POLITECNICO
MILANO 1863

SCUOLA DI INGEGNERIA INDUSTRIALE
E DELL'INFORMAZIONE

Energetic and economic analysis of possible solutions for chilled water production to serve a district cooling network

TESI DI LAUREA MAGISTRALE IN
ENERGY ENGINEERING
INGEGNERIA ENERGETICA

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Academic Year: 2022-23

Abstract

The aim of this study is to examine and evaluate possible solutions for the production of chilled water in a trigeneration technological plant located in Bologna, serving a network of cooling and heating. In the first part, the most important theoretical aspects regarding the chilled water production are outlined, including the study of various refrigeration cycles and technologies involved.

Subsequently, the case study is presented, describing the trigeneration plant and the network of heating and cooling, focusing on chilled water production and the refrigeration units present. The models of the three solutions chosen for this study are then outlined: a vapor compression refrigeration machine air-cooled, a vapor compression refrigeration machine water-cooled tower, and an absorption refrigeration machine cooled with tower water and powered by a natural gas cogeneration unit.

After describing the considered solutions, an energetic analysis is conducted, followed by an economic analysis.

Regarding the energetic analysis, the consumption of electricity, natural gas, and water associated with each solution is calculated based on the amount of refrigeration energy it needs to provide. For the solution involving the absorber coupled with the cogenerator, the amount of electricity produced is also evaluated.

The economic analysis follows the UNI ISO 50044 standard, assessing initial investments, operating and maintenance costs, revenues, and various economic-financial indices, such as the payback period, the NPV, the IRR and the global cost of operation.

Finally, a sensitivity analysis is performed, evaluating how the cost-effectiveness of different solutions changes with varying parameters like energy cost and energy efficiency indices.

The conclusions of the analysis indicate that the most cost-effective solution is the air-condensing refrigeration machine, both in terms of payback period and considering only operating costs, as it has a specific cost of euros per kW of refrigeration produced lower than all others. Based on these results, considerations are made regarding possible future developments for revamping or constructing a new facility in the trigeneration plant.

Key-words: Refrigeration, Absorption, Water-cooled, Air-cooled, district cooling.

Abstract in italiano

L'obiettivo dello studio è indagare e valutare la convenienza energetica ed economica di differenti modalità di produzione dell'acqua refrigerata in una centrale tecnologica di trigenerazione situata a Bologna, al servizio di una rete di teleraffrescamento e teleriscaldamento.

Inizialmente si procede con l'esposizione degli aspetti teorici fondamentali che sottendono la generazione di acqua refrigerata, approfondendo lo studio dei diversi cicli frigoriferi e tecnologie coinvolte. Poi viene presentato il caso studio, descrivendo la centrale di trigenerazione e le reti di teleriscaldamento e teleraffrescamento, portando l'attenzione sul sistema di generazione dell'acqua refrigerata esistente. Vengono descritti i modelli delle tre soluzioni che sono state scelte per questo studio: una macchina frigorifera a compressione di vapore raffreddata ad aria, una a compressione di vapore raffreddata con acqua di torre e una ad assorbimento, alimentata da un cogeneratore a gas naturale, sempre raffreddata con acqua di torre.

Successivamente si procede con l'analisi energetica ed economica.

L'analisi energetica quantifica i consumi di energia elettrica, di gas naturale e di acqua, associati a ciascuna soluzione sulla base della quantità di energia frigorifera da fornire; per la soluzione che prevede l'impiego dell'assorbitore accoppiato con il cogeneratore, viene anche valutata la quantità di energia elettrica autoprodotta.

L'analisi economica, condotta secondo la UNI ISO 50044, valuta investimenti iniziali, costi di gestione e manutenzione, ricavi e i principali indicatori economico-finanziari, come il tempo di ritorno attualizzato dell'investimento, VAN, TIR e il costo globale di esercizio. Da ultimo l'analisi di sensitività indaga come la convenienza delle diverse soluzioni si modifichi al variare di parametri come il costo dell'energia e gli indici di efficienza energetica stagionali (SEER).

