers are taken from manufacturer catalogues. MVC and ABS systems are simulated using technical data (reference chilled, cooling and hot water flow rates) from the manufacturers' catalogues [131] and [130], and the TRNSYS inbuilt performance data file, which allows EER simulation as a function of cooling, chilled and hot water temperatures. The hot water tank is a stratified, 5-layer adiabatic liquid storage tank simulated using TRNSYS *type60*. Cooling water from CTs or DCs serve as an input for the chiller condenser, while the water leaving chiller condenser is used as an input in CTs or DCs depending on the configuration studied. For the simulation of heat rejection units, technical data required as TRNSYS inputs are taken from the LU-VE catalogue [52] and YWCT catalogue [54] for DCs and CTs, respectively.

A weather data file, derived from the EnergyPlus weather database [132], is provided as input to CTs, DCs, and the cabinet building to accurately capture the effect of ambient conditions. Climate zones are selected according to ASHRAE [125]. Table 4.1 shows the cities selected here to represent each climate zone and their climate characteristics. Each of the 3 cooling strategies with 2 different type of condensers are simulated for the 16 selected cities out of 17. Climate zone number 8 is not considered for simulations since cooling towers are inoperable in this zone [133] due to the extremely low temperatures (see Table 4.1). Therefore, a total of 96 simulations are performed.

Additional data, such as performance data files and the complete list of *types* used is reported in Appendix C.

4.2.5 Water consumption calculations

Direct water consumption only occurs in cooling tower configurations. Evaporated quantities are calculated by TRNSYS [134] using *type51b*. The number of transfer units or NTU is calculated from the performance data and the mass flow rates of air and water. Using the NTU and the ambient

TABLE 4.1: Climates zones defined by ASHRAE 90.1 [125] and relative City. Zone number 8 is crossed out since it is not considered in this work.

Climate Zone	City	DRY BULB t. (°C)			WET BULB t. (°C)			RH - %		
		min	MAX	Mean	min	MAX	Mean	min	MAX	Mean
1A	Singapore	21,1	33,8	27,5	16,9	28,2	25,1	44	100	84
1B	New Delhi	5,2	44,3	24,7	4,0	29,5	19,0	9	99	62
2A	Taipei	6,0	38,0	22,8	5,1	29,0	20,3	35	100	81
2B	Cairo	7,0	42,9	21,7	6,0	27,0	15,9	10	100	59
3A	Algiers	-0,8	38,5	17,7	-1,0	27,1	14,6	13	100	75
3B	Tunis	1,3	39,9	18,8	1,2	26,8	15,2	14	100	72
3C	Adelaide	2,0	39,2	16,2	1,2	25,2	11,7	6	100	63
4A	Lyon	-8,5	33,6	11,9	-9,2	26,2	9,4	16	100	76
4B	Seoul	-11,8	32,7	11,9	-13,3	29,6	9,2	9	100	69
4C	Astoria	-3,3	28,3	10,3	-4,7	21,4	8,6	29	100	81
5A	Hamburg	-8,5	32,0	9,0	-9,2	22,8	7,1	26	100	80
5B	Dunhuang	-19,6	39,1	9,8	-20,0	24,3	3,6	4	98	42
5C	Birmingham	-7,4	30,4	9,7	-7,8	20,3	7,7	19	100	78
6A	Moscow	-25,2	30,6	5,5	-25,2	21,7	3,7	28	100	77
6B	Helena	-29,4	36,1	6,8	-29,7	19,1	2,5	11	100	57
7	Ostersund	-25,7	26,5	3,2	-26,1	18,5	1,3	23	100	75
~~8~~	~~Yakutsk~~	-48,3	32,1	-9,1	-48,3	20,0	-11,1	14	100	68

1 – Very Hot, 2 – hot, 3 – warm, 4 – mixed, 5 – cool, 6 – cold, 7 – very cold, A – humid, B – dry, C – marine

conditions, effectiveness of the cooling tower is calculated which is then used to calculate the heat transfer and the outlet conditions [133].

As shown for case study *C1a* with Equation 4.7, additional losses due to bleed off and drift are considered using a multiplicative coefficient k on the total amount of the water consumed. A value of $k = 2$ is taken as a reasonable estimate [58].

4.2.6 Simulations parameters

The simulations consider the electrical cabinet placed outside the steel mill warehouse. Thermal gains are modelled taking into account maintenance downtime as well as scheduled holidays i.e. a control function set to zero all the thermal loads during holidays and maintenance.

The time step used in the simulations is 6 minutes. Start hour is 0 and the stop hour is 8760 (one complete year).

4.2.7 The national energy mix for selected climates

To evaluate the environmental indicators described previously, the national energy mix of every simulated climate (country) is required. Based on the data provided by The Worldbank [100] shown in in Table 4.2, and using results about footprints evaluated on the previous case study, the environmental indicators are assessed.

4.3 Case study C2

It is generally accepted that waste heat recovery from industrial processes is an enabler of energy efficiency and CO_2 emission reduction. Options for industrial waste heat have been presented in the previous section, with the notable exception of improvement and distribution of waste heat through district energy networks to meet a remote heating demand. The case study here examines exactly that in order to get a generalization of the results obtained in *C1*, and in order to consider in detail all the energy conversion technologies presented in this thesis

In this case study the water footprint of district heating and cooling options for recovering low grade industrial waste heat are investigated. Environmental and economic indicators will be evaluated for heat recovery

TABLE 4.2: National energy mix for the analyzed countries (climates) in this study [100].

City	Nation	Biomass & Waste	Solid Fuels	Natural Gas	Geothermal Energy	Hydro-power	Nuclear Energy	Crude Oil	Solar Energy	Wind Energy
Singapore	Singapore	1,39%	0,00%	79,77%	0,00%	0,00%	0,00%	18,82%	0,02%	0,00%
New Delhi	India	0,51%	68,21%	10,35%	0,96%	12,69%	3,02%	1,17%	0,21%	2,88%
Taipei	Taiwan	1,45%	32,65%	10,88%	0,00%	2,40%	16,55%	35,32%	0,01%	0,73%
Cairo	Egypt	0,00%	0,00%	74,59%	0,00%	8,70%	0,00%	15,73%	0,15%	0,83%
Algiers	Algeria	0,00%	0,00%	93,39%	0,00%	1,13%	0,00%	5,48%	0,00%	0,00%
Tunis	Tunisia	0,00%	0,00%	98,19%	0,00%	0,62%	0,00%	0,00%	0,00%	1,18%
Adelaide	Australia	0,98%	69,33%	19,28%	0,00%	5,82%	0,00%	1,41%	0,63%	2,56%
Lyon	France	0,98%	3,99%	3,60%	0,00%	10,89%	76,33%	0,57%	0,84%	2,79%
Seoul	South Korea	0,24%	42,36%	23,03%	0,00%	0,79%	29,03%	4,14%	0,22%	0,18%
Astoria	U.S.A.	1,77%	38,23%	29,77%	0,09%	6,84%	19,04%	0,68%	0,11%	3,48%
Hamburg	Germany	7,68%	46,37%	11,33%	0,00%	3,61%	16,20%	1,53%	4,54%	8,73%
Dunhuang	China	0,95%	74,94%	1,69%	0,00%	18,14%	1,96%	0,16%	0,13%	2,03%
Birmingham	UK	4,23%	40,09%	27,84%	0,14%	1,55%	18,99%	0,99%	0,35%	5,81%
Moscow	Russia	0,30%	15,39%	48,84%	0,00%	16,35%	16,54%	2,57%	0,00%	0,00%
Helena	U.S.A.	1,77%	38,23%	29,77%	0,09%	6,84%	19,04%	0,68%	0,11%	3,48%
Östersund	Sweden	7,15%	1,01%	1,03%	0,00%	48,00%	37,76%	0,65%	0,01%	4,40%
~~Yakutsk~~	~~Russia~~	0,30%	15,39%	48,84%	0,00%	16,35%	16,54%	2,57%	0,00%	0,00%

opportunities, also considering the implications of external and internal energy demand. Configurations analyzed are shown in Figure 4.10 and consist of a significant extension of the case studies proposed in [135].

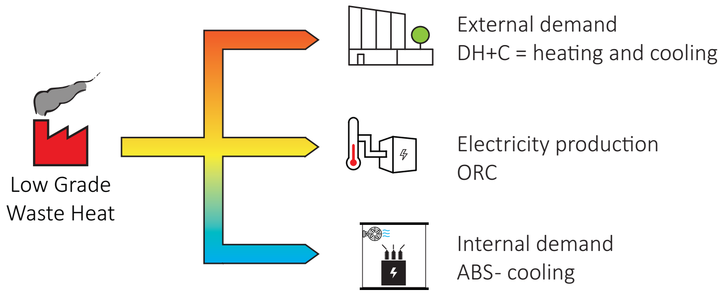


FIGURE 4.10: Low grade waste heat recovery options in case study C2.

The first opportunity is a district heating network recovering waste heat from an industrial site, at the end-user an absorption chiller is supposed to provide cooling energy during the summer. In this case the energy demands are completely external and depend on the end-user simulated.

Alternatives to the district heating are electricity production through an ORC located at the industrial site, or cooling energy production for internal demand using an ABS chiller.

For these alternatives, switching values and trade off curves are calculated and a comparison is drawn between district heating and cooling technologies, and power generation, so as to answer the research question posed as well as to take into account the implications of external and internal energy demand.

4.3.1 Functional units and components

Figure 4.11 presents a schematic representation of the functional units considered for case study C2. The purpose of the configurations shown in Figure 4.10 is to compare different utilization options for a generic low temperature waste heat flow from an industrial process. Thus, the definition of the functional unit is centered on the waste heat flow, and the utilization processes are sized so that the peak demand of the objects to be served matches the available heat flow exactly.

A virtual office building is thus conceived, and assumed to be located at a distance from the waste heat sources that will be changed parametrically. Realistic physical features (shape factors, transmittances) are assigned to this building, denoted as an end-user in figure 4.11. Such features determine the building heating demand profiles over time under different climate conditions as detailed in the energy modelling subsection; however, the building size is varied parametrically in order to adapt it to the maximum available heat flow.

To start the analysis, the reference waste heat flow is evaluated in order to satisfy heating and cooling demand at the end user i.e. at 90 °C for 4900 h/year. It must be clear that the industrial site operates on two shifts allowing the heating supply at the district heating network when the end user is operating.

In the initial (benchmarking) configurations, this waste heat flow is assumed to be completely dissipated. For industrial plants which are not located in the proximity of large surface waters (e.g. the sea) the most common technologies for heat dissipation are forced air cooling, also known as dry cooling (DC in the following), and wet cooling towers (CT in the following). To evaluate the impact of different cooling technologies within the industrial system, three benchmarking scenarios, based on DC or CT, respectively, will be defined. It is also assumed that whenever waste heat is not fully exploited within heat recovery scenarios, residual heat flows continue to be dissipated within the industrial cooling systems that exist prior to waste heat recovery initiatives.

The district heating network structure is simpler than the ones described e.g. by Olivier-Solá et al. [136] or Bartolozzi et al. [137], in that a one-to-one connection between two buildings is assumed and neighborhood networks, connecting multiple end users to the main district heating pipes, are not considered. This serves, however, as a reasonable basis for decision, since, as highlighted in those studies, local distribution networks increase pipeline costs and footprint. Thus, if heat recovery is infeasible for the simplified system presented here, it will be even less reasonable if an equivalent heating demand is shared among multiple, smaller users.

The evaluation is also simplified in that only the main components of

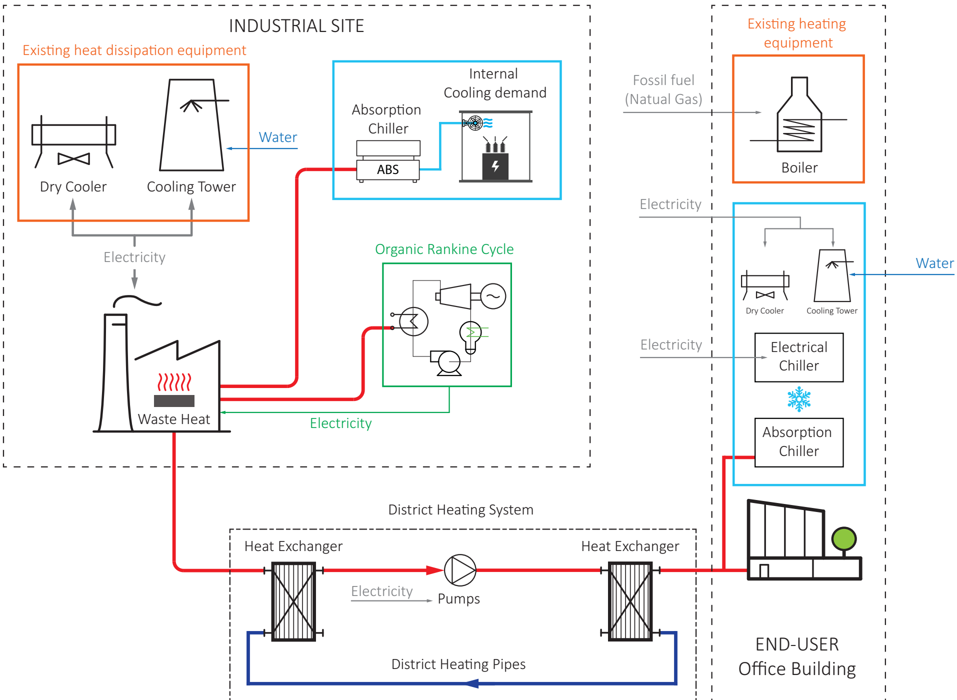


FIGURE 4.11: Schematic diagram of the functional units of study case C2, extension of cases presented in [135].

the district heating system have been considered, i.e. the pipelines, including laying costs and impacts, the pumping system, formed by a set of two pumps in parallel, and heat exchangers at both ends of the network. For the reference case, a network length of 6.6 km was assumed with waste heat flow of 1 MW_{th} at the industrial site.

It is assumed that electricity supply is external and based on the national energy mix both for the "industrial system" and for the "district heating system", whose electricity demand derives from pumps. On the other hand, for the economic and environmental assessment of heat recovery for power generation via ORCs, it assumed that power demand within the industrial system largely exceeds the power flows produced. The electricity obtained from heat recovery is therefore allocated to meet internal demand and substitutes purchases of national electricity.

When heat recovery is not performed, cooling demand is supposed to be supplied with a MVC chiller at both the users. An ABS chiller is supposed to be placed either at the end-user or at the industrial site when heat recovery is exploited.

4.3.2 System boundaries

Direct flows considered within system boundaries include:

- the electricity required by heat dissipation systems, by district heating systems pumps, and by the compressor of the mechanical vapor compression chiller
- the water consumed at CTs dissipation systems
- fossil fuels by boilers at the remote office building
- direct carbon equivalent emissions associated with fossil fuel combustion

For calculating the carbon and water footprints, this research adds up to previous case study *C1a* by also considering the quantities embodied in equipment production.

Only the contribution of additional equipment is considered: the impact of manufacturing boilers, chiller, and original DC/CT equipment is thus neglected, as they are assumed to already exist in the industrial system and in the remote commercial building respectively. It can be noted that this reference system represents a refurbishment situation, which is a more common event than new construction for industrial and commercial areas within the countries of concern. We assume that existing dissipation systems are maintained, and replaced when necessary.

On the other hand, the impact of final disposal stages is out of the system boundaries for footprint calculation in this study. Our steps proceed very much in the same way as Olivier-Solà et al. [136], who observed that "the long durability of [district heating] infrastructure makes it impossible at this time to determine which treatment processes will be followed in the future, which components will be reused." and ultimately to quantify the impact of disposal.

4.3.3 Scenario definition for case study C2

Heat recovery and benchmarking scenarios were defined and named as summarized in 4.3, and the performance of both the industrial system and the virtual building was evaluated as a whole.

Four benchmarking scenarios are defined and evaluated both in Italy and Austria. In benchmarking scenarios, no heat recovery is performed and the whole waste heat flow is dissipated by DCs or CTs. One option to meet the energy requirements of the remote building is considered, i.e. boilers using natural gas (NG) for heat demand and mechanical vapor compression chillers for cooling demand. Heat recovery scenarios are designed for district heating and cooling or, alternatively, either power generation or absorption cooling. When recovered heat is allocated to power generation, no district heating system exists and the remote building continues to use existing boilers and MVC chillers which are replaced twice during the system lifetime.

4.3.4 System modelling and data

To estimate the indirect components of the water and carbon footprints related to equipment and network construction, input and output data for all subsystems have been collected in a life cycle inventory. To estimate direct energy, material and emission flows, technical models of flow balances were developed for heat dissipation, heat losses and pressure drops along the district heat network, including the sizing and operation of the pumping system, ORCs and heating systems at office buildings.

TABLE 4.3: Summary of scenarios and corresponding abbreviations.

Abbreviation	Waste heat dissipation technology	Description
DC/BASE	Dry cooler	No heat recovery, waste heat dissipation with DC, mechanical vapor compression chiller for internal cooling demand using DC, Use of natural gas boilers and mechanical vapor compression with DC chiller at remote building.
DC/DH+C	Dry cooler	Heat recovery for district heating and cooling through absorption chiller at remote building with DC, mechanical vapor compression chiller at the industrial site, dissipation of residual heat and refrigerant condensation with existing DC
DC/ABS	Dry cooler	Heat recovery for absorption cooling on industrial site, dissipation of residual heat and refrigerant condensation with existing DC. Use of boilers and vapor compression chiller with DC at remote building
DC/ORC	Dry cooler	Heat recovery for power generation via ORC, dissipation of residual waste heat and refrigerant condensation at DC, mechanical vapor compression chiller for internal cooling demand. Natural gas boilers and mechanical vapor compression with DC chiller at remote buildings.
CT/BASE	Wet cooling tower	No heat recovery, waste heat dissipation with CT, mechanical vapor compression chiller for internal cooling demand using CT, Use of natural gas boilers and mechanical vapor compression with CT chiller at remote building.
CT/DH+C	Wet cooling tower	Heat recovery for district heating and cooling through absorption chiller at remote building with CT, mechanical vapor compression chiller at the industrial site, dissipation of residual heat and refrigerant condensation with existing CT
CT/ABS	Wet cooling tower	Heat recovery for absorption cooling on industrial site, dissipation of residual heat and refrigerant condensation with existing CT. Use of boilers and vapor compression chiller with CT at remote building
CT/ORC	Wet cooling tower	Heat recovery for power generation via ORC, dissipation of residual waste heat and refrigerant condensation at CT, mechanical vapor compression chiller for internal cooling demand. Natural gas boilers and mechanical vapor compression with CT chiller at remote buildings.

4.3.5 Life cycle inventory

Data was obtained from manufacturers, the scientific literature and the GEMIS database [75]. Table 4.4 presents the life cycle inventory for pipeline manufacturing and laying, including trench works. Data on materials was obtained from manufacturers' catalogues and from the literature [136, 138, 139], and the average value relative to 1 m of network was taken. The carbon and blue water footprint for pipe production and laying are also reported in Table 4.4. Table 4.5 reports the amounts of required materials for equipment production, which were mainly deduced from manufacturers' catalogues. For the sake of parametrization, data and footprints were calculated per kW_{th} of nominal capacity for thermal equipment, per kW_{el} of absorbed power (for pumps) or generated power (for ORCs) for electric equipment.

TABLE 4.4: Materials required and calculated footprints for district heating pipes [135].

Pipe diameter	Materials required							Calculated footprints	
	Pipe			Trench					
DN	Steel	PU	HDPE	Sand	Cement	Concrete	Diesel	$CO2_{eq}$	Water
mm	kg/m	kg/m	kg/m	kg/m	kg/m	kg/m	kg/m	$kg_{CO_{2eq}}$/m	$m^3_{H_2O}$/m
25	1,90	0,46	0,79	194,88	26,88	43,20	0,47	40,58	0,47
32	2,44	0,71	0,98	225,68	29,12	46,80	0,52	45,21	0,52
40	2,80	0,67	0,98	225,68	29,12	46,80	0,52	45,28	0,52
50	3,92	0,83	1,11	250,25	30,80	49,50	0,56	49,09	0,55
65	5,00	0,95	1,25	276,08	32,48	52,20	0,60	52,78	0,59
80	6,46	1,24	1,43	312,48	34,72	55,80	0,66	58,08	0,65
100	9,39	1,92	1,92	392,00	39,20	63,00	0,78	69,39	0,76
125	11,54	2,21	2,29	446,25	42,00	67,50	0,86	76,48	0,83
150	15,48	2,40	2,70	504,00	44,80	72,00	0,94	84,36	0,90
200	22,74	3,62	3,88	670,53	52,08	83,70	1,16	105,85	1,10
250	31,56	6,11	5,78	924,00	61,60	99,00	1,49	137,60	1,39
300	41,98	6,93	7,04	1.092,00	67,20	108,00	1,70	157,68	1,57

TABLE 4.5: Materials required and calculated footprints for equipment [135].

Component	Materials	Mass [kg]	Mass [kg/kW]	Carbon footprint [tCO_{2eq}/kW]	Water footprint [$m^3_{H_2O}$ /kW]
Pump	Stainless Steel	1,51E+01	8,16E-01	4,44E+00	2,04E-02
	Cast Iron	1,36E+02	7,35E+00		
Heat exchanger	Galvanized steel	2,27E+01	9,08E-01	2,14E+01	6,48E-02
	Stainless steel	2,70E+00	1,08E-01		
	Copper	2,16E+01	8,64E-01		
	Foamed polyurethane	2,70E+00	1,08E-01		
ORC	Steel	2,02E+03	4,71E+02	2,56E+02	1,18E+00
Boiler	Cast Iron	1,45E+01	9,60E-01	6,98E+00	6,94E-02
	Steel	1,26E+01	5,80E-01		
	Aluminum	3,18E+00	5,04E-01		
	Brass	2,40E+01	1,27E-01		
Dry Cooler	Stainless steel	8,60E+01	3,58E+00	3,38E+01	4,20E-01
	Aluminum	5,80E+01	2,42E+00		
	Copper	3,50E+01	1,46E+00		
Cooling Tower	Galvanized Steel	1,97E+01	4,10E-01	2,50E+00	1,08E-02
	Fiberglass - reinforced	1,94E+01	4,04E-01		
	Polypropylene	1,15E+01	2,40E-01		
	PVC	2,40E+00	5,00E-02		

4.3.6 Energy models

4.3.6.1 Heating and cooling demand profiles

To account for seasonal variations in heating demand, heating requirement profiles for the building were obtained, simplified as in [140] to express total energy as a function of a peak, an average and a base heating load and the corresponding amount of hours when that load occurs.

Reference sites selected within the research program were Salzburger Seenland for Austria, and Maniago for Italy. Climate data for Salzburg and Aviano Airports were used for calculations.

To get the simplified profiles a realistic model of an office building is used to evaluate the hourly heat and cooling demand over an entire year, based on climate condition for selected towns. The simulated building has a rectangular plan measuring 100 meters x 200 meters and a height from the countryside level of 8 meters (two floors of height 4 meters each). The chosen orientation of the building is in the north-south direction with regard to the short side (100 m).

Heating and cooling loads are evaluated using TRNSYS simulation program, climate data for selected towns is taken from [132]. In addition, to consider the influence of different climates on condensers, EER and SEER are evaluated for MVC and ABS configurations.