I risultati dell'analisi indicano che la soluzione economicamente più conveniente risulta essere la configurazione con macchina frigorifera condensante ad aria, sia in termini di minor tempo di ritorno dell'investimento sia di minor costo globale di esercizio, poiché presenta un costo specifico di euro ogni kW frigorifero prodotto, inferiore a tutti gli altri. In conclusione, sono presentati ragionamenti in merito a possibili sviluppi futuri per un revamping o un ampliamento nella centrale di trigenerazione.

Parole chiave: Produzione acqua refrigerata, Teleraffrescamento, Condensante ad aria, Condensante ad acqua, Frigorifero ad assorbimento.

Contents

Introduction

In a context where the issue of energy efficiency and sustainability are becoming increasingly relevant, chilled water production emerges as an important element in the intertwining of performance and sustainability. The need to develop increasingly efficient practices emerges as a response to growing needs, both economically and energetically, driving progress toward solutions that harmonize performance with environmental awareness.

Within this context, the focus on the in-depth analysis of systems for the production of chilled water within thermal power plants, used to support district heating and cooling networks, is very important. In fact, in particular, this thesis will discuss the production of chilled water to meet the loads of a district cooling network located in the Navile district of Bologna, served by a trigeneration technology plant, designed and managed by the cooperative CPL Concordia.

The first chapter consists of an important theoretical overview in which the processes involved in the production of chilled water will be discussed in detail, analysing different refrigeration cycles and defining important parameters for the evaluation of different technologies.

Then in the second chapter, the Bologna district cooling network will be presented concretely. The load of this network will be considered and analysed in order to outline the real needs of the utilities that make up the district cooling network.

The three different solutions that could be put at the service of this network were then described, each in order to fully meet its needs. Two solutions have been considered that provide a steam compression cycle, in one case the machine is air-cooled, while in another case the machine is water-cooled, through a cooling tower. The third solution, however, involves the use of an absorption refrigerator cooled by a cooling tower and powered by thermal energy from the cogeneration unit.

The third chapter, on the other hand, develops the evaluation of the alternatives considered, breaking it down into two basic steps. An energy analysis is carried out first, going to assess the consumption associated with the production of chilled water and mentioning the carbon dioxide emissions associated with it. Second, an economic-financial analysis is carried out, taking into account the cost of energy and investment costs, leading to the definitions of important economic indices.

Indeed, the goal is to provide as complete and articulate a view as possible of these chilled water production solutions, thus allowing a final comparison among them.

1 State of the art in the production of refrigerated water

1.1. Vapor compression refrigerator

1.1.1. Introduction

The vapor compression refrigerator is a refrigeration cycles, mostly used for air conditioning. The main components of this system are four:

- A condenser
- An evaporator
- An expansion valve.
- A compressor

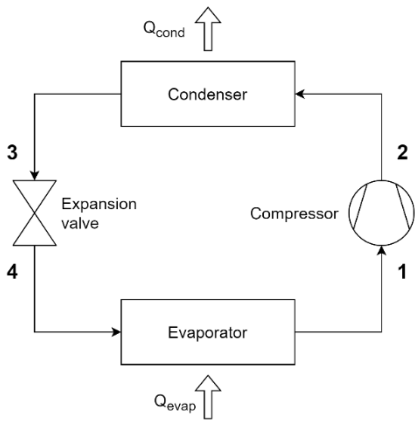


Figure 1.1 : Vapor compression cycle

Moreover, this system exchange heat with a secondary fluid, called heat source at the T_C and heat sink at the T_H.

In the field of air conditioning, considering having water as heat source water, the temperature T_C depend on the characteristic of the terminal used to condition.