Building's HVAC system is supposed to be turned on 2 hours before the opening and turned off at the closing, resulting in a 14 hour/day operation time. The thermostat used for HVAC control requires set temperatures for the cooling and heating modes. During the winter time the internal air temperature is set to more than 20 °C and set to less than 26 °C during the summer.

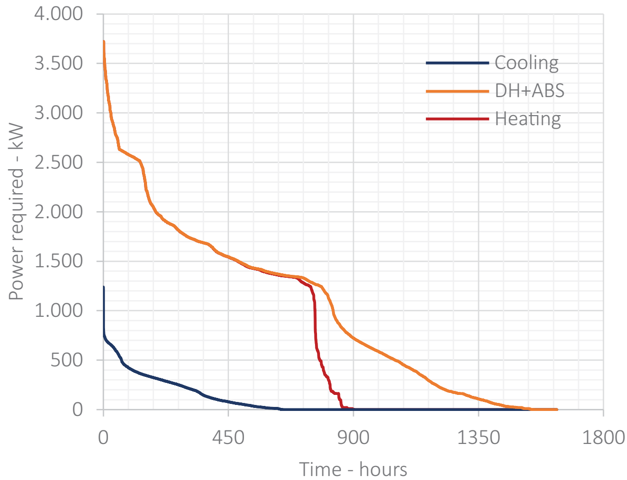
Figure 4.12 shows the simulation results, where power is sorted in descending order. It can be seen that the climate influence increases the heating operation hours in Austria (Figure 4.12(a)), where the red curve is shifted further to the right. Vice versa for warmer climates like Maniago (Figure 4.12(b)), the curve is shifted further towards higher power required for the chiller. The orange curve in both cases should be interpreted as a heating curve, showing the additional heat required by the operation of ABS at the end-user.

4.3.6.2 Discretization and parametrization

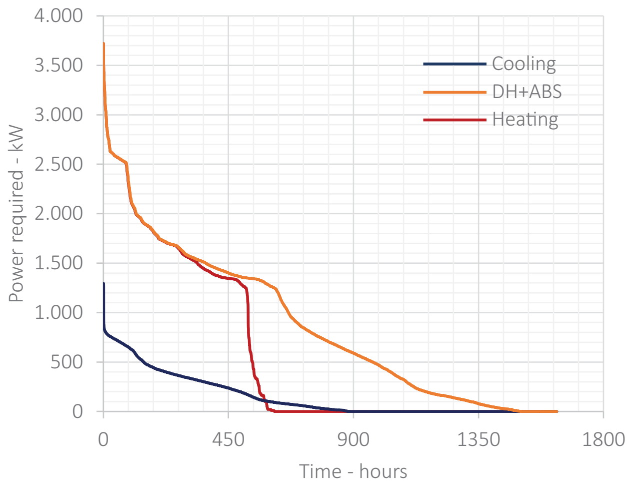
The heating and cooling curves shown in Figure 4.12 are then discretized, so that the total power requested is classified into three levels: 100%, 60% and 20%, and for each level the total number of hours is recalculated preserving the total energy consumption. The resulting discretized duration curves are shown in Figure 4.13 where for the peak load it is assumed to use an additional compensation unit i.e. a boiler (also in the DH cases).

In this way the thermal and cooling load required by the end-user can be parametrized, allowing the simulation of different office buildings depending on the total heat load required. In fact, cooling load and duration curves can be re-evaluated based on the town where the building is supposed to be.

It can be noted that equivalent hours vary depending on the climate zone, in particular Salzburg requires more hours of heating when compared with Maniago, and vice versa, Maniago requires more hours of cooling than Salzburg. The summed curve "DH+ABS" is evaluated by taking into account both the heating demand of the DH network and the heating

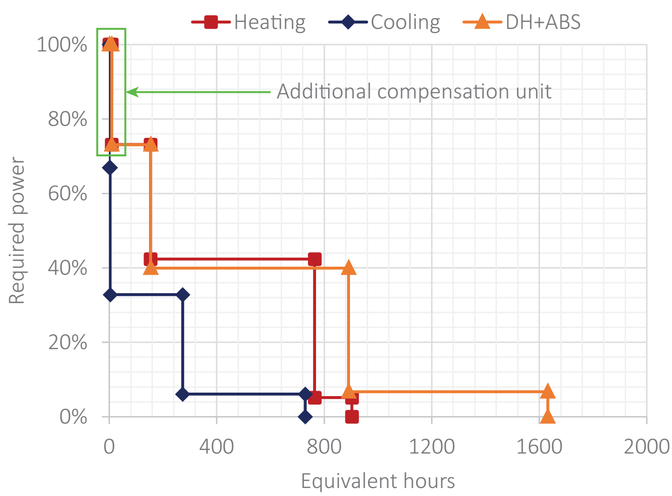


(a)

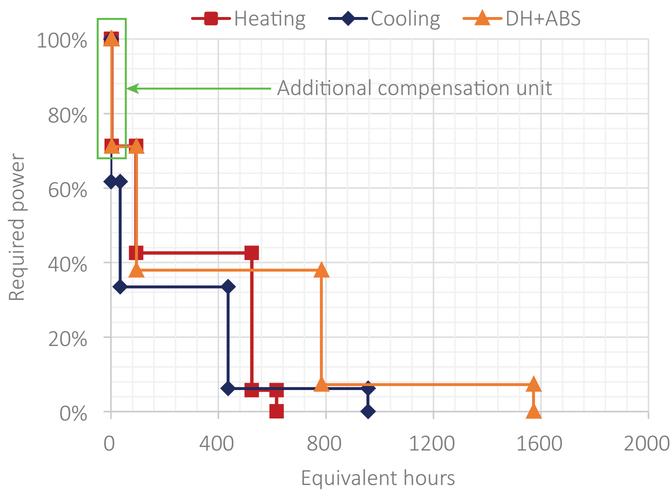


(b)

FIGURE 4.12: Salzburg and Maniago duration curves. Power demand is sorted from high to low values. (a) Salzburg; (b) Maniago.



(a)



(b)

FIGURE 4.13: Salzburg and Maniago discretized duration curves. (a) Salzburg; (b) Maniago.

demand related to the absorption chiller operation, the latter of which is calculated using the EER.

4.3.6.3 Efficiencies and heat losses

In order to vary the peak load of the building in accordance with available waste heat for parametric analysis, it was assumed that the heat produced by the system was exactly equal to the sum of the building load and the waste heat diminished by heat losses in pipes, which were calculated in accordance with [141]. The soil temperature was assumed to be 13 °C for all sites and the temperatures of the supply and return pipes were set at 90 °C and 70 °C, respectively. A thermal conductivity of the soil of 1.5 W/(m K) was used. For the reference case, this results in a heat loss of about 12%. Heat losses at plate heat exchangers were assumed to be negligible as in [142]. To evaluate the energy and environmental performance of benchmarking scenarios, an average seasonal efficiency of 85% was assumed for natural gas, in line with [142].

4.3.6.4 Pumps sizing and operation

For sizing pumps and determining their energy consumption, friction losses in feed and return pipes were calculated with Darcy-Weisbach equations for circular pipes and Nicuradse friction coefficients, and increased by 20% to account for concentrated head losses within the circuit. A minimum pressure of 1.5 bar was assumed to be required at the heat exchanger of the remote building. This results in a total absorbed power of about 26 kW at peak heat load for the reference case.

4.3.6.5 Economic data

For parametric calculation of economic performance, data obtained from manufacturers was used to update and complement cost functions reported in [142]; the resulting cost functions are reported in appendix in Tables A.1 and A.2. Electricity and water costs are derived from case study *C1a*, whereas fuel costs ranges derived from local providers are 73-77 €/MWh_{th} for gas.

An interest rate of 6% was assumed, which was recently used for district heating systems within a similar context [140].

4.3.6.6 Reference case and plant lifetime

The waste heat available at the industrial site is 1 MW_{th} available for heat recovery, the network district heating network is 6.6 km length.

To be conservative, a time horizon of 30 years has been assumed for economic and footprint assessment, reported as nominal lifetime of district heating pipelines by [143]. A lifetime of 15 years has been assumed for pumps and other equipment, and the investment for their replacement at end of life was also considered.

Chapter 5

Results and discussion

This chapter presents the main results obtained for each of the case studies presented. The analysis has been differentiated according to the energy demand context considered.

5.1 Results of case study C1a

The results of case study *C1a* refer to the application of low grade waste heat for *internal* cooling demand i.e. cooling electric cabins considering the average performance of dissipation units (e.g. condensers or heat rejection devices) and a constant low grade waste heat supply and energy demand. The section is structured as follows: the first part is related to the water-energy GHG input data analysis, showing footprints for water, CO_2, and primary energy, based on the source used for electricity production as well as prices for water and electricity over EU-15 countries. The second part is related to the economic analysis of configurations *ii* and *iii*. Then, a sensitivity analysis on carbon tax is given in order to assess configuration changes derived from the effects of CO_2 constraints on the absorption cooling and ORC configurations.

5.1.1 Water, energy and GHG input data

Distributions used for water-energy-carbon related factors associated with electricity purchase are reported in Tables 5.1, 5.2, and 5.3. Table shows the primary source used for electricity generation, the distribution type used in the Monte Carlo simulation and the relative characteristic values. Data sources are also given.

It should be observed that water consumption values (Table 5.1) are the most uncertain, especially for hydropower, and even though a number of data sources exist, data fitting invariably leads to uniform distributions, i.e. every value within the usually wide range is equally probable. Table 5.1 reports the extreme values of these ranges for the distributions of water consumption factors per MWh_e per energy source.

TABLE 5.1: Consumed water per $MW{\cdot}h_e$ generated [58].

Primary source	Distribution type	Characteristic Values $l/MW{\cdot}h_e$		Data sources
		min	max	
Nuclear Energy	Uniform	1677	2900	[15, 75, 113–117]
Solid Fuels	Uniform	1336	2600	[75, 113, 115–117]
Natural Gas	Uniform	687	1400	[75, 113, 115–117]
Crude Oil	Uniform	971	1697	[75, 114, 116]
Solar Energy	Uniform	7	4700	[113–117]
Biomass & Waste	Uniform	1145	1853	[75, 113, 114]
Geothermal Energy	Uniform	5824	9033	[113, 114]
Hydropower	Uniform	5394	68137	[75, 113, 114]
Wind Energy	Uniform	0	4	[15, 113, 114, 116, 117]

For the CO_2 equivalent emissions per GWh_e reported in Table 5.2, the best fit for data was obtained with triangular distributions when more data was available (e.g. for solid fuels, natural gas, solar energy, hydropower and wind energy), and it was also possible to develop subjective triangular distributions for remaining energy sources. For triangular distributions, minimum, maximum and most likely values are reported in all tables.

For primary energy factors per kWh_e, uniform distributions, whose range extremes are reported in Table 5.3, were usually the best fit. They have been also subjectively applied to solar energy, for which a single data source [75] was available.

TABLE 5.2: Carbon dioxide emitted per GW·h_e generated [58].

Primary source	Distribution type	Characteristic Values $tCO_2/GW{\cdot}h_e$			Data sources
		min	ML	MAX	
Nuclear Energy	Triangular	16.0	23.2	30.0	[75, 111, 112]
Solid Fuels	Triangular	905.7	1001	987.6	[75, 110–112]
Natural Gas	Triangular	353.6	481.4	563.2	[75, 110–112]
Crude Oil	Triangular	677.7	742.1	875.0	[75, 110, 112]
Solar Energy	Triangular	35.0	49.7	130.0	[75, 110–112]
Biomass & Waste	Triangular	18.0	34.5	51.0	[75, 111]
Geothermal Energy	Triangular	15.0	45.0	104.0	[75, 111, 112]
Hydropower	Triangular	4.0	17.6	40.0	[75, 110–112]
Wind Energy	Triangular	7.0	17.0	29.5	[75, 110–112]

* ML = most likely.

TABLE 5.3: Consumed Primary Energy per kW·h_e generated [58].

Primary source	Distribution type	Characteristic Values			Data sources
		min	MAX	ML	
Nuclear Energy	Uniform	3.07	3.50		[75, 108, 109]
Solid Fuels	Uniform	2.98	3.26		[75, 108]
Natural Gas	Uniform	2.02	2.63		[75, 108]
Crude Oil	Uniform	2.76	3.40		[75, 108]
Solar Energy	Uniform	0.50	1.00		[75, 108]
Biomass & Waste	Uniform	0.20	4.53		[75, 108]
Geothermal Energy	Triangular	0.40	6.16	4.24	[75, 108, 109]
Hydropower	Uniform	0.06	1.15		[75, 108]
Wind Energy	Uniform	0.03	1.00		[75, 108]

* ML = most likely.

5.1.2 Calculation of water-energy-GHG nexus indicators for electricity generation in the EU-15

As a first step, the model developed in 4.1 is used to estimate carbon and blue water footprint and primary energy consumption indicators for electricity production in the EU-15 countries, based on the energy mix as of year 2012 [100, 101]. The expected values for their distributions are summarized in Table 5.4.

TABLE 5.4: Expected values of carbon, water and primary energy indicators for electricity production and relevant correlation coefficients [58].

Country	Simulated CO_{2eq} t/MWh$_e$	Simulated H_2O consumption l/MWh$_e$	Simulated Primary Energy TOE/GWh$_e$
Austria	0.18	24972	110
Belgium	0.22	1800	240
Denmark	0.42	894	175
Germany	0.52	2575	224
Ireland	0.49	2152	189
Greece	0.66	3879	216
Spain	0.34	3767	191
France	0.08	5765	246
Italy	0.43	6488	180
Luxembourg	0.29	12366	144
Netherlands	0.53	1273	217
Portugal	0.42	5568	170
Finland	0.12	3839	244
Sweden	0.03	18786	157
United Kingdom	0.53	1942	234

The validation of the results and error analysis was performed by comparison with data sources available in literature. Quantitative or qualitative methods were used, as illustrated in Table 5.5, depending on the availability of data.

TABLE 5.5: Validation of water-energy-GHG nexus indicators [58].

Indicator	Reference for validation and geographical data scope	Validation method and parameters	Results
CO_{2eq} emission factor [t/MWh_e]	[144]	Quantitative, calculation of MAPE and MPE	MAPE = 17.4% MPE = -7.0%
Primary energy consumption [TOE/GWh_e]	[109], graphical data for a subset of EU-15 (missing countries: Austria, Belgium, Luxembourg and UK)	Qualitative, by comparison of country rankings	Country ranking is similar, exceptions are Finland and the Netherlands, which have lower primary energy consumption according to [109]
Water consumption for electricity generation [l/MWh_e]	[15], European average	Qualitative, comparison of simulations of single countries with EU average	EU average according to [15] = 11660 l/MWh. Values calculated in this study are above average by [15] only in Austria, Luxembourg and Sweden

For CO_2 equivalent emissions data are available and officially reported by the European Environment Agency [144], thus differences between indicators reported in Ref. [144] and expected values calculated for the countries in question could be calculated, as well as the mean absolute percentage error (MAPE) and the mean percentage error (MPE). Based on outcomes reported in Table 5.5, the model estimates tend to be lower than values reported in Ref. [144], probably because our estimates are based on the energy mix of year 2012, while data available from Ref. [144] refers to the year 2009.

For primary energy consumption, validation is qualitative, in that most data in Ref. [109] are only graphically represented and only available for a subset of the EU-15. Ordering countries by decreasing values of primary energy consumption per GWh leads to approximately the same country ranking represented in Ref. [109], with the notable exception of the Netherlands and Finland. According to [109], those countries ranked more toward the lower end of the range, while their primary energy consumption seems to be overestimated by the present simulation. For the Netherlands, this may be due to decreasing efficiency or increasing use of fossil fuels over time. In fact, 2009 $CO_{2,eq}$ emission factors according to [145] were also significantly (22%) lower than model estimates for 2012. On the other hand, Finland consistently has very low carbon emissions and is found to have the highest share of biomass and waste used for power generation in the EU-15 [100, 101]. Such a high share of bioenergy may be the cause of the model's overestimation of the primary energy consumption for this country. In fact, comparing data sources [75, 108], the expected values of primary energy demand for bioenergy calculated by the model are relatively high, because the upper limits of the range, obtained from Ref. [112], are particularly high. A wider set of data, especially from a European context, would improve the model accuracy in evaluating the impact of bioenergy for the countries of concern.

Apart from the mentioned exceptions, primary energy consumption indicators calculated with the model for 2012 are generally lower than values reported in Ref. [109] for the year 2009. A trend towards increased efficiency, previously highlighted in Ref. [109], is thus confirmed in the present work.

To validate water consumption footprints, only the continental average data reported by Ref. [15] could be used, whose estimates for the average water comsumption footprints are considerably higher than our estimates for individual countries. A possible cause is that the upper bound of the consumption water footprints assumed by Ref. [15] for bioenergy is significantly higher than ours. A subsequent validation of these bottom up models with hybrid approaches based on input output models [12] could be the subject of future research, but is out of the scope of this study.

Since model estimates for the three nexus indicators are based on the same dataset, some significant statistic correlation can be expected. Calculated correlation coefficients are reported in Table 8.

TABLE 5.6: Correlation coefficients between indicators [58].

Indicator	CO_{2eq} emission factor [t/MWh$_e$]	Primary energy consumption [TOE/GWh$_e$]	Water consumption for electricity generation [l/MWh$_e$]
CO_{2eq} emission factor [t/MWh$_e$]	1	0.153	-0.55
Primary energy consumption [TOE/GWh$_e$]	0.153	1	-0.76
Water consumption for electricity generation [l/MWh$_e$]	-0.55	-0.76	1

* significant at 5% significance level.

A significant negative correlation is found between water consumption and CO_2 equivalent emissions and between water consumption and primary energy consumption. Hence, countries with higher carbon and primary energy indicators usually have significantly lower water consumption indicators, and vice versa. This is mainly due to the role of hydropower in energy systems, which is associated with low life cycle carbon equivalent emission factors (Table 5.2) and primary energy consumption (Table 5.3), but has the highest freshwater consumption footprints (Table 5.1) mainly due to evaporation from hydropower reservoirs [15]. The result is in line with similar findings recently reported in the literature [146].

The statistical correlation between CO_2 equivalent emissions and primary energy consumption is not significant. This result may appear counterintuitive, but it is justified by the effect of nuclear energy, which has minimum carbon impact but high primary energy consumption factors.

5.1.3 Economic feasibility of the project with average EU-15 conditions

The economic feasibility of the project has been evaluated first in absence of carbon related obligations or incentives, i.e. at a null carbon price, with *average* EU-15 conditions as for water and electricity prices. The investment analysis is performed on a differential analysis basis, by considering the differences between the required investment and resulting cash flows of proposed heat recovery and energy conversion alternatives and the reference base case with full dissipation through dry cooling. The investment indicators considered are equivalent annual costs, and simple payback times, presented in Fig. 5.1 and Fig. 5.2, respectively, through box and whisker diagrams highlighting medians, quartiles and extreme values. In this case, the interest rate is fixed at 7% and investment duration at 10 years, while the sensitivity of project profitability to annual operation hours is tested by varying this parameter between 2500 h/year and 7000 h/year.

The box and whisker diagrams presented in Fig. 5.1, Fig. 5.2 highlight that at 7000 h/year, median equivalent costs of all heat recovery projects are lower than base case medians, and simple payback time medians are lower than investment duration. Absorption cooling (configuration *ii*) alternatives, however, pay off in about one year, with minimum variance in case dry cooling systems are used. The payback of ORC projects (configuration *iii*) is much longer and has the highest level of uncertainty. With 2500 operating hours per year, power generation projects are not feasible without incentives or carbon prices, while median costs of absorption cooling systems remain below base case costs both for DC and CT alternatives.

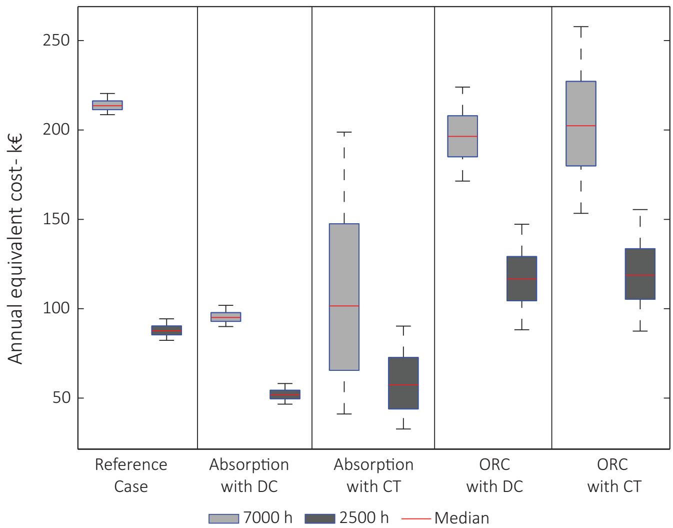


FIGURE 5.1: Equivalent annual costs of systems at average EU-15 conditions [58].

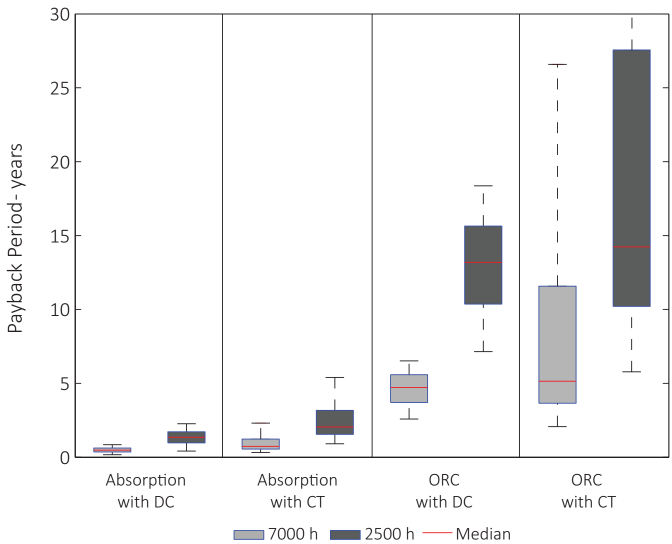


FIGURE 5.2: Pay back times at average EU-15 conditions [58].

With the conditions considered, median payback times are also acceptable for all absorption cooling variants with a 2500 h/year operating time. However, solutions with CT have higher uncertainty, with the widest spans between maximum and minimum values and quartiles, especially if the operation time increases.

For all heat recovery options, as we are considering an average distribution of power and water prices over the EU-15, the introduction of wet cooling systems as a complement to heat recovery leads to lower profitability and higher uncertainty, because of considerable variations in industrial water prices among EU-15 countries. In general, we would expect that without carbon incentives, projects including dry cooling would be preferred. Looking at national outcomes, however, results would be different, especially for Italy and other countries, as discussed in the following sections.

5.1.4 Nexus impact of carbon reduction policies in different EU-15 countries

Policies aimed at carbon emission reduction have been debated in Europe since the early nineties, and a variety of instruments have been proposed, including voluntary agreements, unilateral programs and multilateral programs [147], until the emission trading scheme for carbon-intensive companies was launched in 2005. Research confirms that allowance prices are now integrated into several aspects of corporate decision making, although technological changes induced by the EU ETS are moderate, in that the industry prefers small scale projects with short term horizons rather than large scale projects with higher returns, but higher risks [148].

A sensitivity analysis was performed by varying carbon prices between zero (corresponding to the situation that firms are not subject to the EU ETS or other unilateral carbon taxes) and 120€/tCO_2 equivalent emissions. The current market value of ETS allowances is around 8€/tCO_2 [149]. While an increase to 20€/tCO_2 is expected in the next few years, the upper boundary of the proposed range may seem extremely high. However, renewable energy and other forms of investment in energy efficiency are subsidized with other instruments in some countries (e.g. white certificates or renewable energy feed in tariffs in Italy) whose cost, related to carbon equivalent reduction, is comparable with these ranges.

Assuming that companies invariably choose the technology with the lowest annual equivalent costs, including costs from carbon allowances, the aim of this sensitivity analysis is to determine what carbon prices may induce:

> a technology switch from configuration (*i*) to configurations (*ii*) or (*iii*), respectively; - a switch from one cooling system technology for residual heat dissipation to another.

Dry cooling is more energy intensive, and thus more carbon intensive than wet cooling, which requires high water supplies, with corresponding costs. Hence, we test whether and where higher carbon prices may lead to a technology switch from energy intensive dry cooling to water intensive wet cooling.

For each analysis, the impact of technology switches on the expected values of water consumption, CO_2 equivalent emissions and primary energy consumption of the project is displayed, in Fig. 5.3, Fig. 5.4, Fig. 5.5 for configuration (*ii*), i.e. absorption cooling, and in Fig. 5.6, Fig. 5.7, Fig. 5.8 for configuration (*iii*), i.e. ORC for power generation, respectively.

5.1.5 Absorption cooling

Based on the model results, the heat recovery project for absorption cooling would be feasible in every country, even where electricity prices are lowest, mainly due to the long operation time associated with the case study of concern.

In Fig. 5.3, Fig. 5.4, Fig. 5.5, the resource efficiency indicators of the projects are plotted at zero carbon price and compared with the indicators for the base case (i), represented as white bars. Water footprints are represented in Fig. 5.3, $CO_{2,eq}$ emissions in Fig. 5.4 and primary energy consumption in Fig. 5.5.

Fig. 5.3, Fig. 5.4, Fig. 5.5 also highlight the cooling system type selected at zero carbon price, which in all cases remains unchanged even at the current market price of about 8 €/$tCO_{2,eq}$, as well as carbon prices inducing some technology switch. For most countries, the situation remains unchanged even with growing carbon prices, but in four countries, namely the United Kingdom, Germany, Austria and the Netherlands, marked with continuous arrows in Fig. 5.4, Fig. 5.5, a switch from dry cooling to wet cooling systems for heat dissipation happens at the threshold values reported above the bars, i.e. at 20 €/t for the UK, 50 €/t for Germany and 80 €/t for Austria and the Netherlands.

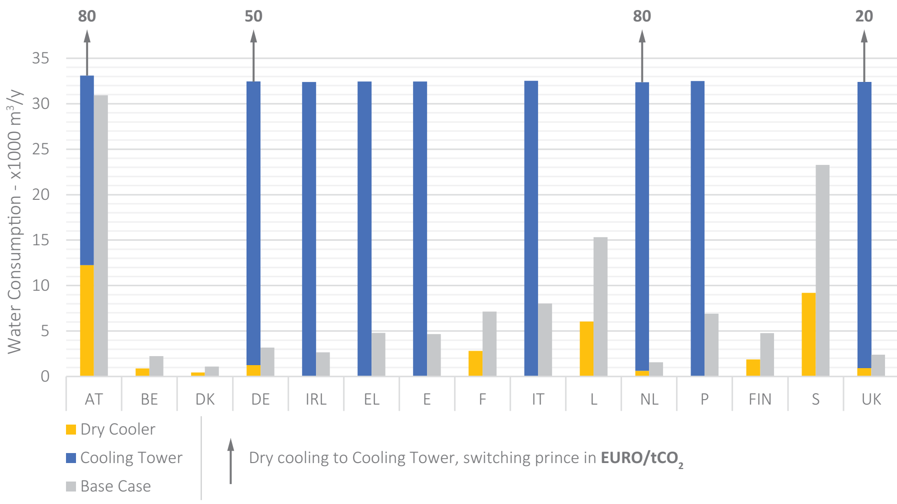


FIGURE 5.3: Water consumption for absorption cooling project in EU-15 at zero and current carbon price and at carbon prices determining cooling systems switch (labels above the bars) [58].

Wet cooling systems would be selected even at zero or current carbon costs in Italy, Portugal, Spain, Greece and Ireland. According to our estimates, these countries have the lowest water prices, with expected tariffs ranging from less than 90 €¢/m^3 in Greece to almost 140 €¢/m^3 in Portugal. With the exception of Ireland, these are Mediterranean countries, exposed to the highest risk of water scarcity. As shown in Fig. 5.3, for configuration (*ii*) wet cooling entails a total water consumption of about 33000 m^3/year, including the generally low indirect water consumption, which is a function of the electricity demand for auxiliaries, and ranges from approximately 40 m^3/year in the Netherlands to almost 770 m^3/year in Austria.

Independent of carbon price, dry cooling is the technology option chosen in Belgium, Denmark, France, Luxembourg, Finland and Sweden. Belgium, Denmark and Luxembourg have the highest water prices in Europe,

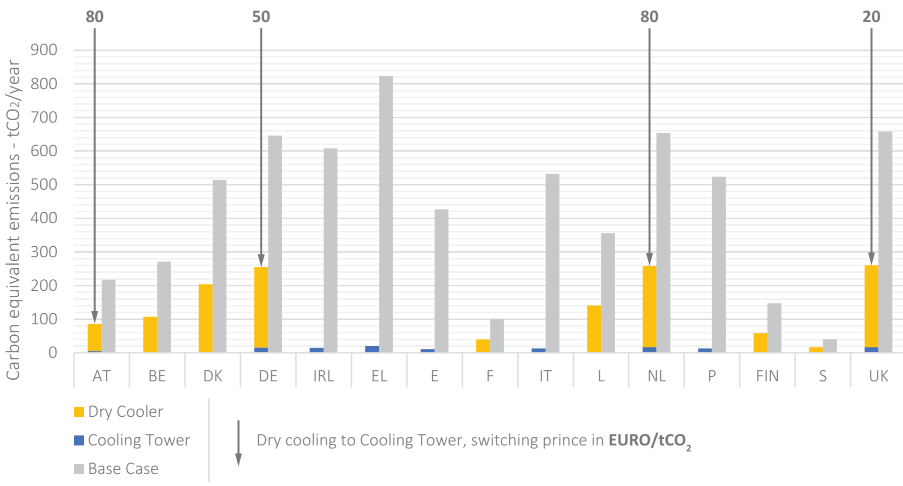


FIGURE 5.4: Primary energy consumption for absorption cooling project in EU-15 at zero and current carbon prices and at carbon prices determining technology switch (labels above the bars) [58].

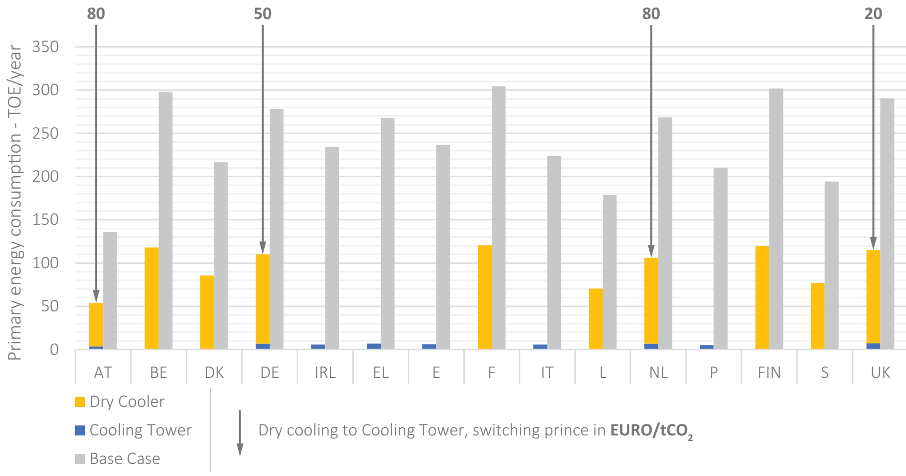


FIGURE 5.5: CO_2 equivalent emissions for absorption cooling project in EU-15 at zero and current carbon price and at carbon prices determining technology switch (labels above the bars) [58].

while in France, Finland and Sweden water prices are intermediate, but industrial electricity prices and CO_2 equivalent emissions for electricity generation are the lowest in Europe.

When dry cooling is chosen, the configuration (*ii*) of the heat recovery project invariably leads to an improvement of water footprint indicators, more evident in the countries with the highest shares of hydropower in their national energy mix.

Compared with the wet cooling option, the additional power demand for dry cooling is about 450 MW·h/year, which implies additional carbon emissions and primary energy consumption of variable size, depending on country energy mix. Where dry cooling is preferred for the project, total CO_2 equivalent emissions range between 16 t/year in Sweden and 203 t/year in Denmark (Fig. 5.4) and primary energy consumption ranges from 54 TOE/year in Austria to 120 TOE/year in France and Finland (Fig. 5.5).

Net benefits deriving from the heat recovery project in configuration (*ii*) are always high even with dry cooling, both in terms of CO_2 emissions and primary energy consumption. When wet cooling is preferred, however, the increase in direct water consumption is never offset by the decrease in indirect water consumption associated with lower electricity consumption, even in the countries with the highest water footprints for electricity production, such as Austria.

5.1.6 Power generation with ORC

Based on the model results, the heat recovery project for electricity generation would be feasible in most countries, with the exception of Finland, Sweden and France, which are characterized by the lowest electricity tariffs. Water footprints are reported in Fig. 5.6, CO_2 equivalent emissions in Fig. 10 and primary energy consumption is represented in Fig. 11. Each figure shows the values of indicators for optimal solutions at a null carbon price, which remain unchanged even at current market conditions, as well as the levels of carbon price inducing a technology switch.

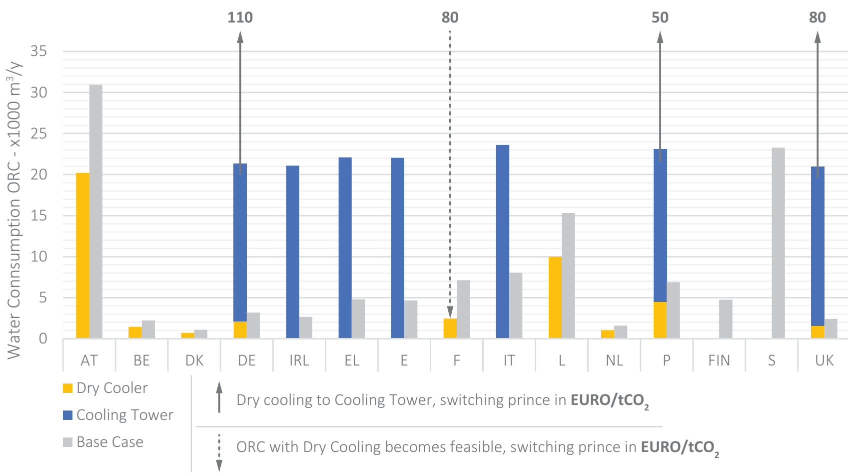


FIGURE 5.6: Water consumption for ORC project in EU-15 at zero and current carbon price and at carbon prices determining technology switch (labels above the bars) [58].

In France, the project configuration (*iii*) with dry cooling becomes feasible at a carbon price of 80 $t/CO_{2,eq}$ (dotted arrows in Fig. 5.6, Fig. 5.7, Fig. 5.8), in spite of the relatively modest reduction in carbon emissions.

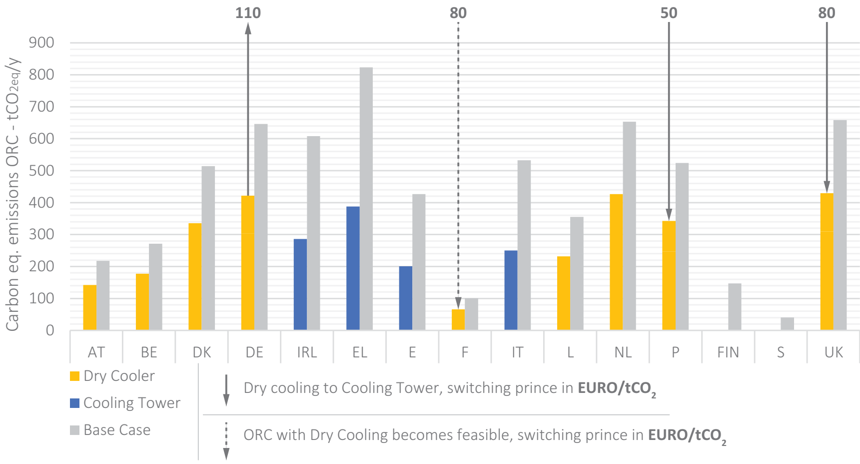


FIGURE 5.7: CO_2 equivalent emissions for ORC project in EU-15 at zero and current carbon price and at carbon prices determining technology switch (labels above the bars) [58].

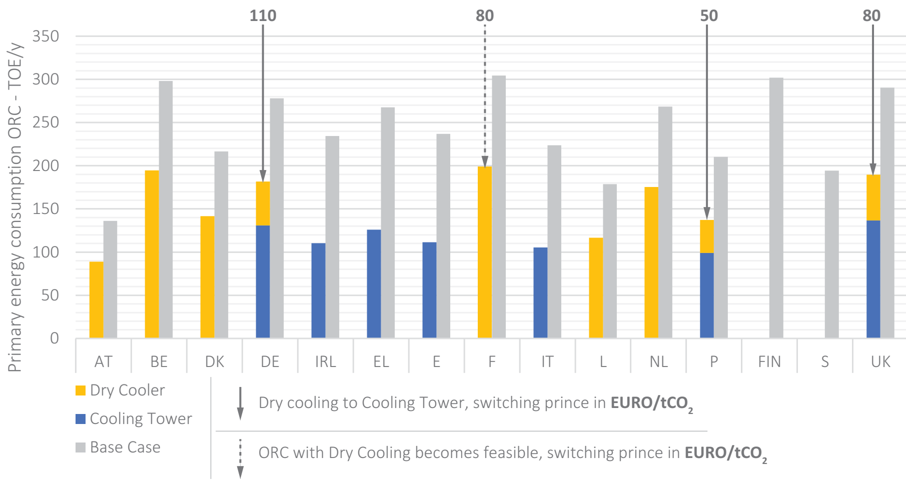


FIGURE 5.8: Primary energy consumption for ORC project in EU-15 at null carbon prices and at carbon prices determining technology switch (labels above the bars) [58].

The introduction of waste-heat-to-power technology in France is associated with a more evident decrease in primary energy consumption and indirect water demand, which is a result of the high nuclear power share in the electricity generation mix. As dry cooling is preferred in France, waste heat recovery is also associated with lower water consumption than the base case (dotted arrows in Fig. 5.6, Fig. 5.7, Fig. 5.8). The opposite occurs when higher carbon prices induce a switch in cooling systems technologies (continuous arrows in Fig. 5.6, Fig. 5.7, Fig. 5.8).

Like in configuration (*ii*), in Italy, Spain, Ireland and Greece wet cooling is the preferred option, which is associated with a somewhat lower water demand than in configuration (*ii*) (see Fig. 5.6). This is mainly due to the fact that wet cooling is only used to dissipate waste heat from the ORC condenser and residual waste heat from the hot water circuit, while dry cooling is maintained for dissipation at the condenser of low temperature vapor compression chillers. For ORC, the ratio between heat loads at cooling systems and the net electricity consumption avoided through waste heat recovery would also be less favorable than for absorption cooling.

In Portugal, dry cooling is the preferred option at zero and current carbon price levels, while a switch towards wet cooling happens at 50 €/t$CO_{2,eq}$. Switch prices are higher than in configuration (*ii*) because the residual heat dissipation capacity required in configuration (iii) is smaller, and so are the reductions in carbon emissions (Fig. 5.7) and primary energy consumption (Fig. 5.8) associated with changing cooling systems technologies. The switch from dry to wet cooling systems in Germany, for instance, would result in an expected reduction in $CO_{2,eq}$ emissions of 120 t/year and a reduction in primary energy consumption of about 51 TOE/year. Corresponding reductions for configuration (*ii*) in Germany would be almost 240 t$CO_{2,eq}$/year and about 103 TOE/year, respectively. The shift towards water intensive technologies at higher carbon prices is thus limited to Germany and the UK.

5.2 Results of case study C1b

Case study *C1b* consists of an in depth analysis of the cooling strategies analyzed in case study *C1a* while dynamically simulating air conditioning systems depending on the climate zone.

First the effects of climates on cabin cooling load required are shown, then, before the analysis of the nexus indicators for the configuration simulated, the footprints of the national energy mix are shown.

5.2.1 Cabin cooling load on various climates

The Figure 5.9 shows the cooling load of the electrical cabin on the climate zones defined in [125]. The cabin load reported is averaged over the operating hours of one year. Based on simulations performed, the climate does not substantially affect the cabin load even though heat transfer through the cabinet envelope leads to lower cooling loads. In fact the thermal load difference between zone 1A (hot) and zone 7 (cold) is 27 kW_{th}, less than 3% (2.75%) and therefore negligible.

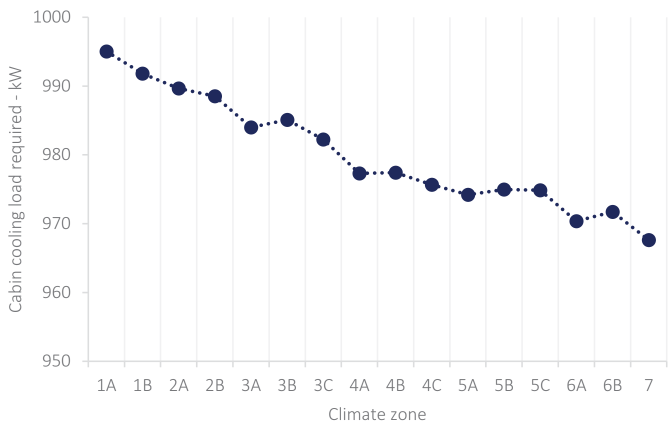


FIGURE 5.9: Simulated annual cabin load for 16 ASHRAE climate zones [125].

The other device performances which could be affected by different climates are condensers/heat rejection units, and therefore the chillers involved in the air conditioning system. Condensation units are located outside the steelworks warehouse, and these results will be presented later.

5.2.2 National energy mix footprints

Using the data results in shown in Section 5.1.2 and the national electricity mix available at [100], it is possible to estimate the nexus indicators for analyzed climates; results are shown in Table 5.7.

Within the frame of an EU-ETS scheme, the evaluation of the national electricity mix in non-European countries could be seen as not necessary due to stark differences in terms of industrialization and prevention of the air pollution. However this analysis can highlight whether the national energy mix also affects substantially the footprint analysis in favorable climate zones.

TABLE 5.7: Estimate of the environmental indicators for ASHRAE climates [125] and considering National electric mix available at [100].

Climate Zone	City	Nation	CO_2	H_2O	TOE
			tCO_2/GWh_e	l/MWh_e	TOE/GWh_e
1A	Singapore	Singapore	524,17	1104,78	212,15
1B	New Delhi	India	745,39	6284,49	227,85
2A	Taipei	Taiwan	646,26	2509,82	254,18
2B	Cairo	Egypt	477,57	4189,21	195,77
3A	Algiers	Algeria	490,45	1462,87	201,80
3B	Tunis	Tunisia	473,01	1253,74	197,15
3C	Adelaide	Australia	799,30	3754,36	234,80
4A	Lyon	France	82,33	5909,40	244,47
4B	Seoul	South Korea	572,73	2093,88	253,77
4C	Astoria	U.S.A.	537,91	4057,07	226,74
5A	Hamburg	Germany	540,83	2972,66	221,17
5B	Dunhuang	China	763,81	8225,79	222,72
5C	Birmingham	UK	550,03	2180,49	232,19
6A	Moscow	Russia	415,10	7242,93	201,58
6B	Helena	U.S.A.	537,91	4057,07	226,74
7	Östersund	Sweden	40,31	18657,32	154,60
~~8~~	~~Yakutsk~~	~~Russia~~	~~415,10~~	~~7242,93~~	~~201,58~~

It can be seen from Table 5.7 that climate zone 7, (Östersund - Sweden), has the lowest CO_2 footprint, in contrast with the water footprint which is particularly intensive.

5.2.3 Electric energy consumption

The electric energy consumption, depending on the configuration studied, is shown in Figure 5.10. Results are plotted in two rows based on the condenser used; the upper row shows configurations using DC, while the lower row shows configurations using CTs. The climate zone and configurations are reported in abscissa where the same pattern is used: base case configuration (BASE), free cooling configuration (FC), and absorption chiller configuration (ABS).

The legend shown in the middle summarizes the energy consumption into three types based on the elements consuming the energy. Auxiliary, e.g. recirculation pumps or fans, are indicated using the abbreviation AUX. Refrigeration energy consumption, including the condensation unit, is referred to as REF. The rejection of excess heat at the water cooled duct is labelled WCD. The heat rejection units used by the air conditioning system and the WCD cooling system are the same.

The BASE case is the worst configuration in terms of electricity consumption, the MVC chiller plays a major role this case. Switching from DCs to CTs can lead to a 22% reduction of the energy consumption, but if a comparison with the same condenser is performed the configuration remains the most intensive. An energy consumption reduction depending on the climate zone can be seen until climate *4A*, after this zone energy reduction is no longer evident.

When CTs are used the energy consumption reduction is even less evident as shown in the second part of Figure 5.10, where starting from climate zone *3C*, the energy reduction is no longer evident.

Generally it can be noted that refrigeration system performances improve when the average external dry bulb and wet bulb air temperature decrease, but there is also a threshold beyond which climate influence is not evident.

The effect of climate could be recognized when the free cooling configuration was analyzed. Free cooling configurations are plotted in the middle between the BASE and ABS configurations, and shows a strong decrease in energy consumption when the climate is colder. Due to the low energy required by the cabin and the direct use of external air (the cooling plant is completely turned off) the total consumption of energy could be reduced by 50% both for dry cooler and cooling tower configurations.

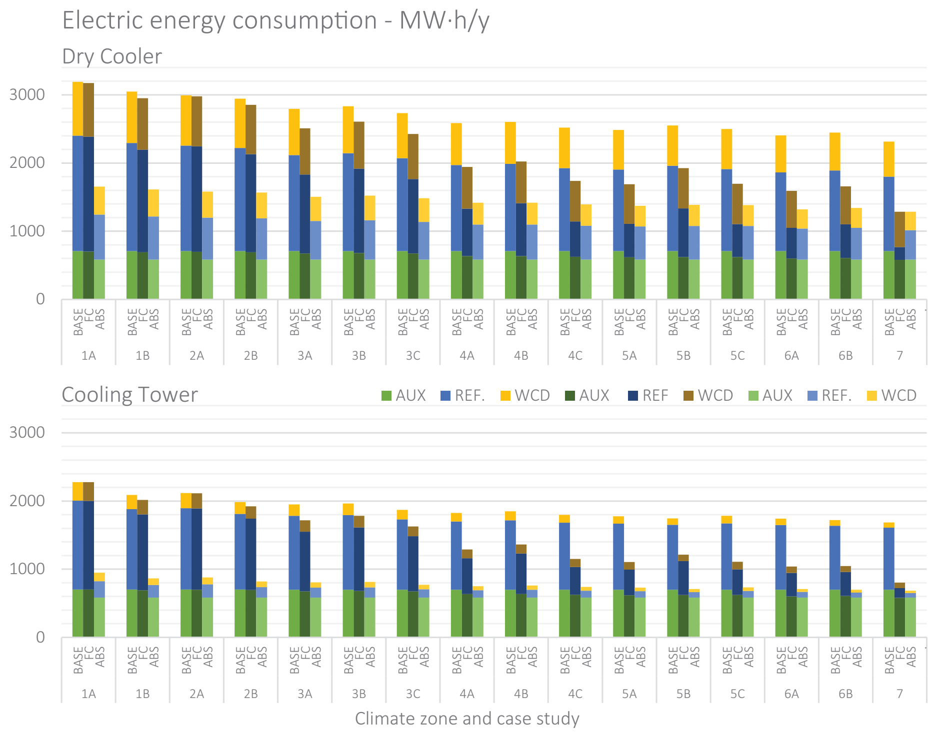


FIGURE 5.10: Yearly electric energy consumption based on configuration and climate. Configurations are arranged in the following order: BASE, FC, and ABS; AUX, REF, and WCD shown in the legend refer to the total energy used by auxiliaries (AUX), chiller (REF), and water cooled duct heat rejection system (WCD).