The working fluid used is a circulating liquid refrigerant, which in particular undergoes four different process: the fluid enters in the compressor as a saturated vapour and then its pressure is increased thanks to the compressor, increasing also the temperature and so a superheated vapor is obtained; thanks to a secondary fluid the refrigerant undergoes a condensation in the condenser, the heat is transferred from the secondary fluid to refrigerant, which return liquid, in particular as a saturated liquid. At the end the refrigerant pass through an expansion valve where it undergoes a reduction of pressure that bring a reduction of the temperature, obtaining a two-phase refrigerant. In the last step the cold refrigerant liquid and vapor mixture goes into the evaporator, in which the secondary fluid released the heat, and the refrigerant becomes saturated vapor and returns into the compressor. It's important that all the refrigerant evaporated to not have liquid in the compressor.

The simple ideal cycle that represents this process, can be represented on a P-h diagram or on a T-s diagram, figure 1.2 and 1.3:

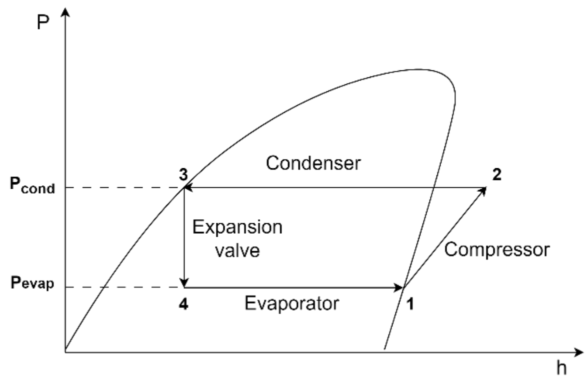


Figure 1.2: P-h diagram of simple ideal cycle

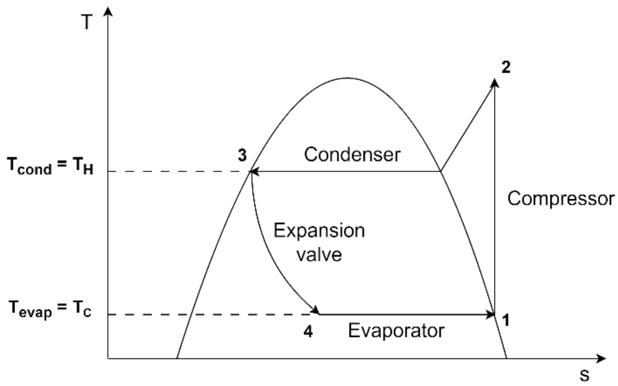


Figure 1.3: T-s diagram of simple ideal cycle

The four different states are in the following conditions, considering an ideal cycle:

1. T_1 and P_1 are the temperature and the pressure of evaporation, and the fluid is a saturated vapor.

2. The fluid is a superheated vapor and P_2 is the pressure of condensation. From the point 1 to the point 2 the entropy is constant.
3. T_3 and P_3 are the temperature and the pressure of condensation and the fluid is a saturated liquid.
4. The fluid is a two-phase and P_4 is the pressure of evaporation. From the point 3 to the point 4 the enthalpy is constant.

In reality the cycle is quite different:

- This point is usually a superheated vapor with $T_1 > T_{sat}(P_1)$ in order to avoid the presence of some liquid at the inlet of the compressor.
- A real compressor usually leads an entropy increase and so $s_2 > s_1$.
- Some pressure drops occur in the condenser and in the evaporator, when the fluid flows into them, so $P_2 > P_{3'}, > P_3$ and $P_4 > P_1$
- The liquid at the outlet of the condenser is usually a subcooled liquid so $T_3 < T_{sat}(P_3)$ for two main reasons: introducing a subcooling the efficiency increase and to avoid the formation of some bubbles at the entrance of the expansion valve, allowing the valve to function properly.
- The temperatures of the refrigerant in the evaporator and in the condenser are quite different than the temperature of the heat source (T_C) and heat sink (T_H), in particular $T_C > T_{evap}$ and $T_H < T_{cond}$