The absorption cooling configurations are the least influenced by the climate. This configuration is particularly efficient in hot climates, in fact for both CT and DC, the electric energy consumption is the lowest reached, comparable with FC only for cold climates. This benefit relies on the absorption chiller, which uses electricity only for solution pumping, which most of the time is considered negligible.

Lastly, comparing the condensers used, the energy consumption by the chillers using cooling towers are less intensive when compared with the ones using DCs. The temperature of cooling water exiting from a CT is close to the wet bulb air temperature, which is usually at most the dry bulb temperature, allowing higher EER values. The cooling water temperature exiting from a DC is close to the dry bulb temperature.

5.2.4 Water consumption

The yearly water consumption is illustrated in Figure 5.11. Details as reported in Figure 5.10 are not the same, since the water consumption of the plant elements are *indirect*, except for cooling towers. The consumption of water shown in Figure 5.11 with gray bars underlines that water consumption for the DC configurations is *indirect*, that is consumed as a result of water being required for the generation of electricity that they use. Blue scale bars show the direct use of water.

Starting with configurations using DCs (first row of bar chart in Figure 5.11) it can be noted that the national energy mix plays a relevant role in terms of water footprint.

The water consumption trend for each climate is the same as shown for electricity consumption in Figure 5.10, i.e. the BASE scenario is the most intensive followed by FC and ABS. In particular FC and ABS configurations are equal only for climate zone 7.

Water consumption of CTs are shown on the second row of Figure 5.11. To understand the effect of climate in terms of water evaporation, BASE and ABS configurations should be observed since in FC configurations the CT does not operate for the same amount of time.

In these configurations the amount of water required for condensation and heat rejection is related to climate parameters i.e. wet bulb temperature, dry bulb temperature and relative humidity. Since relative humidity is low in type B zones (dry climates) in comparison to type A (humid) and

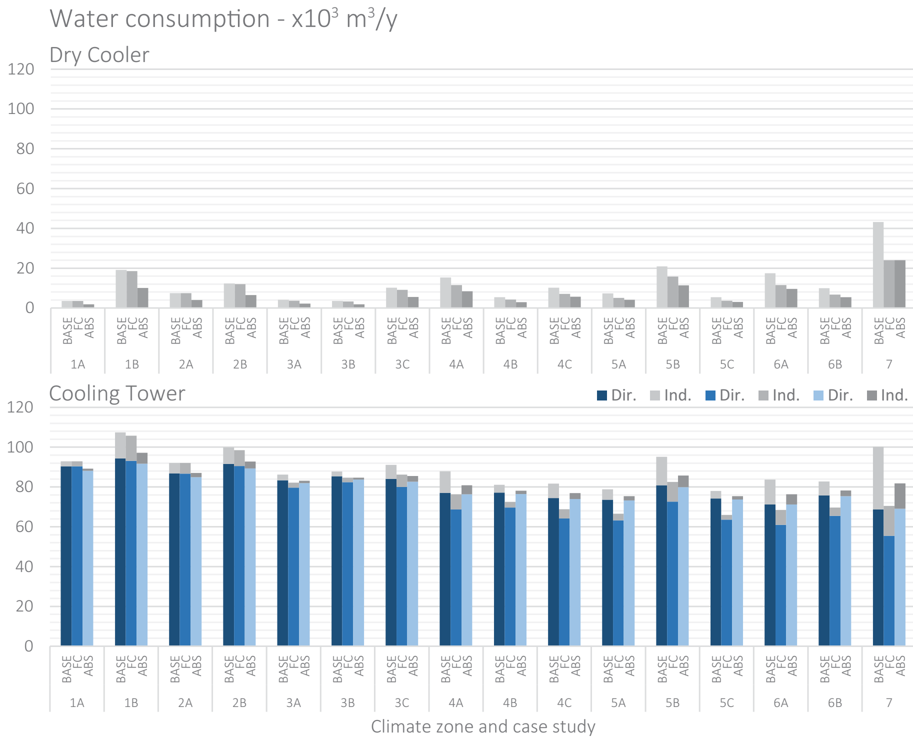


FIGURE 5.11: Yearly water consumption for various configuration types and climates. The pattern used in the bar chart plot is: BASE, FC, and ABS. Grey bars indicate the indirect consumption of water derived from the electricity utilization based on the national energy mix. Blue bars are used to indicate the direct consumption of water, in this case by evapora-tion.

type C (marine) zones, ambient air can absorb more water (see Table 4.1 for further details).

Figure 5.11 underlines the impact of the national energy mix on the water footprint. It can be noted that the total sum can vary substantially depending on the climate and the nation involved. Climate *7* (Östersund, Sweden) shows a water footprint comparable with climate *2B* (Cairo) for MVC scenario despite the strong difference in climate.

5.2.5 CO_2 emissions

The CO_2 footprint is shown in Figure 5.12, and since no combustion is required for plant operation, carbon emissions are only *indirect* and based on electricity consumption.

The trends are the same as shown for the previous indicators, i.e. the most intensive carbon configuration is the BASE case, followed by FC and ABS. The influence of climate here is not clearly defined, in contrast with the national energy mix, in which it is particularly clear. Climates 4A and 7, Lyon and Östersund, show a low carbon footprint thanks to the high presence of nuclear power generation (76% France, 38% Sweden) and hydropower (48% Sweden).

With this indicator it can be observed that, for all the cases shown, by increasing the energy efficiency incentives or carbon tax a switch from low intensity configurations to high intensity configuration could theoretically happen.

5.2.6 Primary energy consumption

Lastly, primary energy consumption is shown in Figure 5.13. It can be seen that configurations using dry coolers are more intensive than configurations using cooling towers. The trend between BASE, FC, and ABS is the same, as was shown for the previous indicators.

Regarding the influence of climate, it can be seen that it does not significantly affect the total amount of primary energy consumed. Again the national energy mix is particularly influential even though climate influence is more visible if compared with the CO_2 footprint analysis in Figure 5.12.

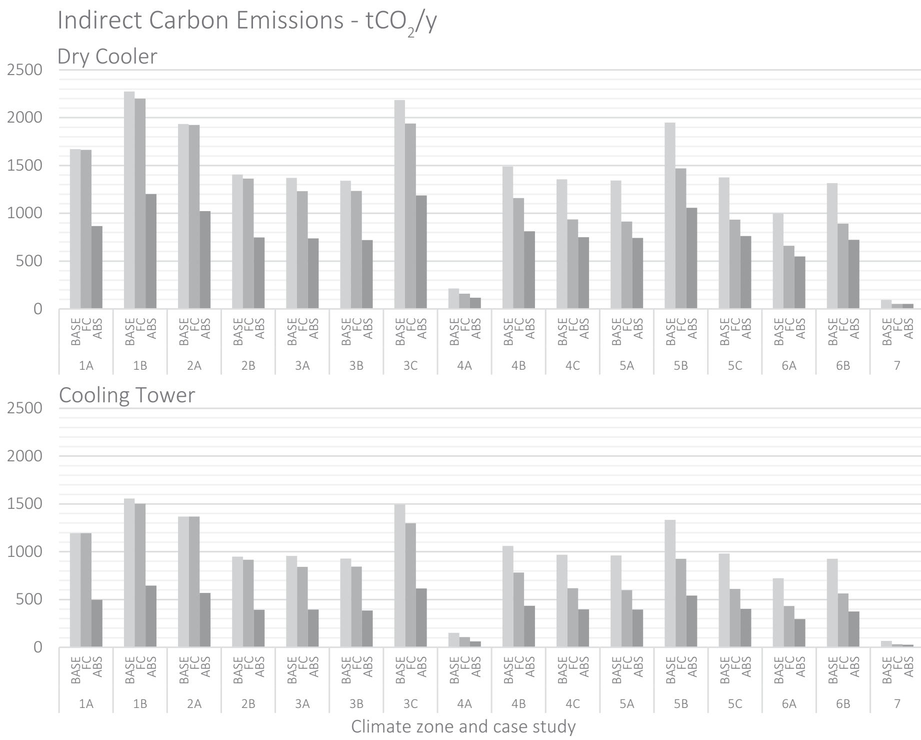


FIGURE 5.12: Carbon emissions for various configurations and climates. The pattern used in the bar chart plot is: BASE, FC, and ABS. All the emissions here reported are indirect, i.e. derived from electricity usage.

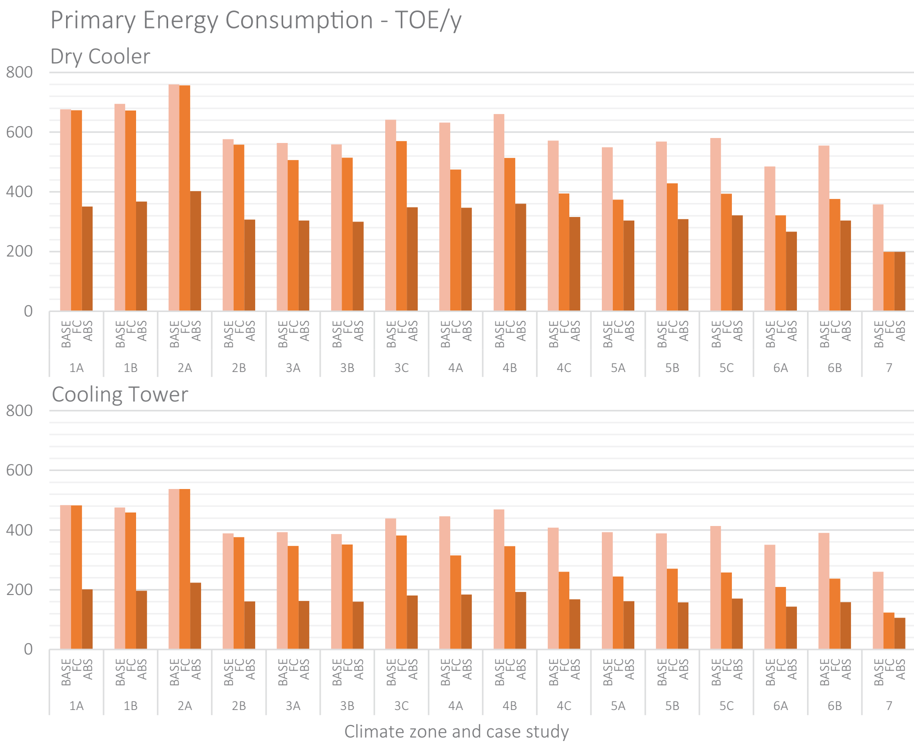


FIGURE 5.13: Primary energy consumption for various configurations and climates. The pattern used in the bar chart plot is: BASE, FC, and ABS.

5.2.7 Free cooling in-depth analysis

A special section is dedicated to free cooling configurations, since they were not considered in case study *C1a*. Free cooling applied to electric cabin air conditioning is a simple way to retrofit an existing air conditioning plant, since it requires only minor changes to the ventilation system, the control systems of the MVC, the relative condenser, as well as at the cabin building.

In this work a simplified version of free cooling for a steelworks plant is simulated. In particular, the control function of the thermostat is an "upper bound filter" which sets the use of FC when the outside temperature does not exceed a fixed temperature. Four threshold temperatures are simulated: 12 °C, 15 °C, 18 °C, and 21 °C. After simulation runs, it can be stated immediately that a temperature of 21 °C is too high, as it leads to an internal air temperature in the cabin greater than 40 °C which is above the limit imposed by the manufacturer.

The figure 5.14 shows the percentage of the free cooling mode usable depending on the climate zones. As one might expect, the colder the climate is the more the FC mode is usable.

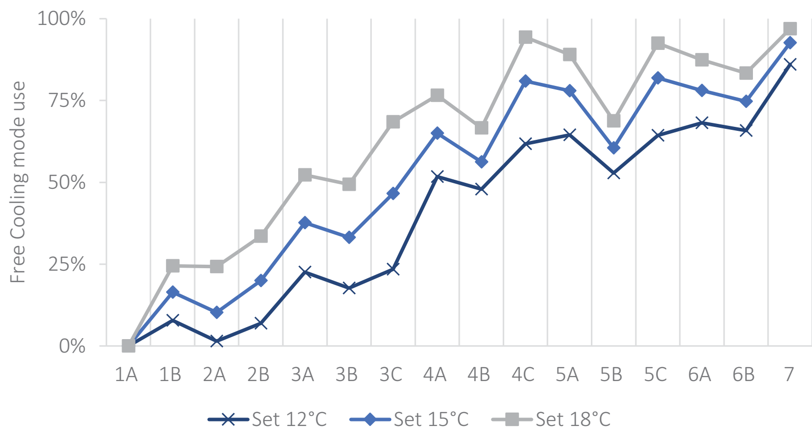


FIGURE 5.14: Simulated annual cabin load for 16 climates zones and percentage of Free Cooling.

Within the scope of this work, the economic feasibility is evaluated by comparing FC with its main competitor, which is ABS. From the point of view of the environmental indicators, the results shown in the previous section relate directly to the temperature that the FC system is set at. As 12 °C is realistically a very low temperature to set the system at, values derived from it are precautionary and are likely to underestimate the potential energy saved.

The economic evaluation of these two configurations is performed using the payback period as indicator. In particular, the evaluation of the payback period is evaluated using equation 3.7, therefore the result is the break-even point between absorption and free cooling. It is reasonable that the payback period evaluated is low for hot climates and increases substantially when climates become colder i.e. ABS chiller is not convenient when compared with FC.

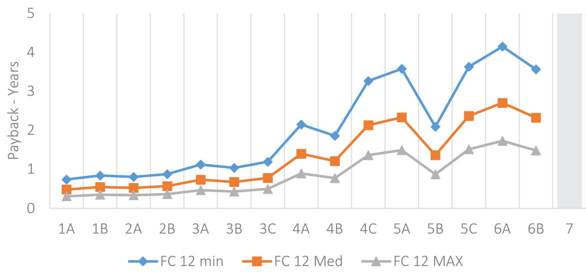
The payback period for configurations using a dry cooler as a condenser are shown in Figure 5.15(a), these results are evaluated based on three prices for electricity: the minimum, the average, and the maximum of electricity cost, shown in Table B.1.

As expected, payback periods are shorter for hot climates i.e. it is better to consider an ABS configuration instead of the FC. In the worst price condition (low electricity price) PB period is less than 1.5 years up to climate zone *4A*. When the climate becomes particularly cold the payback period increases substantially, i.e. it is better to consider FC instead of ABS. Climate *5B* is an exception, related to the fact that FC mode is available only for 60% during the year (60.5% see Figure 5.14).

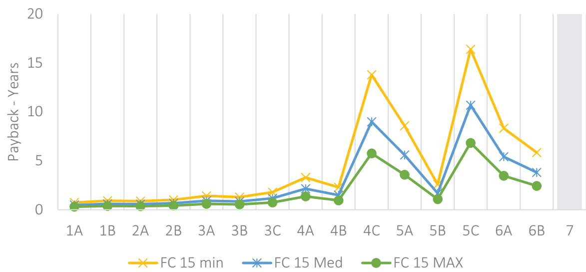
A further sensitivity analysis based on different set temperatures for FC is illustrated in Figure 5.15(b) and Figure 5.15(c) with temperatures of 15 and 18 °C respectively. As the temperature the FC system is set at increases, the payback period of FC drops below the commonly preferred maximum of five years and ABS becomes less favorable. In Figure 5.15(c) payback for climates *4C*, *5A*, *5C*, *6A* and *7* are outside the extremes of the ordinates.

The sensitivity analysis for FC systems using CTs is shown in Figure 5.16. Since CTs are used, another parameter is introduced: the water price. As evaluated for electricity, water prices are the minimum, the average and the maximum of the prices available in Table B.1.

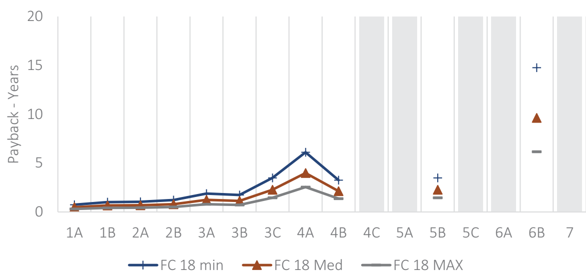
Figure 5.16(a) shows the sensitivity analysis with a fixed water price and variable electricity price. In contrast with the configurations using dry coolers, for the minimum electricity price (green line) after climate *4A*,



(a)

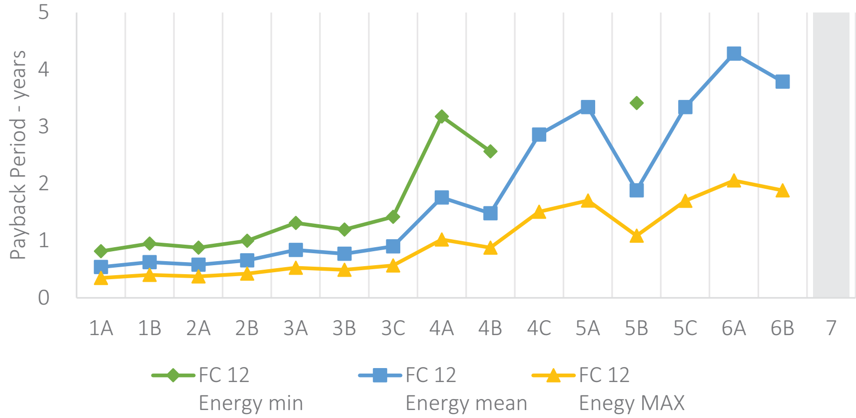


(b)

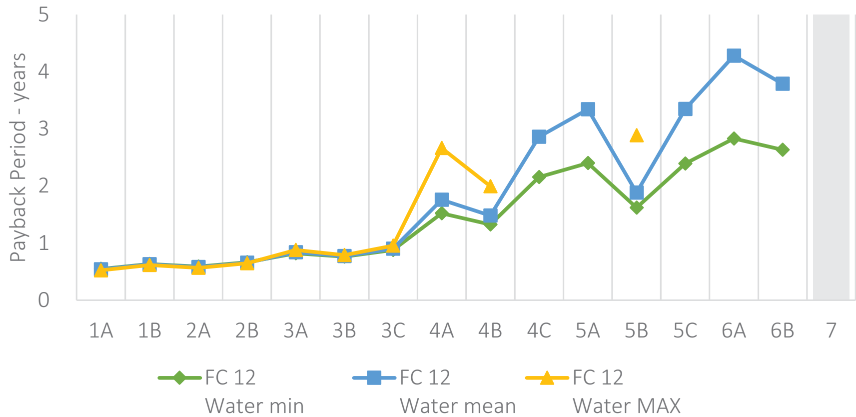


(c)

FIGURE 5.15: Sensitivity analysis for FC configurations with DC based on the electricity price $min = 0,073$ €/kWh$_e$, $Mean = 0,112$ €/kWh$_e$, and $MAX = 0,175$ €/kWh$_e$. (a) FC with DC temperature set to 12 °C; (b) FC with DC temperature set to 15 °C; (c) FC with DC temperature set to 18 °C.



(a)



(b)

FIGURE 5.16: Sensitivity analysis for FC configurations with CT based on the water price $min = 0,771$ €/m^3, $Mean = 1,735$ €/m^3, and $MAX = 3,813$ €/m^3. (a) FC with CT temperature set to 12 °C, sensitivity on electricity price with fixed water price; (b) FC with CT temperature set to 12 °C, sensitivity on water price with fixed electricity price.

FC is the best option, with an exception made for climate *5B* which has a reasonable PB period of 3.5 years. By increasing the electricity prices, a reduction in the PB period is achieved.

The sensitivity analysis illustrated in Figure 5.16 is evaluated using a fixed the electricity price (mean) and varying the water price. A reduction in the water price improves the performance of absorption cooling over free cooling. This is consistent with the higher water consumption of absorption cooling than the FC configuration, as can be observed in Figure 5.11.

5.3 Results of case study C2

The aim of case study *C2* is to understand what to do with industrial waste heat from a water-energy nexus perspective, considering the distinction between internal and external energy demand.

The reference case consists of the complete dissipation of excess heat, while for heat and cooling loads the boilers and the MVC chillers are used respectively. Waste heat recovery configurations are: district heating and cooling (DH+C), the direct use of heat for cooling purposes *on site*, and electricity production with an ORC.

The waste heat available for heat recovery at the industrial site is 1 MW_{th}, and the district heating network is 6.6 km in length. The results presented are based on a lifetime of 30 years, which is a reasonable estimate of the lifetime of a DH pipeline. In this period of time, some elements are assumed to be replaced.

5.3.1 Economic analysis

In the economic analysis, the LCC is used as an economic indicator. Results are illustrated using horizontal stacked bars. Configurations are ordered first by the condensation unit, and then by the heat recovery configuration. Cost typology is highlighted using different colors depending on the plant component analyzed.

From an economic point of view, Figure 5.17 shows that DH is by far the most unfruitful option for heat recovery in both countries analyzed, generating cost overruns of more than 1.7 M€ when compared with the base case

(1.71 Austria and 1.85 Italy). In Austria, power generation reduces LCC by less than 313.8 k€ against a total investment of almost 390.81 k€ for ORC.

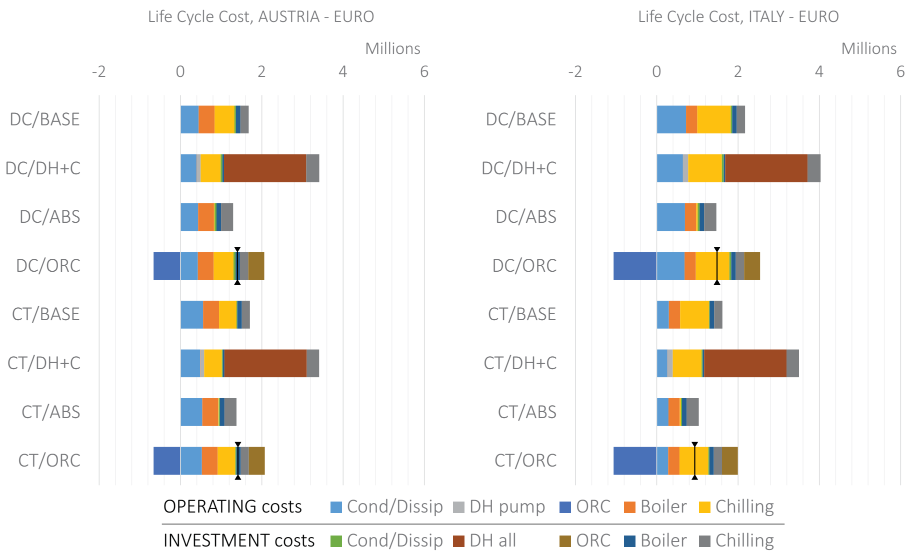


FIGURE 5.17: Life cycle cost analysis.

The best solution for Austria is the use of excess heat for absorption cooling, which with an investment of 207 k€ reduces the LCC by 384 k€. In Italy, savings from ORC are more attractive than in Austria due to the high cost of electricity, with for the same investment than as Austria resulting in an LCC reduction of 727 k€.

An ABS configuration reduces The LCC by 708 k€ against the same total investment as in Austria. So in the Italian scenario, ORC power generation should be preferred to ABS for cooling. The ineligibility of district heating as the best solution a result of to two factors: the low heat demand of the end user and the high cost of pipeline construction (6.6 km in length).

5.3.2 Carbon footprint and energy analysis

For the reference case, the results of comparing different scenarios in terms of carbon footprint (Figure 5.18) and in terms of primary energy consumption (Figure 5.19) need to be assessed depending on the country and the indicator.

Starting with Austria, it can be seen that district heating is the best solution, reducing the total amount of CO_2 by 3100 tCO_2 for both condensers used. The second competitor is ORC, which reduces the total amount of CO_2 by 2300 CO_2 when compared with the base case. Regarding the primary energy consumption, the best solution is ORC when CT is used as a condenser. The total primary energy consumption is reduced by 1500 TOE when the ORC configuration is used with either CT or DC, but when DC is used a higher consumption of electricity usage is observed.

FIGURE 5.18: Study case *C2* configurations CO_2 results.

The construction of DH pipelines may account for up to 7.6% of the system carbon footprint, and the pump operation may account for up to 3% of the system carbon footprint (achieved in DC/DH scenarios), yet the carbon footprint of a district heating network enabling waste heat recovery is negligible, compared with that of fossil fuels consumed by the remote building in benchmarking scenarios.

In Italy the situation is different, carbon emissions are generally higher when compared with Austria. It can be noted that boiler emissions are

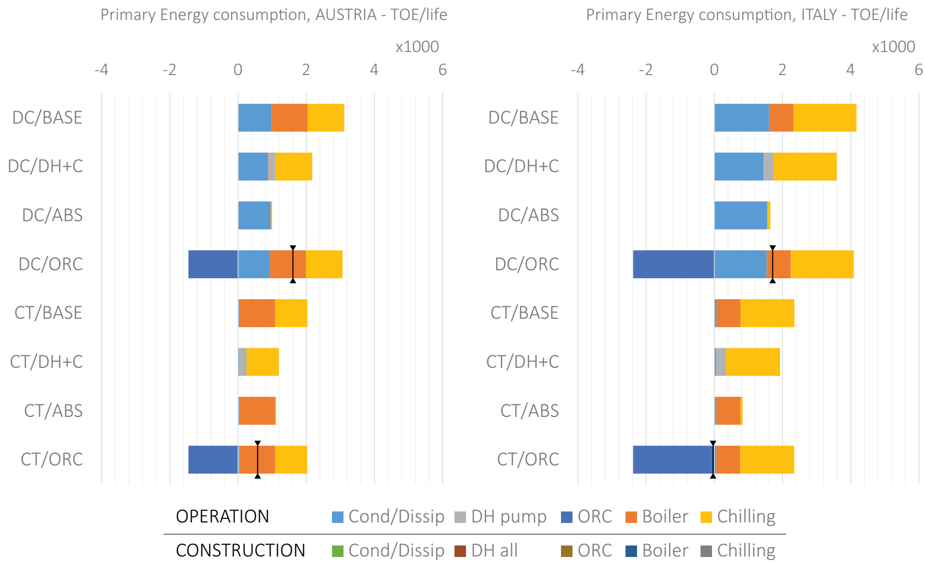


FIGURE 5.19: Primary energy consumption results.

lower than Austria due to the fewer operation hours required, while condensation and chilling are more intensive. This is related to a higher cooling demand and to the national energy mix, which has a higher carbon footprint. The difference in emissions between Italy and Austria for the DC/BASE configuration is 3773 tCO_2, while for the CT/BASE configuration it is 1194 tCO_2.

ORC configurations are the best solution for both CT and DC condensers. The effects of the electricity produced can be seen in the reduction in CO_2 generated, which is 5688 tCO_2 in Italy, and 2381 tCO_2 in Austria. The high CO_2 intensive national energy mix of Italy suggest also to use heat recovered for ABS cooling that can reduce by 4313 tCO_2 and 3639 tCO_2 when DC and CT are used.

The most sensitive parameters of the system are the footprints of the national energy mix and the energy demand of the end-user. The carbon footprint is evidently represented by fossil fuel demand by boilers, so benefits are slightly higher in Austria than in Italy due to the longer operating hours of heating systems. Vice versa, benefits in Italy are slightly higher when absorption cooling is used since it does not require electricity during operation. By reducing the operation of heat dissipation systems, heat recovery for DH does not provide strong advantages in DC scenarios.

In terms of carbon footprint and primary energy savings, allocating low grade waste heat to power generation is beneficial, as we can deduce by comparing total net amounts (i.e. direct emissions – credits for reduced electricity consumption). This holds true in Italy in particular, where chilling systems consume more energy and the national energy mix has a low carbon footprint.

For the reference system, allocating waste heat recovery to district heating offers carbon emission savings between 1852 tCO_2 (DC/STD - IT) and 3116 tCO_2 (DC/STD - AT) over the system lifecycle. Allocating the same heat to the ORC would offer savings between 2379 tCO_2 (AT, DC scenarios) and 5795 tCO_2 (IT, DC scenarios).

5.3.3 Water footprint analysis

Recommendations are in some cases reversed when taking into account lifecycle water footprints (Figure 5.20). The water footprint is mainly determined by the contribution of waste heat dissipation systems and the electricity consumption.

Wet cooling systems (CT) have a remarkably higher impact due to direct consumption of water, but in countries where electricity generation has a high water footprint, the indirect impact of electricity consumption for DC is also not negligible. Compared with the impact of heat dissipation in the system, the contribution of fossil fuel boilers to the water footprint is negligible, and so is the embodied footprint of equipment and network manufacturing.

The indirect impact of electricity consumption for pumping in DH networks is very small, but not negligible in DC scenarios. As a result, district heating actually produces a higher water footprint for the reference case, both in Italy and in Austria, but the increased footprints in DC scenarios are small (1.15 and 3.11% of the initial footprint in Italy and Austria respectively). Wider networks, with higher heat and pressure losses, could have a negative impact on water footprint in DC scenarios.

On the other hand, in spite of low efficiency power generation with ORC significantly reduces water footprints by avoiding consumption of electricity from the national grid. As a result, the net balance for water footprint is even negative in DC/ORC scenarios not only in Austria, but also in Italy.

If waste heat dissipation occurs with CTs, the reduction in cooling load achieved through heat recovery results in a lower direct water consumption. Such savings are higher in ORC scenarios, due to the higher national mix footprint, and lower for DH.

Nevertheless, taking into account credits for indirect water consumption, the net water footprint balance in CT scenarios is more favorable to power generation in Austria. Due to the lower water footprint of the Italian national energy mix, ABS is a good solution when ORC is not available.

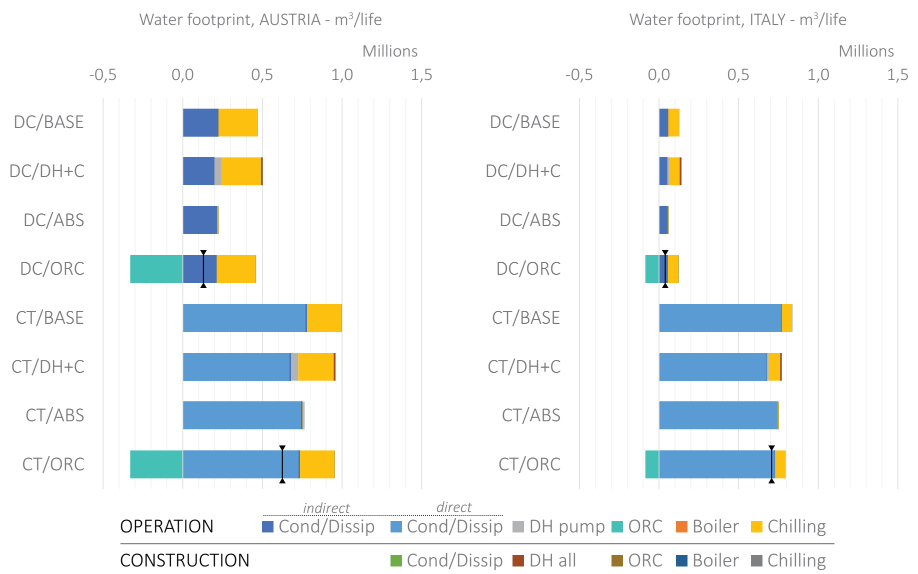


FIGURE 5.20: Water consumption.

5.3.4 Sensitivity analysis

The analysis of case study *C2* has been performed for a fixed distance between the waste heat source and the final user; however, investment costs, heat dispersion and pumping costs are a function of pipe length.

Sensitivity analysis related to this case has been performed by varying the length of the district heating pipes from zero (which corresponds to a hypothetical internal use of waste heat) to a maximum length of 10 km.

Figure 5.21 shows the sensitivity analysis performed for the configurations of concern in the two pilot regions, considering the economic indicator LCC.

As expected from figure 5.17, LCC is higher in Italy than Austria, except for ABS and ORC configurations which are strongly dependent on the cost of electricity. For all configurations, a break-even point between the reference configuration and DH can be identified. Values of break-even point are always lower or equal to 1.5 km, and for Italy even lower, reaching the value of 1 km, underling the economic weakness of this configuration at least within the boundaries and conditions assumed in this work.

The sensitivity analysis for the CO_2 footprint is shown in Figure 5.22. From the emissions point of view, break-even points are higher when compared with the previous cases. In Austria, when DC is used as a condenser/heat rejection unit, DHC outperforms ORCs if the length of the pipe does not exceed 4.5 km, while, to outperform local ABS alternatives, the pipeline length should not be longer than 8 km.

A completely different situation can be found in Italy, where due to the high footprint of the national energy mix, it is preferable to use excess heat for power or cooling generation.

In Austria, when CTs are used, a small variation can be found in break-even point between ABS and DH which is shifted towards the value of 8.5 km, and vice versa for Italy where the shift is toward lower distances.

An interesting part of the sensitivity analysis is related to the water footprint evaluation, Figure 5.23 shows the results for the various configurations of concern.

Starting with configurations using DCs it can be noted that, the overall water footprint of DH grows when the pipe length is increased, and that only when the length is close to 0 the overall blue water footprint of DH is lower than the total blue water footprint of the reference case.

When CTs are used, the water consumption of every configuration is increased. An interesting trend can be observed in this case: the slope of the line is negative, which means that by increasing the pipeline length the water footprint is reduced. We can deduce that in CT scenarios DH pipes favorably substitute water intensive heat rejection systems as dissipation

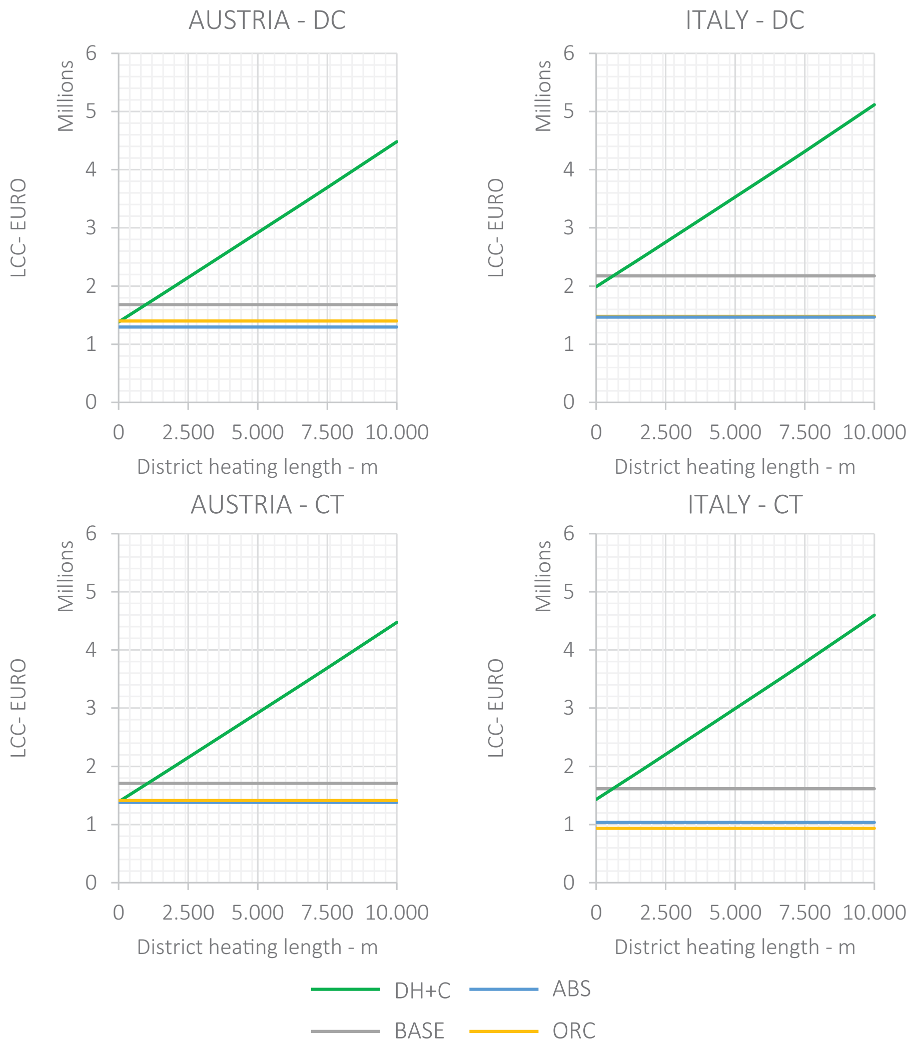


FIGURE 5.21: Sensitivity analysis on LCC considering different district heating lengths.

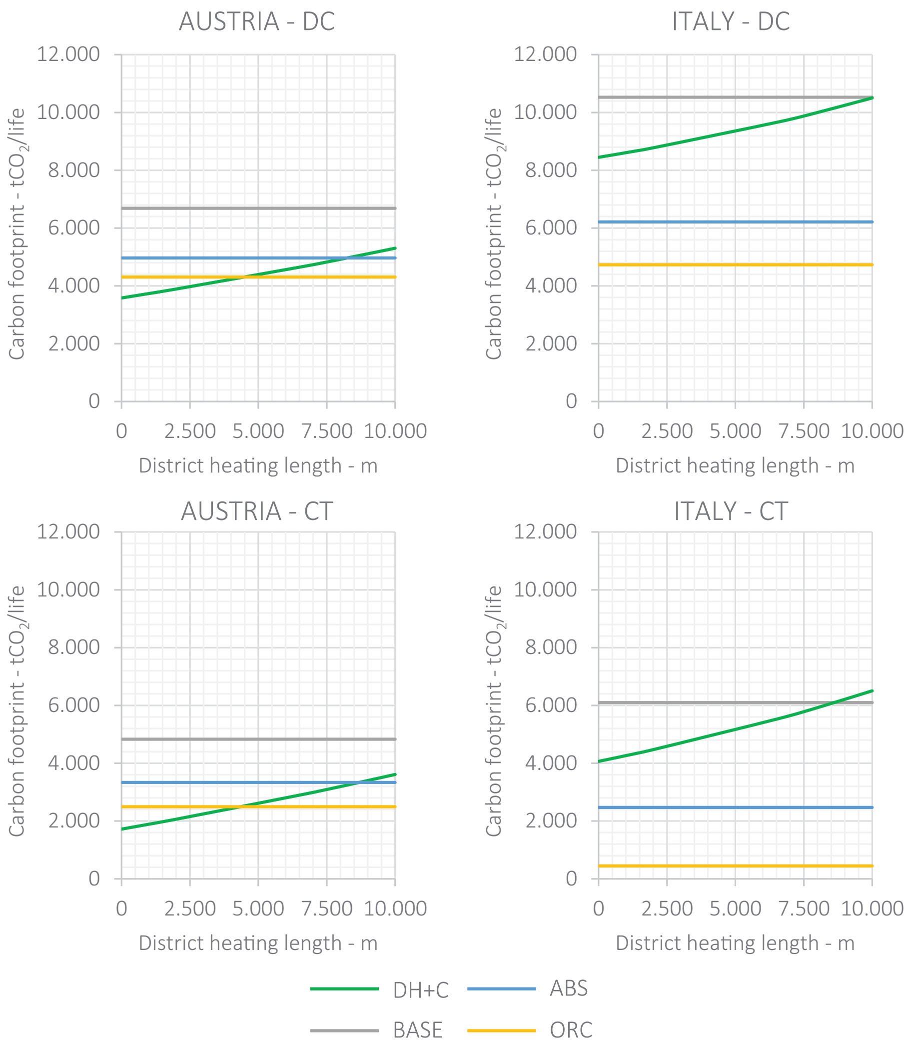


FIGURE 5.22: Sensitivity analysis on carbon footprint, pipeline length as variable.

systems, with a more evident effect in longer systems due to higher heat dispersion. It should be noted that the scales of the Figures 5.23 are not the same, in particular the scale of CT configuration for Italy has a limited range along the ordinates in order to show this trend.

When the water footprint is high a threshold can be established between the dissipation benefits of DH and the indirect consumption of water derived by electricity used for pumping. This is the case for Austria, where when CT is used in the DH configuration, after a certain length the water footprint increases with the incrementation of the pipeline length. The threshold point for the case analyzed has a value of 8 km.

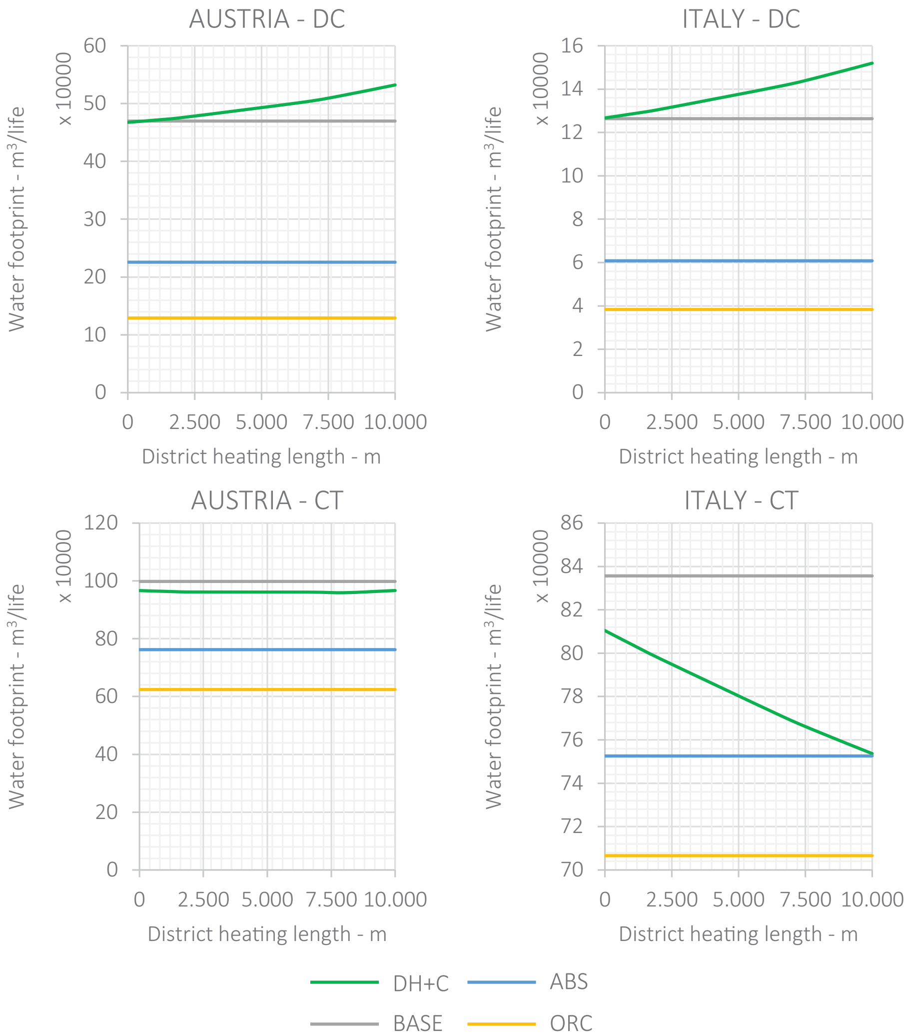


FIGURE 5.23: Sensitivity analysis on water footprint, pipeline length as variable.

Chapter 6

Conclusions

The concluding chapter presents the answers to the research questions posed in the introduction. Conclusions will be presented in the same order as used for the results description. Future research, limitations, and further developments are also presented.

6.1 Case study C1a

Starting from a case study of low grade waste heat recovery for absorption cooling or for power generation through ORC at an Italian electric steelmaking site subject to EU ETS obligations, a generalization was attempted by evaluating the same project as if it were in each of the EU-15 countries.

To this end, a first estimate of the national water footprints for electricity generation in each of the EU-15 countries was proposed. As to the water-energy-GHG nexus at country level, it has been found that EU-15 countries with higher carbon and primary energy indicators have significantly lower water consumption indicators, and vice versa, mainly due to hydro-power footprints and shares. This is in line with recent findings by Ref. [146], which presents the challenge of reducing both carbon and water footprints at the same time. The same challenge applies to industrial heat recovery projects.

In this case study, it has been shown that absorption cooling is a viable technology for exploiting low grade waste heat from water cooling systems at a typical electric steelmaking sites under the economic conditions present in all EU-15 countries. This study has highlighted that, in countries with low water prices and high electricity prices, a technology switch from dry coolers to cooling towers may be the most cost-effective option, though depending on the national energy mix, it may result in as much as a tenfold increase in the consumptive water footprint of the process. It has thus been verified that recovering waste heat from cooling systems does not always generate a reduction in the consumptive water footprint of the process in question.

Based on the data evaluated in this study, the economic conditions that foster a switch toward water intensive cooling options can be found to the greatest degree in southern European countries, which are typically affected by water scarcity the most. Particularly severe climate conditions, and increasing summer temperatures, may further hinder the application of dry coolers in southern Europe by reducing systems efficiency. Further research is being conducted to evaluate these aspects, since climate differences between different countries have not been considered in this study. A further limitation of the present investigation is that it relies on a particular size and shape of the examined solutions. There is scope for future research to investigate further technologies, sizes and waste heat flows with this approach.

The present study case was also limited by the difficulty in obtaining data, particularly on the real costs of water and electricity for industrial customers. Although appropriate tools exist to handle uncertainty, including the Monte Carlo approach used, in our view more accurate results could be the outcome of collaborative research in an international framework, which could also link a water-energy-GHG nexus perspective to the investigation of industrial waste heat potentials. Since Southern Europe appears to be the most problematic area, and since our analysis had to be limited to the EU-15 due to the lack of economic data for the remaining EU-27 countries, countries located in South-Eastern Europe could be a promising target for future research projects.

It has been shown that the economic performance of different technologies and configurations is mainly determined by water and power prices. However, technology choice was also found to be affected by policies aimed at carbon emission reduction, which cause a shift towards water intensive technologies for carbon prices of 20 €/tCO_2 and above. It is thus recommended that analysts evaluating energy efficiency projects in the steelmaking industry, and in process industries on the whole, calculate performance indicators both for energy and water consumption, as well as for carbon equivalent emissions.

It is also recommended that policy makers designing incentives supporting energy efficiency or GHG reduction projects for the industry combine them with constraints, incentives or goals for water consumption reduction, taking a nexus approach.

6.2 Case study C1b

In this study case the energy, water, CO_2 and primary energy consumption of different configurations of air-cooling systems for electrical cabins have been analyzed across different climates zones.

Using the results obtained from the previous case (*C1a*) an evaluation of footprints for water, carbon and primary energy based on the national energy mix have been performed for 16 of 17 ASHRAE climates zones.

It was found that the cooling load required by the electric cabin is not affected substantially by the climate zones: the difference between climate *1A* (hot) and *7* (cold) is around 3%.

The effects of climate in the BASE case lead to a reduction of 27% of the total electric energy consumption moving from very hot to very cold climates. The increased efficiency is related to the best EER being achieved in cold climates. The electric energy reduction of ABS configurations over the same climates is 22%, and they are also the overall least resource intensive configurations studied. As expected, free cooling configurations are very effective in cold climates, with a peak reduction of 60% when compared with the hottest climate. It should be noted that after climate *4A* the energy efficiency improvement only reached 10% i.e. the plant's electricity consumption of climate 7 is only 10% smaller than the plant located in climate *4A*.

From the water consumption point of view, configurations using CTs are the most intensive. Variations in the direct water consumption of CT based cooling plants in different climate zones are on the order of 20% to 25% for MVC and ABS configurations. The introduction of FC configurations may generate a reduction of up to 40% due to the switch-off of MVC plants.

The amount of water required for condensation and heat rejection in CTs is related to climate parameters i.e. wet bulb temperature, dry bulb temperature and relative humidity. Since relative humidity is low in type B zones (dry climates) in comparison to type A (humid) and type C (marine) zones, the ambient air can absorb more water in cooling towers.

However, the total amount of *direct* and *indirect* water consumption is higher in climate *7*, where the MVC configuration has a 7% higher consumption than the same configuration in climate zone *1A*. This suggest that indirect consumption, which depends on the national energy mix, has a higher impact on the water footprint than the climate does, even considering such disparate climatic conditions.

The water consumption of DCs is only indirect, and mainly depends on the national energy mix: it was observed that climate plays a minor role in these configurations. The carbon footprint and primary energy consumption indicators also show a stronger dependence on the national energy mix than on the climate. For instance, for the traditional MVC configuration it has also been observed that systems operating in climate *2B* would have the same primary energy consumption as those in climate *5C*.

The economic analysis of ABS over FC configurations has shown that ABS configurations are more profitable in hot climates, and a sensitivity analysis has demonstrated how this trend is affected by the temperature that is set in the control function of the FC system, which moves the break-even point toward hot climates.

The ABS, heat recovery based configuration, which has been proven to be more favorable than traditional configurations in Europe in case study *C1a*, is found to be outperformed by FC configurations in typical Central and Northern European settings.

The results presented in this study suggest a high dependence of energy efficiency on the climate for FC configurations and to a lesser by the MVC configurations. ABS configuration is less affected by the climate zone. Therefore, to find the most suitable cooling strategy based on the cooling load and the climate, it is generally useful to evaluate the dynamic performance of plant components under different climatic conditions.

Taking a nexus perspective, low grade waste heat recovery coupled with absorption cooling is always the best option in terms of energy and carbon footprint, even in cold climates and with a low-carbon national energy generation mix. However, in temperate to cold climates, free cooling has been found to be a better option in terms of the water footprint, due to a lower direct water consumption than corresponding ABS configurations.

6.3 Case study C2

The analysis of the reference case study and the subsequent parametric analysis allows the research questions to be answered at a systems level, i.e. considering how the combined performance of industrial systems (currently dissipating waste heat) and of remote users (currently using fossil fuels) is affected by the introduction of DH and ABS as a means for waste heat recovery. The answer to our first research question, i.e. under what circumstances is the allocation of low grade industrial waste heat to district heating preferable to power generation, depends on the goals and preference criteria.

The assessment of the carbon footprint and the water footprint of industrial waste heat allocation alternatives is relevant for policy makers and for public decision makers interested in supporting or authorizing only projects generating overall environmental benefits, e.g. not depleting water resources, to achieve energy savings. The economic performance of the overall system is of interest for any potential investor in DH systems, including process industries interested in symbiosis projects. On the basis of energy efficiency indicators and calculated footprints, investors may estimate additional contributions from country specific incentives. If the decision makers aim to minimize systems costs as well as carbon footprint and primary energy consumption, then absorption chilling for internal cooling demand is the best option for systems similar to the reference cases studied here, i.e. when waste heat availability is constant, and the heating systems that would be substituted have a reduced heat demand. If it is also desired to minimize overall blue water footprints, or at least to avoid their deterioration, then decision makers should also consider:

- what energy flows will be substituted for different waste heat utilization options
- what their current footprint is
- how much waste heat should be dissipated before and after energy recovery and how dissipation is performed, i.e. by consuming water or not

Our analysis has shown that, if the goal is to minimize the water footprint, then the allocation of low grade waste heat to power generation should be preferred. The best allocation of industrial waste heat depends on the regional electricity mix: if it is very water intensive, then waste heat-to-power solutions allow minimization of water footprints.

In general terms, our results for water footprints lead us to agree with the recommendation by Bartolozzi et al. (2017) that the carbon footprint should not be taken as a proxy for environmental impact.

Overall, it can be concluded that whether low grade industrial waste heat should be allocated to absorption chilling rather than to power generation or district heating is affected by local conditions (e.g. distance, demand), and by regional or national conditions (e.g. climate, energy mix).

The water-energy nexus, whose importance for "macro" energy systems (electricity generation) is widely acknowledged in the literature, is also reflected in projects involving "micro" energy systems (industrial waste heat) depending on the shifting of resources they may cause.

More generally, calculations made in this study case confirm, in line with the observations of Ivner and Viklund [150], that performance assessment for district heating systems is significantly affected by the choice of system boundaries (in our case, inclusion of industrial dissipation modes and of indirect impacts within the boundaries): decision makers to whom studies of this kind are presented should be made more aware of this fact.

6.4 Answer to specific research questions

1. *Which forms and uses of low grade waste heat recovery are economically preferable in process industries, considering internal and external use of heat recovery products?*

   The results coming from case study *C1a* and *C2* indicate that from the economic point of view, cooling configurations are the best options for low grade heat recovery especially for internal cooling, i.e. for process cooling purposes. An exception is made for Italy when CTs are used as heat rejection units. In this case ORC is preferable due to the high operating costs related to internal electricity production, which would be avoided through power generation. Case study *C2* also highlights that cooling configurations intended to meet an external cooling demand of an office building are not notably profitable when compared with power generation through ORC for both of the countries analyzed. Based on the assumptions made for case study *C2*, if the connection distance is high, as is realistically the case, DH is unprofitable when compared with the other configurations.

2. *Considering the lack of literature concerning low grade heat recovery for cooling purposes, is this configuration competitive or even preferable to other configurations, considering the economic feasibility and the water-energy-GHG perspective?*

   Absorption cooling is a profitable solution for heat recovery. The results shown *C1a* indicate that ABS(c) configurations have the shortest PB periods, especially when DCs are used as condensers. In reference to the second case *C2*, ABS configurations are the second best option when used for internal cooling. In line with findings by other researchers [40], absorption cooling is not a profitable option when the operating time is particularly low, as shown in case study *C2* when absorption chilling is coupled with district heating for external cooling demand. Taking a nexus perspective, if we just focus on process cooling as the final use as in case *C1b*, low grade waste heat recovery coupled with absorption cooling is always the best option as to energy and carbon footprint, even in cold climates and with a low-carbon national energy generation mix. Compared with power generation with low efficiency ORCs, absorption cooling is generally a better option in terms of economics, as well as under all water-energy-GHG nexus dimensions for process cooling with a long operation time, as in case study *C1a*. Considering building air conditioning applications or simulating more accurate cabin heating demand, such as in case *C2*, absorption cooling is found to be the second best option after ORCs, essentially due to the longer operating time of the latter.

3. *Since it is generally accepted that heat recovery is a good solution for CO_2 reduction and energy efficiency, does LGWHR always generate synergies for CO_2 footprint, primary energy consumption, and water footprint?*

   Low grade waste heat recovery does not always generate synergies for CO_2 footprint, primary energy and water consumption. Under certain economic conditions any efforts to reduce CO_2 emissions, when not guided by an overview of the system, may prove to be a trap ultimately leading to a switch from carbon intensive to water intensive configurations. Case study *C1a* highlights the pitfalls related to CO_2 reduction constraints. Incentives or taxes focusing on carbon emissions alone, combined with local economic conditions such as low water prices, could lead to a water intensive reconfiguration of recovery plants, or an implementation of suboptimal configurations, leading to carbon emissions reductions but to higher water footprints, such as with district heating in dry cooling configurations for case *C2*. It is thus recommended that policy makers designing incentives supporting energy efficiency or GHG reduction projects for the industry combine them with constraints, incentives or goals for water consumption reduction, taking a nexus approach.

4. *Following on from the previous question, how do the performances of LGWHR options depend on local conditions, such as market prices, the local energy mix and climate?*

   Case study *C1b* shows that climate does not affect heat recovery configurations for cooling purposes as substantially as the national energy mix (indirect emissions/consumption). For instance, configurations using CTs simulated in *Climate zone 7* have the lowest *direct* blue water consumption, but when the *indirect* consumption is taken into account the total amount of blue water consumption can reach the same intensity as in a hot climate. Market prices could play a valuable role, as demonstrated in case study *C1a*: based on data evaluated, the economic conditions fostering a switch toward water intensive cooling options can be found to a large degree in southern European countries, which are typically affected by water scarcity.

If decision makers aim to minimize systems costs as well as carbon footprint and primary energy consumption, then absorption chilling for internal cooling demand is the best option for systems. If it is also desired that overall blue water footprints are minimized, or at least to their deterioration avoided, then decision makers should also consider the footprint of the energy flows to be replaced and compare them with the footprints of the recovery opportunities, how much waste heat should be dissipated before and after energy recovery and how dissipation is performed, i.e. by consuming water or not.

In general terms, our results for water footprints lead us to agree with the literature recommendation that the carbon footprint should not be taken as a proxy for environmental impact.

6.5 Research limitations and further development

Like all models, the economic and thermodynamic models as well as software and input data adopted in this thesis are based on assumptions and simplifications, in part due to computational requirements and in part due to the complexity of a detailed analysis of heat recovery opportunities, especially when economic and footprint assessments are introduced.

Case study *C1a* was limited by the difficulty in obtaining data, particularly on the real costs of water and electricity for industrial customers. Although appropriate tools exist to handle uncertainty, including the Monte Carlo approach used, in our view more accurate results could be the outcome of collaborative research in an international framework.

The evaluation of plant energy demands as well as ORC power output was taken, considering average conditions. Part of this limit was overcome by case study *C1b*, which evaluates the transient energy demand by the plant's elements as well as by the end user. However ORC was not analyzed with a transient model, and in order to reduce the computational requirements of the simulation, some elements have been simplified.

Even though the air conditioning energy demand of the end user was simulated for summer and winter, in case study *C2* a simplification was introduced whereby the calculation of duration curves based on the simulation data was analyzed instead of a dynamic simulation being performed. Thus, although the energy consumption is the same, the time trend was lost. Another simplification is related to the model itself, a one-to-one (one source to one end user) analysis was performed while in real cases district heating should be treated as one-to-many (one source to many end users).

Further development of this project could be a more detailed analysis of ORC within the framework of study case *C1b* and with more detailed economic data on worldwide energy and water prices. Also, in this work it was always assumed that electricity from the national energy mix was used, or substituted by waste heat to power plants, which is a reasonable assumption in the long term and if our goal is to reduce the carbon footprint as well as the water footprint of electricity generation. Other substitution options could be investigated, e.g. by considering substituting only the most expensive energy sources, which might more realistically reflect short term market behavior.

Finally, within this work only a one-to-one district heating configuration with the point heating demand of a virtual building was investigated. The end user evaluated in case study *C2* could be evaluated using district heating and cooling design calculation codes like [140]. With this tool the analysis could be switched from one-to-one to one-to-many, giving the possibility of performing a wider analysis of realistic urban district energy systems.

Appendix A

Economic data

TABLE A.1: Plant components cost regression (with Monte-Carlo analysis).

Component	Cost function structure	Parameter	Characteristic parameter values, triangular distribution		
	Y in €		Min	ML	Max
Heat Storage	$Y = \alpha$	α	15.000	20.000	30.000
MVC Chiller	$Y = \alpha + \beta \cdot Q$	α	15.000	20.000	35.000
	Q in kW_f	β	90	112	150
Absorption Chiller	$Y = \alpha + \beta \cdot Q$	α	86.000	95.000	110.000
	Q in kW_f	β	90	94	100
Dry cooling system	$Y = C_0 \cdot (Q/Q_0)^m$	C_0	6.000	8.000	12.000
	Q_0 = 200 kW, Q = dissipation capacity (kW)	m	0,55	0,7	0,75
Wet cooling system	$Y = C_0 \cdot (Q/Q_0)^m$	C_0	48.000	60.000	110.000
	Q_0 = 8000 kW, Q = dissipation capacity (kW)	m	0,55	0,7	0,75
Organic Rankine Cycle	$Y = C_0 \cdot (P/P_0)^m$	C_0	150.000	300.000	400.000
	P_0 = 100 kW, P = nominal power (kW)	m	0,7	0,8	0,9

TABLE A.2: Plant component costs (without Monte-Carlo analysis).

Component	Cost function structure	Parameter	Value
	Y in €		
Boiler Natural Gas	$Y = C_0 \cdot (P/P_0)^m$	C_0 [€]	31.795
		P_0 [kW]	460
		m	0,758
Pump Water Circulation	$Y = C_0 \cdot (P/P_0)^m$	C_0 [€]	8.136
		P_0 [kW_t]	28,8674
		m	0,035
Heat Exchanger*	$Y = \alpha + \beta \cdot P$	α	14
	P in kW_t	β	6.272
Heat Exchanger‡	$Y = \alpha + \beta \cdot P$	α	21
	P in kW_t	β	11.372
Heat Recovery unit	$Y = \alpha \cdot P$	α	4,5

* For District Heating (DH) systems.
‡ For District Cooling systems.

Appendix B

Electricity and water prices

TABLE B.1: Expected values of electricity and freshwater prices for industrial consumers.

Country	Electricity Price[a]	Industrial Water Price[b]	Sources for water prices
	€/kW·h_e	€/m^3	
Austria	0,109	1,523	[120]
Belgium	0,110	2,329	[120]
Denmark	0,094	3,813	[120]
Germany	0,141	2,024	[122, 151]
Ireland	0,136	1,113	[145]
Greece	0,126	0,886	[123]
Spain	0,119	1,199	[120]
France	0,085	1,726	[120]
Italy	0,175	0,771	[120]
Luxemburg	0,100	2,255	[124]
Netherlands	0,093	1,723	[144]
Portugal	0,117	1,381	[120]
Finland	0,074	1,754	[120]
Sweden	0,073	1,791	[120]
United Kingdom	0,127	1,743	[120]

For sources in dollars, the historical (Dec 2013) exchange rate of 1 USD = 0,73 € was used.
[a] Elaboration on Eurostat [119] and OECD [120].
[b] Elaboration on cited sources.

Appendix C

TRNSYS decks, types and performance files

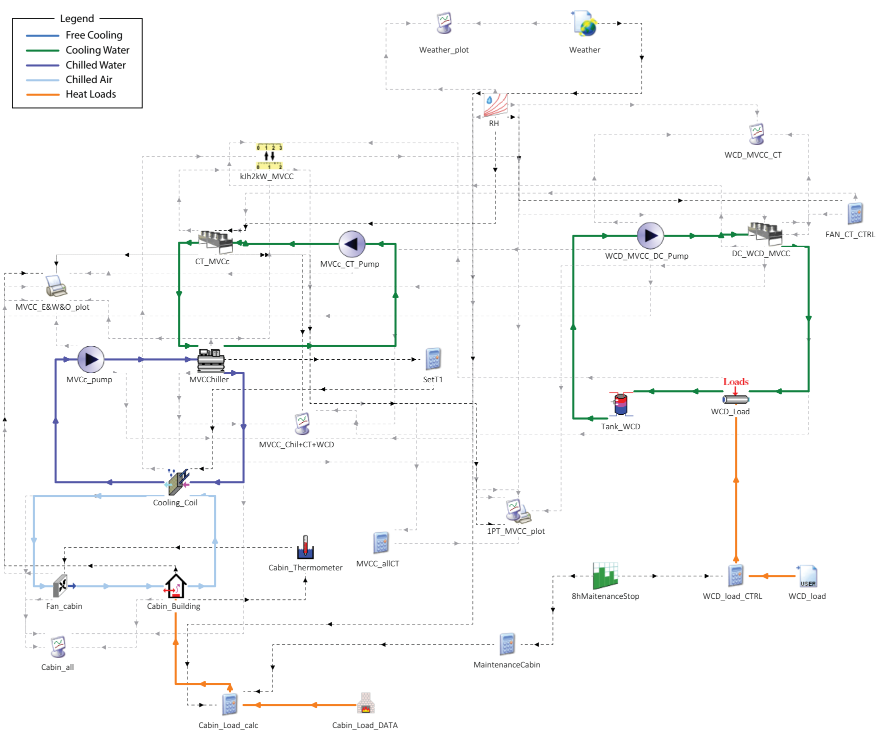


FIGURE C.1: TRNSYS deck of MVC with DC.

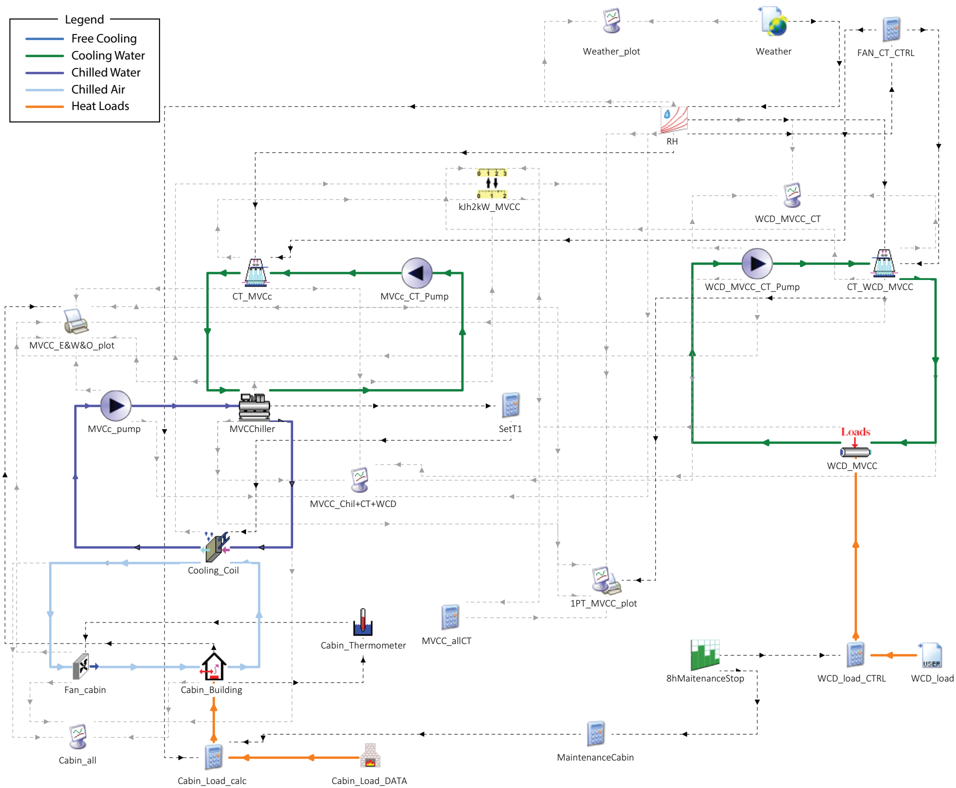


FIGURE C.2: TRNSYS deck of MVC with CT.

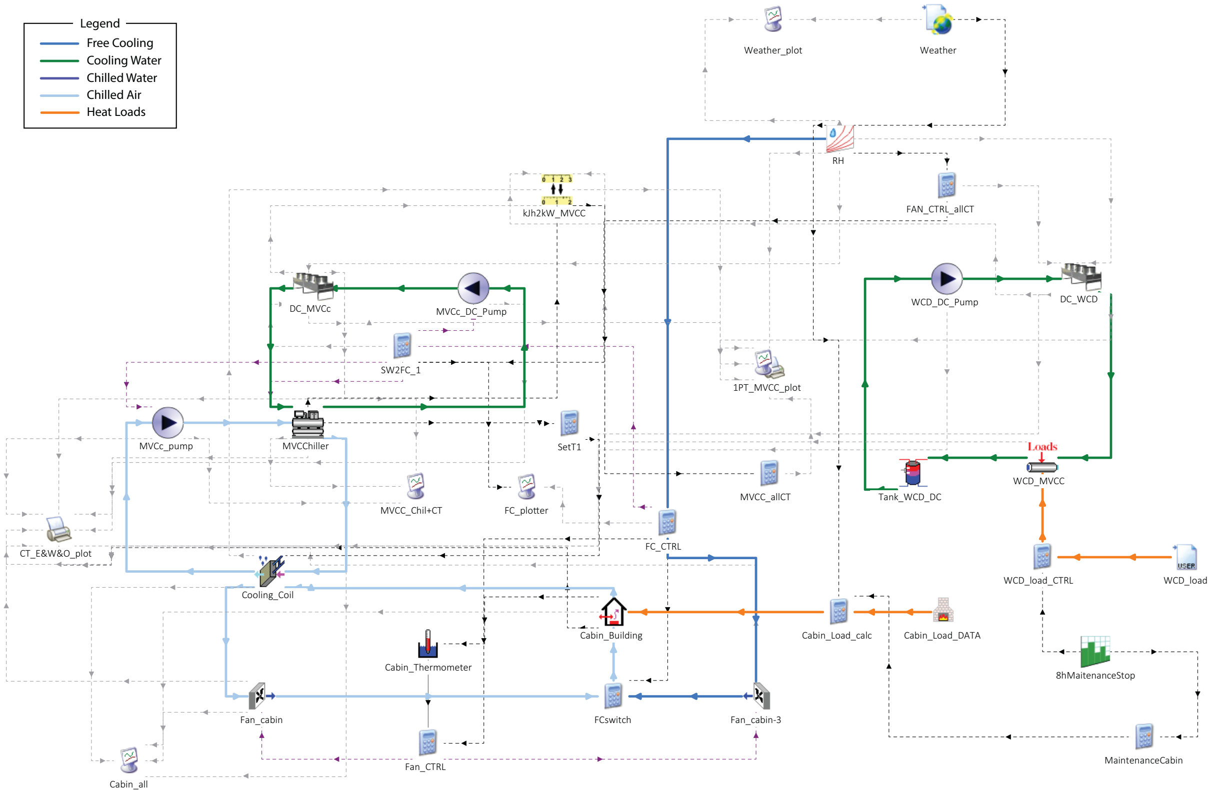


FIGURE C.3: TRNSYS deck of MVC+FC with DC.

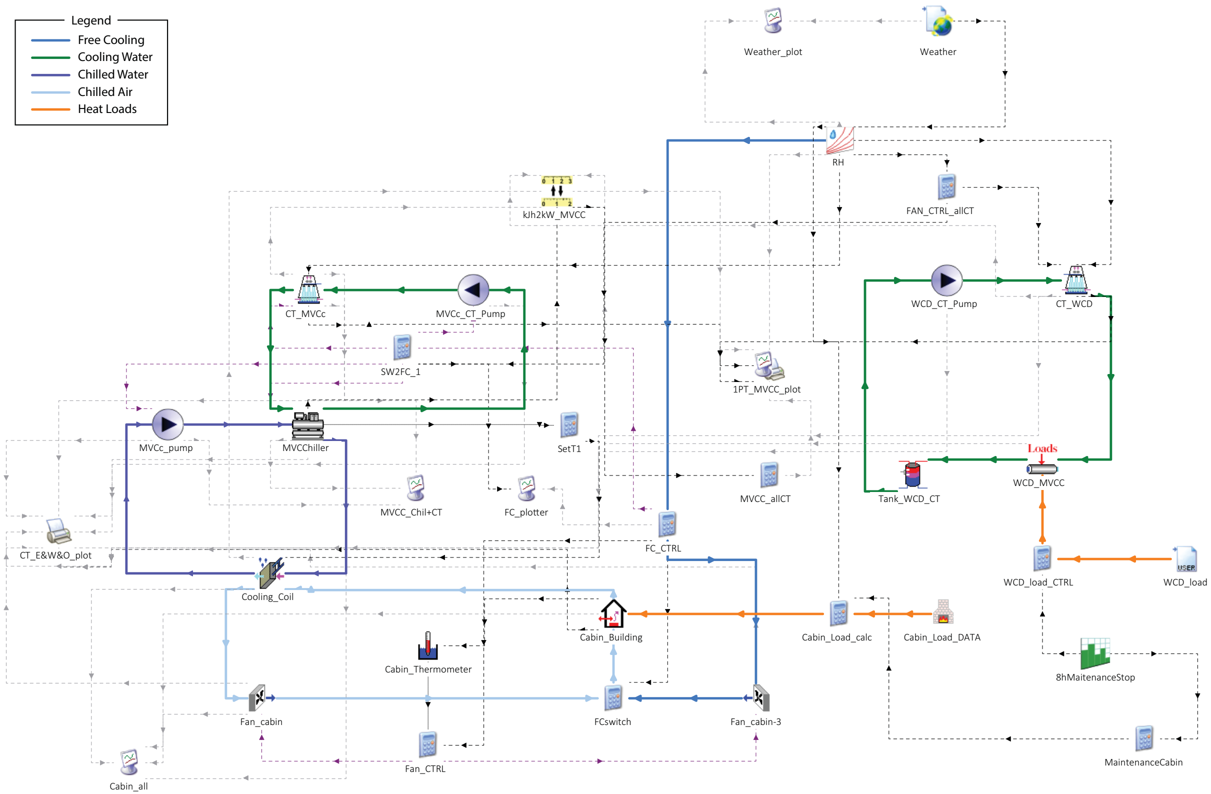


FIGURE C.4: TRNSYS deck of MVC+FC with CT.

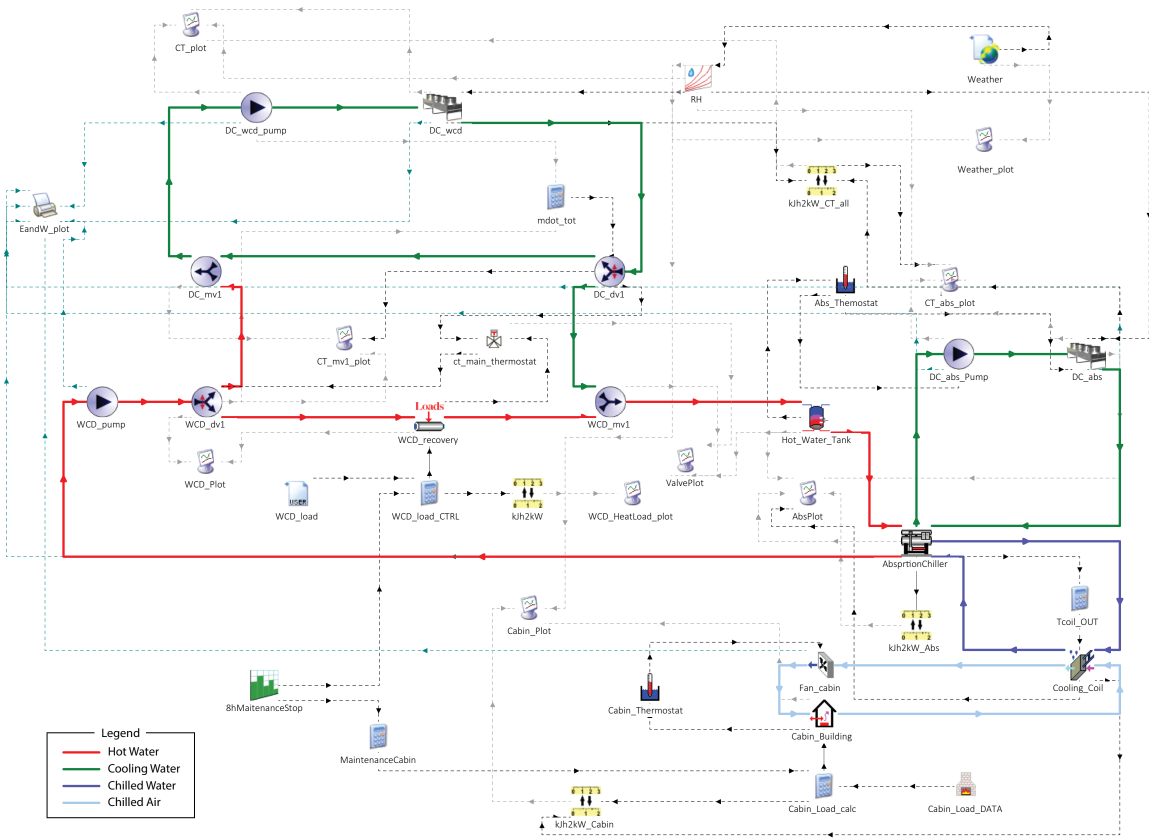


FIGURE C.5: TRNSYS deck of ABS with DC.

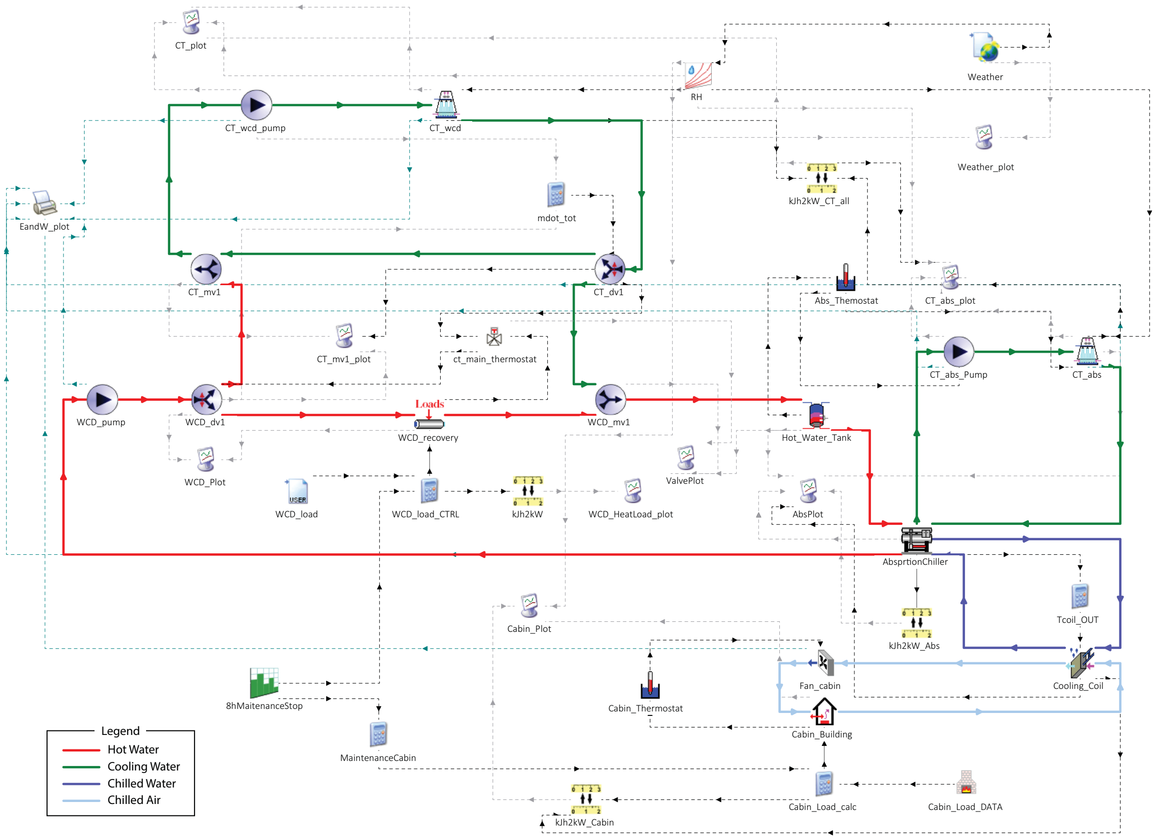


FIGURE C.6: TRNSYS deck of ABS with CT.

C.1 Types used

TABLE C.1: TRNSYS *types* used for the simulation of configurations proposed in case study *C1b*.

Simulated component	type used	subcategory
Mechanical V. C. chiller	666	
Cooling tower	51	*b*
Pump	3	*b*
Weather and Climate	15	*3*
Psychrometrics	33	*e*
Tank	60	*d*
WCD	682	
Coil	508	*b*
Fan	642	
Cabin building	88	
Thermostat	1503	
Cabin load	14	*c*
Data reader	9	*a*
Absorption chiller	107	
Thermostatic valve	953	
Valves	11	*f* and *h*
Dry Cooler	511	
Maintenance Schedule	14	*h*

C.2 Performance data files

LISTING C.1: MVC chiller performance data file.

```
5 6 7 8 9 10 !Chilled water leaving temperature (C)
16 20 25 30 35 40 !Cooling water inlet temperature (C)
1.0444 1.3400    !Capacity ratio and COP ratio at 5  C oulet CHWT and 16 C inlet CWT
1.0201 1.1889    !Capacity ratio and COP ratio at 5  C oulet CHWT and 20 C inlet CWT
0.9897 1.0784    !Capacity ratio and COP ratio at 5  C oulet CHWT and 25 C inlet CWT
0.9593 0.9633    !Capacity ratio and COP ratio at 5  C oulet CHWT and 30 C inlet CWT
0.9289 0.8663    !Capacity ratio and COP ratio at 5  C oulet CHWT and 35 C inlet CWT
0.8986 0.7805    !Capacity ratio and COP ratio at 5  C oulet CHWT and 40 C inlet CWT
1.0664 1.3626    !Capacity ratio and COP ratio at 6  C oulet CHWT and 16 C inlet CWT
1.0421 1.2340    !Capacity ratio and COP ratio at 6  C oulet CHWT and 20 C inlet CWT
1.0120 1.0987    !Capacity ratio and COP ratio at 6  C oulet CHWT and 25 C inlet CWT
0.9816 0.9836    !Capacity ratio and COP ratio at 6  C oulet CHWT and 30 C inlet CWT
```

```
0.9512 0.8843   !Capacity ratio and COP ratio at 6  C oulet CHWT and 35 C inlet CWT
0.9211 0.8009   !Capacity ratio and COP ratio at 6  C oulet CHWT and 40 C inlet CWT
1.0885 1.3874   !Capacity ratio and COP ratio at 7  C oulet CHWT and 16 C inlet CWT
1.0642 1.2566   !Capacity ratio and COP ratio at 7  C oulet CHWT and 20 C inlet CWT
1.0341 1.1213   !Capacity ratio and COP ratio at 7  C oulet CHWT and 25 C inlet CWT
1.0000 1.0000   !Capacity ratio and COP ratio at 7  C oulet CHWT and 30 C inlet CWT
0.9737 0.9046   !Capacity ratio and COP ratio at 7  C oulet CHWT and 35 C inlet CWT
0.9434 0.8189   !Capacity ratio and COP ratio at 7  C oulet CHWT and 40 C inlet CWT
1.1104 1.4100   !Capacity ratio and COP ratio at 8  C oulet CHWT and 16 C inlet CWT
1.0863 1.2792   !Capacity ratio and COP ratio at 8  C oulet CHWT and 20 C inlet CWT
1.0562 1.1415   !Capacity ratio and COP ratio at 8  C oulet CHWT and 25 C inlet CWT
1.0262 1.0242   !Capacity ratio and COP ratio at 8  C oulet CHWT and 30 C inlet CWT
0.9960 0.9227   !Capacity ratio and COP ratio at 8  C oulet CHWT and 35 C inlet CWT
0.9659 0.8370   !Capacity ratio and COP ratio at 8  C oulet CHWT and 40 C inlet CWT
1.1325 1.4326   !Capacity ratio and COP ratio at 9  C oulet CHWT and 16 C inlet CWT
1.1084 1.3017   !Capacity ratio and COP ratio at 9  C oulet CHWT and 20 C inlet CWT
1.0785 1.1618   !Capacity ratio and COP ratio at 9  C oulet CHWT and 25 C inlet CWT
1.0485 1.0445   !Capacity ratio and COP ratio at 9  C oulet CHWT and 30 C inlet CWT
1.0183 0.9430   !Capacity ratio and COP ratio at 9  C oulet CHWT and 35 C inlet CWT
0.9883 0.8550   !Capacity ratio and COP ratio at 9  C oulet CHWT and 40 C inlet CWT
1.1546 1.4551   !Capacity ratio and COP ratio at 10 C oulet CHWT and 16 C inlet CWT
1.1305 1.3242   !Capacity ratio and COP ratio at 10 C oulet CHWT and 20 C inlet CWT
1.1006 1.1844   !Capacity ratio and COP ratio at 10 C oulet CHWT and 25 C inlet CWT
1.0708 1.0648   !Capacity ratio and COP ratio at 10 C oulet CHWT and 30 C inlet CWT
1.0407 0.9610   !Capacity ratio and COP ratio at 10 C oulet CHWT and 35 C inlet CWT
1.0108 0.8731   !Capacity ratio and COP ratio at 10 C oulet CHWT and 40 C inlet CWT
```

LISTING C.2: MVC chiller part-load performance data.

```
0.00 0.25 0.5 0.75 1 ! Part Load Ratio
0.0000        ! Fraction of Full Load Power at PLR=0.00
0.2507        ! Fraction of Full Load Power at PLR=0.25
0.4975        ! Fraction of Full Load Power at PLR=0.50
0.6929        ! Fraction of Full Load Power at PLR=0.75
1.0000        ! Fraction of Full Load Power at PLR=1.00
```

LISTING C.3: ABS chiller Fraction design energy input data.

```
0.00 0.10 0.20 0.30 0.40 0.50 0.60 0.70 0.80 0.90 1.00 !Fraction of Design Load
5.556 6.111 6.667 7.222 7.778 8.889 10.000 !Chilled Water Setpoint (C)
26.667 29.444 32.222 !Entering Cooling Water Temperature (C)
108.89 111.67 113.89 115.00 116.11 !Inlet Hot Water Temperature (C) Load CHW Set ECWT
0.9917 0.0000 !Capacity and Design Energy Input Fraction at  0.0 5.556  26.667 108.89
1.0407 0.0000 !Capacity and Design Energy Input Fraction at  0.0 5.556  26.667 111.67
1.0879 0.0000 !Capacity and Design Energy Input Fraction at  0.0 5.556  26.667 113.89
1.1084 0.0000 !Capacity and Design Energy Input Fraction at  0.0 5.556  26.667 115.00
1.1309 0.0000 !Capacity and Design Energy Input Fraction at  0.0 5.556  26.667 116.11
0.8646 0.0000 !Capacity and Design Energy Input Fraction at  0.0 5.556  29.444 108.89
0.9137 0.0000 !Capacity and Design Energy Input Fraction at  0.0 5.556  29.444 111.67
0.9608 0.0000 !Capacity and Design Energy Input Fraction at  0.0 5.556  29.444 113.89
0.9834 0.0000 !Capacity and Design Energy Input Fraction at  0.0 5.556  29.444 115.00
1.0039 0.0000 !Capacity and Design Energy Input Fraction at  0.0 5.556  29.444 116.11
0.7376 0.0000 !Capacity and Design Energy Input Fraction at  0.0 5.556  32.222 108.89
0.7888 0.0000 !Capacity and Design Energy Input Fraction at  0.0 5.556  32.222 111.67
0.8359 0.0000 !Capacity and Design Energy Input Fraction at  0.0 5.556  32.222 113.89
0.8564 0.0000 !Capacity and Design Energy Input Fraction at  0.0 5.556  32.222 115.00
0.8769 0.0000 !Capacity and Design Energy Input Fraction at  0.0 5.556  32.222 116.11
1.0141 0.0000 !Capacity and Design Energy Input Fraction at  0.0 6.111  26.667 108.89
1.0633 0.0000 !Capacity and Design Energy Input Fraction at  0.0 6.111  26.667 111.67
1.1104 0.0000 !Capacity and Design Energy Input Fraction at  0.0 6.111  26.667 113.89
1.1330 0.0000 !Capacity and Design Energy Input Fraction at  0.0 6.111  26.667 115.00
1.1535 0.0000 !Capacity and Design Energy Input Fraction at  0.0 6.111  26.667 116.11
```

```
0.8871 0.0000 !Capacity and Design Energy Input Fraction at  0.0 6.111  29.444 108.89
0.9363 0.0000 !Capacity and Design Energy Input Fraction at  0.0 6.111  29.444 111.67
0.9814 0.0000 !Capacity and Design Energy Input Fraction at  0.0 6.111  29.444 113.89
1.0039 0.0000 !Capacity and Design Energy Input Fraction at  0.0 6.111  29.444 115.00
1.0264 0.0000 !Capacity and Design Energy Input Fraction at  0.0 6.111  29.444 116.11
0.7601 0.0000 !Capacity and Design Energy Input Fraction at  0.0 6.111  32.222 108.89
0.8092 0.0000 !Capacity and Design Energy Input Fraction at  0.0 6.111  32.222 111.67
0.8564 0.0000 !Capacity and Design Energy Input Fraction at  0.0 6.111  32.222 113.89
0.8769 0.0000 !Capacity and Design Energy Input Fraction at  0.0 6.111  32.222 115.00
0.8994 0.0000 !Capacity and Design Energy Input Fraction at  0.0 6.111  32.222 116.11
1.0346 0.0000 !Capacity and Design Energy Input Fraction at  0.0 6.667  26.667 108.89
1.0858 0.0000 !Capacity and Design Energy Input Fraction at  0.0 6.667  26.667 111.67
1.1330 0.0000 !Capacity and Design Energy Input Fraction at  0.0 6.667  26.667 113.89
1.1555 0.0000 !Capacity and Design Energy Input Fraction at  0.0 6.667  26.667 115.00
1.1760 0.0000 !Capacity and Design Energy Input Fraction at  0.0 6.667  26.667 116.11
0.9076 0.0000 !Capacity and Design Energy Input Fraction at  0.0 6.667  29.444 108.89
0.9568 0.0000 !Capacity and Design Energy Input Fraction at  0.0 6.667  29.444 111.67
1.0039 0.0000 !Capacity and Design Energy Input Fraction at  0.0 6.667  29.444 113.89
1.0264 0.0000 !Capacity and Design Energy Input Fraction at  0.0 6.667  29.444 115.00
1.0490 0.0000 !Capacity and Design Energy Input Fraction at  0.0 6.667  29.444 116.11
0.7806 0.0000 !Capacity and Design Energy Input Fraction at  0.0 6.667  32.222 108.89
0.8297 0.0000 !Capacity and Design Energy Input Fraction at  0.0 6.667  32.222 111.67
0.8769 0.0000 !Capacity and Design Energy Input Fraction at  0.0 6.667  32.222 113.89
0.8994 0.0000 !Capacity and Design Energy Input Fraction at  0.0 6.667  32.222 115.00
0.9199 0.0000 !Capacity and Design Energy Input Fraction at  0.0 6.667  32.222 116.11
1.0572 0.0000 !Capacity and Design Energy Input Fraction at  0.0 7.222  26.667 108.89
1.1084 0.0000 !Capacity and Design Energy Input Fraction at  0.0 7.222  26.667 111.67
1.1555 0.0000 !Capacity and Design Energy Input Fraction at  0.0 7.222  26.667 113.89
1.1781 0.0000 !Capacity and Design Energy Input Fraction at  0.0 7.222  26.667 115.00
1.2006 0.0000 !Capacity and Design Energy Input Fraction at  0.0 7.222  26.667 116.11
0.9281 0.0000 !Capacity and Design Energy Input Fraction at  0.0 7.222  29.444 108.89
0.9793 0.0000 !Capacity and Design Energy Input Fraction at  0.0 7.222  29.444 111.67
1.0264 0.0000 !Capacity and Design Energy Input Fraction at  0.0 7.222  29.444 113.89
1.0490 0.0000 !Capacity and Design Energy Input Fraction at  0.0 7.222  29.444 115.00
1.0715 0.0000 !Capacity and Design Energy Input Fraction at  0.0 7.222  29.444 116.11
0.8011 0.0000 !Capacity and Design Energy Input Fraction at  0.0 7.222  32.222 108.89
0.8502 0.0000 !Capacity and Design Energy Input Fraction at  0.0 7.222  32.222 111.67
0.8974 0.0000 !Capacity and Design Energy Input Fraction at  0.0 7.222  32.222 113.89
0.9199 0.0000 !Capacity and Design Energy Input Fraction at  0.0 7.222  32.222 115.00
0.9425 0.0000 !Capacity and Design Energy Input Fraction at  0.0 7.222  32.222 116.11
1.0797 0.0000 !Capacity and Design Energy Input Fraction at  0.0 7.778  26.667 108.89
1.1309 0.0000 !Capacity and Design Energy Input Fraction at  0.0 7.778  26.667 111.67
1.1801 0.0000 !Capacity and Design Energy Input Fraction at  0.0 7.778  26.667 113.89
1.2027 0.0000 !Capacity and Design Energy Input Fraction at  0.0 7.778  26.667 115.00
1.2232 0.0000 !Capacity and Design Energy Input Fraction at  0.0 7.778  26.667 116.11
0.9506 0.0000 !Capacity and Design Energy Input Fraction at  0.0 7.778  29.444 108.89
1.0019 0.0000 !Capacity and Design Energy Input Fraction at  0.0 7.778  29.444 111.67
1.0490 0.0000 !Capacity and Design Energy Input Fraction at  0.0 7.778  29.444 113.89
1.0715 0.0000 !Capacity and Design Energy Input Fraction at  0.0 7.778  29.444 115.00
1.0941 0.0000 !Capacity and Design Energy Input Fraction at  0.0 7.778  29.444 116.11
0.8216 0.0000 !Capacity and Design Energy Input Fraction at  0.0 7.778  32.222 108.89
0.8728 0.0000 !Capacity and Design Energy Input Fraction at  0.0 7.778  32.222 111.67
0.9199 0.0000 !Capacity and Design Energy Input Fraction at  0.0 7.778  32.222 113.89
0.9425 0.0000 !Capacity and Design Energy Input Fraction at  0.0 7.778  32.222 115.00
0.9629 0.0000 !Capacity and Design Energy Input Fraction at  0.0 7.778  32.222 116.11
1.1248 0.0000 !Capacity and Design Energy Input Fraction at  0.0 8.889  26.667 108.89
1.1781 0.0000 !Capacity and Design Energy Input Fraction at  0.0 8.889  26.667 111.67
1.2252 0.0000 !Capacity and Design Energy Input Fraction at  0.0 8.889  26.667 113.89
1.2498 0.0000 !Capacity and Design Energy Input Fraction at  0.0 8.889  26.667 115.00
1.2722 0.0000 !Capacity and Design Energy Input Fraction at  0.0 8.889  26.667 116.11
0.9937 0.0000 !Capacity and Design Energy Input Fraction at  0.0 8.889  29.444 108.89
1.0449 0.0000 !Capacity and Design Energy Input Fraction at  0.0 8.889  29.444 111.67
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0.9629 0.0000 !Capacity and Design Energy Input Fraction at  0.0 8.889  32.222 113.89
0.9854 0.0000 !Capacity and Design Energy Input Fraction at  0.0 8.889  32.222 115.00
1.0080 0.0000 !Capacity and Design Energy Input Fraction at  0.0 8.889  32.222 116.11
1.1719 0.0000 !Capacity and Design Energy Input Fraction at  0.0 10.00  26.667 108.89
1.2252 0.0000 !Capacity and Design Energy Input Fraction at  0.0 10.00  26.667 111.67
```

```
1.2744 0.0000 !Capacity and Design Energy Input Fraction at  0.0 10.00  26.667 113.89
1.2948 0.0000 !Capacity and Design Energy Input Fraction at  0.0 10.00  26.667 115.00
1.2948 0.0000 !Capacity and Design Energy Input Fraction at  0.0 10.00  26.667 116.11
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1.0899 0.0000 !Capacity and Design Energy Input Fraction at  0.0 10.00  29.444 111.67
1.1391 0.0000 !Capacity and Design Energy Input Fraction at  0.0 10.00  29.444 113.89
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1.1842 0.0000 !Capacity and Design Energy Input Fraction at  0.0 10.00  29.444 116.11
0.9055 0.0000 !Capacity and Design Energy Input Fraction at  0.0 10.00  32.222 108.89
0.9568 0.0000 !Capacity and Design Energy Input Fraction at  0.0 10.00  32.222 111.67
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1.0285 0.0000 !Capacity and Design Energy Input Fraction at  0.0 10.00  32.222 115.00
1.0510 0.0000 !Capacity and Design Energy Input Fraction at  0.0 10.00  32.222 116.11
0.9917 0.0994 !Capacity and Design Energy Input Fraction at  0.1 5.556  26.667 108.89
1.0407 0.0994 !Capacity and Design Energy Input Fraction at  0.1 5.556  26.667 111.67
1.0879 0.0994 !Capacity and Design Energy Input Fraction at  0.1 5.556  26.667 113.89
1.1084 0.0994 !Capacity and Design Energy Input Fraction at  0.1 5.556  26.667 115.00
1.1309 0.0994 !Capacity and Design Energy Input Fraction at  0.1 5.556  26.667 116.11
0.8646 0.1024 !Capacity and Design Energy Input Fraction at  0.1 5.556  29.444 108.89
0.9137 0.1024 !Capacity and Design Energy Input Fraction at  0.1 5.556  29.444 111.67
0.9608 0.1024 !Capacity and Design Energy Input Fraction at  0.1 5.556  29.444 113.89
0.9834 0.1024 !Capacity and Design Energy Input Fraction at  0.1 5.556  29.444 115.00
1.0039 0.1024 !Capacity and Design Energy Input Fraction at  0.1 5.556  29.444 116.11
0.7376 0.1044 !Capacity and Design Energy Input Fraction at  0.1 5.556  32.222 108.89
0.7888 0.1044 !Capacity and Design Energy Input Fraction at  0.1 5.556  32.222 111.67
0.8359 0.1044 !Capacity and Design Energy Input Fraction at  0.1 5.556  32.222 113.89
0.8564 0.1044 !Capacity and Design Energy Input Fraction at  0.1 5.556  32.222 115.00
0.8769 0.1044 !Capacity and Design Energy Input Fraction at  0.1 5.556  32.222 116.11
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0.8994 0.1034 !Capacity and Design Energy Input Fraction at  0.1 6.667  32.222 115.00
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0.9199 0.3102 !Capacity and Design Energy Input Fraction at  0.3 6.667  32.222 116.11
1.0572 0.2921 !Capacity and Design Energy Input Fraction at  0.3 7.222  26.667 108.89
1.1084 0.2921 !Capacity and Design Energy Input Fraction at  0.3 7.222  26.667 111.67
1.1555 0.2921 !Capacity and Design Energy Input Fraction at  0.3 7.222  26.667 113.89
1.1781 0.2921 !Capacity and Design Energy Input Fraction at  0.3 7.222  26.667 115.00
1.2006 0.2921 !Capacity and Design Energy Input Fraction at  0.3 7.222  26.667 116.11
0.9281 0.2982 !Capacity and Design Energy Input Fraction at  0.3 7.222  29.444 108.89
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1.0490 0.2982 !Capacity and Design Energy Input Fraction at  0.3 7.222  29.444 115.00
1.0715 0.2982 !Capacity and Design Energy Input Fraction at  0.3 7.222  29.444 116.11
0.8011 0.3072 !Capacity and Design Energy Input Fraction at  0.3 7.222  32.222 108.89
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0.8974 0.3072 !Capacity and Design Energy Input Fraction at  0.3 7.222  32.222 113.89
0.9199 0.3072 !Capacity and Design Energy Input Fraction at  0.3 7.222  32.222 115.00
0.9425 0.3072 !Capacity and Design Energy Input Fraction at  0.3 7.222  32.222 116.11
1.0797 0.2891 !Capacity and Design Energy Input Fraction at  0.3 7.778  26.667 108.89
1.1309 0.2891 !Capacity and Design Energy Input Fraction at  0.3 7.778  26.667 111.67
1.1801 0.2891 !Capacity and Design Energy Input Fraction at  0.3 7.778  26.667 113.89
1.2027 0.2891 !Capacity and Design Energy Input Fraction at  0.3 7.778  26.667 115.00
1.2232 0.2891 !Capacity and Design Energy Input Fraction at  0.3 7.778  26.667 116.11
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1.0715 0.2951 !Capacity and Design Energy Input Fraction at  0.3 7.778  29.444 115.00
1.0941 0.2951 !Capacity and Design Energy Input Fraction at  0.3 7.778  29.444 116.11
0.8216 0.3042 !Capacity and Design Energy Input Fraction at  0.3 7.778  32.222 108.89
0.8728 0.3042 !Capacity and Design Energy Input Fraction at  0.3 7.778  32.222 111.67
0.9199 0.3042 !Capacity and Design Energy Input Fraction at  0.3 7.778  32.222 113.89
0.9425 0.3042 !Capacity and Design Energy Input Fraction at  0.3 7.778  32.222 115.00
0.9629 0.3042 !Capacity and Design Energy Input Fraction at  0.3 7.778  32.222 116.11
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1.1781 0.2861 !Capacity and Design Energy Input Fraction at  0.3 8.889  26.667 111.67
1.2252 0.2861 !Capacity and Design Energy Input Fraction at  0.3 8.889  26.667 113.89
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0.9137 0.3012 !Capacity and Design Energy Input Fraction at  0.3 8.889  32.222 111.67
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1.0858 0.3935 !Capacity and Design Energy Input Fraction at  0.4 6.667  26.667 111.67
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0.8994 0.4136 !Capacity and Design Energy Input Fraction at  0.4 6.667  32.222 115.00
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0.9199 0.4096 !Capacity and Design Energy Input Fraction at  0.4 7.222  32.222 115.00
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1.1309 0.3855 !Capacity and Design Energy Input Fraction at  0.4 7.778  26.667 111.67
1.1801 0.3855 !Capacity and Design Energy Input Fraction at  0.4 7.778  26.667 113.89
1.2027 0.3855 !Capacity and Design Energy Input Fraction at  0.4 7.778  26.667 115.00
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0.9425 0.4056 !Capacity and Design Energy Input Fraction at  0.4 7.778  32.222 115.00
0.9629 0.4056 !Capacity and Design Energy Input Fraction at  0.4 7.778  32.222 116.11
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1.2252 0.3815 !Capacity and Design Energy Input Fraction at  0.4 8.889  26.667 113.89
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1.2722 0.3815 !Capacity and Design Energy Input Fraction at  0.4 8.889  26.667 116.11
0.9937 0.3895 !Capacity and Design Energy Input Fraction at  0.4 8.889  29.444 108.89
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1.1165 0.3895 !Capacity and Design Energy Input Fraction at  0.4 8.889  29.444 115.00
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0.9137 0.5120 !Capacity and Design Energy Input Fraction at  0.5 5.556  29.444 111.67
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0.8769 0.5220 !Capacity and Design Energy Input Fraction at  0.5 6.111  32.222 115.00
0.8994 0.5220 !Capacity and Design Energy Input Fraction at  0.5 6.111  32.222 116.11
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0.9199 0.5170 !Capacity and Design Energy Input Fraction at  0.5 6.667  32.222 116.11
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1.0715 0.4969 !Capacity and Design Energy Input Fraction at  0.5 7.222  29.444 116.11
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0.9425 0.5120 !Capacity and Design Energy Input Fraction at  0.5 7.222  32.222 116.11
1.0797 0.4819 !Capacity and Design Energy Input Fraction at  0.5 7.778  26.667 108.89
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1.0941 0.4919 !Capacity and Design Energy Input Fraction at  0.5 7.778  29.444 116.11
0.8216 0.5070 !Capacity and Design Energy Input Fraction at  0.5 7.778  32.222 108.89
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0.9425 0.5070 !Capacity and Design Energy Input Fraction at  0.5 7.778  32.222 115.00
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1.2498 0.4769 !Capacity and Design Energy Input Fraction at  0.5 8.889  26.667 115.00
1.2722 0.4769 !Capacity and Design Energy Input Fraction at  0.5 8.889  26.667 116.11
0.9937 0.4869 !Capacity and Design Energy Input Fraction at  0.5 8.889  29.444 108.89
1.0449 0.4869 !Capacity and Design Energy Input Fraction at  0.5 8.889  29.444 111.67
1.0941 0.4869 !Capacity and Design Energy Input Fraction at  0.5 8.889  29.444 113.89
1.1165 0.4869 !Capacity and Design Energy Input Fraction at  0.5 8.889  29.444 115.00
1.1391 0.4869 !Capacity and Design Energy Input Fraction at  0.5 8.889  29.444 116.11
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0.9137 0.5020 !Capacity and Design Energy Input Fraction at  0.5 8.889  32.222 111.67
0.9629 0.5020 !Capacity and Design Energy Input Fraction at  0.5 8.889  32.222 113.89
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1.2744 0.4718 !Capacity and Design Energy Input Fraction at  0.5 10.00  26.667 113.89
1.2948 0.4718 !Capacity and Design Energy Input Fraction at  0.5 10.00  26.667 115.00
1.2948 0.4718 !Capacity and Design Energy Input Fraction at  0.5 10.00  26.667 116.11
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1.1391 0.4819 !Capacity and Design Energy Input Fraction at  0.5 10.00  29.444 113.89
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1.1309 0.6957 !Capacity and Design Energy Input Fraction at  0.7 5.556  26.667 116.11
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0.9608 0.7168 !Capacity and Design Energy Input Fraction at  0.7 5.556  29.444 113.89
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0.7888 0.7308 !Capacity and Design Energy Input Fraction at  0.7 5.556  32.222 111.67
0.8359 0.7308 !Capacity and Design Energy Input Fraction at  0.7 5.556  32.222 113.89
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0.8769 0.7308 !Capacity and Design Energy Input Fraction at  0.7 6.111  32.222 115.00
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0.7888 0.8352 !Capacity and Design Energy Input Fraction at  0.8 5.556  32.222 111.67
0.8359 0.8352 !Capacity and Design Energy Input Fraction at  0.8 5.556  32.222 113.89
0.8564 0.8352 !Capacity and Design Energy Input Fraction at  0.8 5.556  32.222 115.00
0.8769 0.8352 !Capacity and Design Energy Input Fraction at  0.8 5.556  32.222 116.11
1.0141 0.7871 !Capacity and Design Energy Input Fraction at  0.8 6.111  26.667 108.89
1.0633 0.7871 !Capacity and Design Energy Input Fraction at  0.8 6.111  26.667 111.67
1.1104 0.7871 !Capacity and Design Energy Input Fraction at  0.8 6.111  26.667 113.89
1.1330 0.7871 !Capacity and Design Energy Input Fraction at  0.8 6.111  26.667 115.00
1.1535 0.7871 !Capacity and Design Energy Input Fraction at  0.8 6.111  26.667 116.11
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1.0264 0.8112 !Capacity and Design Energy Input Fraction at  0.8 6.111  29.444 116.11
0.7601 0.8352 !Capacity and Design Energy Input Fraction at  0.8 6.111  32.222 108.89
0.8092 0.8352 !Capacity and Design Energy Input Fraction at  0.8 6.111  32.222 111.67
0.8564 0.8352 !Capacity and Design Energy Input Fraction at  0.8 6.111  32.222 113.89
0.8769 0.8352 !Capacity and Design Energy Input Fraction at  0.8 6.111  32.222 115.00
0.8994 0.8352 !Capacity and Design Energy Input Fraction at  0.8 6.111  32.222 116.11
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1.1330 0.7871 !Capacity and Design Energy Input Fraction at  0.8 6.667  26.667 113.89
1.1555 0.7871 !Capacity and Design Energy Input Fraction at  0.8 6.667  26.667 115.00
1.1760 0.7871 !Capacity and Design Energy Input Fraction at  0.8 6.667  26.667 116.11
0.9076 0.8031 !Capacity and Design Energy Input Fraction at  0.8 6.667  29.444 108.89
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1.0039 0.8031 !Capacity and Design Energy Input Fraction at  0.8 6.667  29.444 113.89
1.0264 0.8031 !Capacity and Design Energy Input Fraction at  0.8 6.667  29.444 115.00
1.0490 0.8031 !Capacity and Design Energy Input Fraction at  0.8 6.667  29.444 116.11
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0.8297 0.8272 !Capacity and Design Energy Input Fraction at  0.8 6.667  32.222 111.67
0.8769 0.8272 !Capacity and Design Energy Input Fraction at  0.8 6.667  32.222 113.89
0.8994 0.8272 !Capacity and Design Energy Input Fraction at  0.8 6.667  32.222 115.00
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0.9137 0.9216 !Capacity and Design Energy Input Fraction at  0.9 5.556  29.444 111.67
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0.7376 0.9397 !Capacity and Design Energy Input Fraction at  0.9 5.556  32.222 108.89
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0.8564 0.9397 !Capacity and Design Energy Input Fraction at  0.9 5.556  32.222 115.00
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0.8297 0.9306 !Capacity and Design Energy Input Fraction at  0.9 6.667  32.222 111.67
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0.8994 0.9306 !Capacity and Design Energy Input Fraction at  0.9 6.667  32.222 115.00
0.9199 0.9306 !Capacity and Design Energy Input Fraction at  0.9 6.667  32.222 116.11
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0.9199 1.0340 !Capacity and Design Energy Input Fraction at  1.0 6.667  32.222 116.11
1.0572 0.9738 !Capacity and Design Energy Input Fraction at  1.0 7.222  26.667 108.89
1.1084 0.9738 !Capacity and Design Energy Input Fraction at  1.0 7.222  26.667 111.67
1.1555 0.9738 !Capacity and Design Energy Input Fraction at  1.0 7.222  26.667 113.89
1.1781 0.9738 !Capacity and Design Energy Input Fraction at  1.0 7.222  26.667 115.00
1.2006 0.9738 !Capacity and Design Energy Input Fraction at  1.0 7.222  26.667 116.11
```

```
0.9281 0.9939 !Capacity and Design Energy Input Fraction at  1.0 7.222  29.444 108.89
0.9793 0.9939 !Capacity and Design Energy Input Fraction at  1.0 7.222  29.444 111.67
1.0264 0.9939 !Capacity and Design Energy Input Fraction at  1.0 7.222  29.444 113.89
1.0490 0.9939 !Capacity and Design Energy Input Fraction at  1.0 7.222  29.444 115.00
1.0715 0.9939 !Capacity and Design Energy Input Fraction at  1.0 7.222  29.444 116.11
0.8011 1.0240 !Capacity and Design Energy Input Fraction at  1.0 7.222  32.222 108.89
0.8502 1.0240 !Capacity and Design Energy Input Fraction at  1.0 7.222  32.222 111.67
0.8974 1.0240 !Capacity and Design Energy Input Fraction at  1.0 7.222  32.222 113.89
0.9199 1.0240 !Capacity and Design Energy Input Fraction at  1.0 7.222  32.222 115.00
0.9425 1.0240 !Capacity and Design Energy Input Fraction at  1.0 7.222  32.222 116.11
1.0797 0.9637 !Capacity and Design Energy Input Fraction at  1.0 7.778  26.667 108.89
1.1309 0.9637 !Capacity and Design Energy Input Fraction at  1.0 7.778  26.667 111.67
1.1801 0.9637 !Capacity and Design Energy Input Fraction at  1.0 7.778  26.667 113.89
1.2027 0.9637 !Capacity and Design Energy Input Fraction at  1.0 7.778  26.667 115.00
1.2232 0.9637 !Capacity and Design Energy Input Fraction at  1.0 7.778  26.667 116.11
0.9506 0.9838 !Capacity and Design Energy Input Fraction at  1.0 7.778  29.444 108.89
1.0019 0.9838 !Capacity and Design Energy Input Fraction at  1.0 7.778  29.444 111.67
1.0490 0.9838 !Capacity and Design Energy Input Fraction at  1.0 7.778  29.444 113.89
1.0715 0.9838 !Capacity and Design Energy Input Fraction at  1.0 7.778  29.444 115.00
1.0941 0.9838 !Capacity and Design Energy Input Fraction at  1.0 7.778  29.444 116.11
0.8216 1.0139 !Capacity and Design Energy Input Fraction at  1.0 7.778  32.222 108.89
0.8728 1.0139 !Capacity and Design Energy Input Fraction at  1.0 7.778  32.222 111.67
0.9199 1.0139 !Capacity and Design Energy Input Fraction at  1.0 7.778  32.222 113.89
0.9425 1.0139 !Capacity and Design Energy Input Fraction at  1.0 7.778  32.222 115.00
0.9629 1.0139 !Capacity and Design Energy Input Fraction at  1.0 7.778  32.222 116.11
1.1248 0.9537 !Capacity and Design Energy Input Fraction at  1.0 8.889  26.667 108.89
1.1781 0.9537 !Capacity and Design Energy Input Fraction at  1.0 8.889  26.667 111.67
1.2252 0.9537 !Capacity and Design Energy Input Fraction at  1.0 8.889  26.667 113.89
1.2498 0.9537 !Capacity and Design Energy Input Fraction at  1.0 8.889  26.667 115.00
1.2722 0.9537 !Capacity and Design Energy Input Fraction at  1.0 8.889  26.667 116.11
0.9937 0.9738 !Capacity and Design Energy Input Fraction at  1.0 8.889  29.444 108.89
1.0449 0.9738 !Capacity and Design Energy Input Fraction at  1.0 8.889  29.444 111.67
1.0941 0.9738 !Capacity and Design Energy Input Fraction at  1.0 8.889  29.444 113.89
1.1165 0.9738 !Capacity and Design Energy Input Fraction at  1.0 8.889  29.444 115.00
1.1391 0.9738 !Capacity and Design Energy Input Fraction at  1.0 8.889  29.444 116.11
0.8626 1.0039 !Capacity and Design Energy Input Fraction at  1.0 8.889  32.222 108.89
0.9137 1.0039 !Capacity and Design Energy Input Fraction at  1.0 8.889  32.222 111.67
0.9629 1.0039 !Capacity and Design Energy Input Fraction at  1.0 8.889  32.222 113.89
0.9854 1.0039 !Capacity and Design Energy Input Fraction at  1.0 8.889  32.222 115.00
1.0080 1.0039 !Capacity and Design Energy Input Fraction at  1.0 8.889  32.222 116.11
1.1719 0.9437 !Capacity and Design Energy Input Fraction at  1.0 10.00  26.667 108.89
1.2252 0.9437 !Capacity and Design Energy Input Fraction at  1.0 10.00  26.667 111.67
1.2744 0.9437 !Capacity and Design Energy Input Fraction at  1.0 10.00  26.667 113.89
1.2948 0.9437 !Capacity and Design Energy Input Fraction at  1.0 10.00  26.667 115.00
1.2948 0.9437 !Capacity and Design Energy Input Fraction at  1.0 10.00  26.667 116.11
1.0387 0.9637 !Capacity and Design Energy Input Fraction at  1.0 10.00  29.444 108.89
1.0899 0.9637 !Capacity and Design Energy Input Fraction at  1.0 10.00  29.444 111.67
1.1391 0.9637 !Capacity and Design Energy Input Fraction at  1.0 10.00  29.444 113.89
1.1637 0.9637 !Capacity and Design Energy Input Fraction at  1.0 10.00  29.444 115.00
1.1842 0.9637 !Capacity and Design Energy Input Fraction at  1.0 10.00  29.444 116.11
0.9055 0.9838 !Capacity and Design Energy Input Fraction at  1.0 10.00  32.222 108.89
0.9568 0.9838 !Capacity and Design Energy Input Fraction at  1.0 10.00  32.222 111.67
1.0059 0.9838 !Capacity and Design Energy Input Fraction at  1.0 10.00  32.222 113.89
1.0285 0.9838 !Capacity and Design Energy Input Fraction at  1.0 10.00  32.222 115.00
1.0510 0.9838 !Capacity and Design Energy Input Fraction at  1.0 10.00  32.222 116.11
```

LISTING C.4: Simulation parameters.

```
*************************************************************************
*** Control cards
*************************************************************************
* START, STOP and STEP
CONSTANTS 3
START=0
STOP=8760
STEP=6/60
SIMULATION   START  STOP  STEP ! Start time End time Time step
TOLERANCES 0.001 0.001     ! Integration  Convergence
```

```
LIMITS 30 1000000 50     ! Max iterations Max warnings Trace limit
DFQ 1      ! TRNSYS numerical integration solver method
WIDTH 80     ! TRNSYS output file width, number of characters
LIST       ! NOLIST statement
      ! MAP statement
SOLVER 0 1 1    ! Solver statement Minimum relaxation factor Maximum relaxation factor
NAN_CHECK 0     ! Nan DEBUG statement
OVERWRITE_CHECK 0   ! Overwrite DEBUG statement
TIME_REPORT 0    ! disable time report
EQSOLVER 0     ! EQUATION SOLVER statement
```

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