Considering these aspects the real diagrams are represented in the figure 1.4 and 1.5:

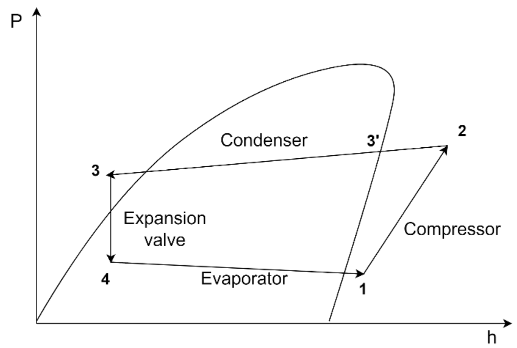


Figure 1.4 : P-h diagram of real cycle

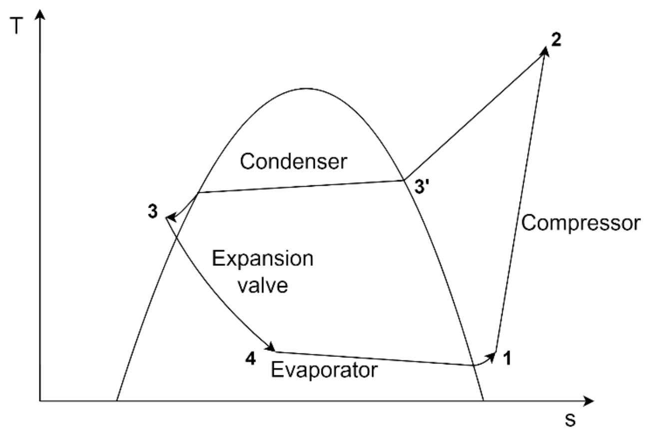


Figure 1.5 : T-s diagram of real cycle

1.1.2. Performance indexes

In a vapor compression system for a cooling system, the useful effect is the heat that the evaporator extract from the heat source Q_{evap} (cooling capacity), for example from water to obtain refrigerated water. The performance index is this case is the Energy Efficiency Ratio (EER) defines like the ratio between the useful heat and the electric work used to power the compressor W, as in formula 1.1:

$$EER = \frac{Q_{evap}}{W} \tag{1.1}$$

The efficiency and so the cooling capacity Q_{evap}, strongly depends on the temperature of condensation T_{cond} and on the temperature of evaporation T_{evap}, in particular:

- The higher is the T_{evap}, the higher is the cooling capacity and so the efficiency.
- The lower is the T_{cond}, the higher is the cooling capacity and so the efficiency.

By examining the P-h diagram in Figure 1.2, the efficiency index can alternatively be expressed in terms of the enthalpy difference, as illustrated in formula 1.2.

$$EER = \frac{h_1 - h_4}{h_2 - h_1} \tag{1.2}$$

Defining the maximum efficiency achievable by an ideal refrigeration cycle is crucial. This cycle operates with an evaporation temperature $T_{evap}\,[K]$, a condensing temperature $T_{cond}\,[K]$, and zero superheating and subcooling. The theoretical efficiency is articulated in formula 1.3.

$$EER_{TH} = \frac{T_{evap}}{T_{cond} - T_{evap}} \quad [1.3]$$

As written above, for the calculation of theoretical efficiency, subcooling and superheating are considered null, but in reality, the refrigeration cycles always work with a subcooling and superheating between 5 °C and 8 °C. In particular, the subcooling increases the efficiency by about 1 % per degree, while superheating has less influence.

In order to change from an effective value to a real one, the formula 1.4 must be used, where ε_E is the exergetic efficiency of the cycle, which depends on the type of refrigerant and η_C is the efficiency of the compressor.

$$EER = EER_{TH} * \varepsilon_E * \eta_C \quad [1.4]$$

EUROPEAN SEASONAL PERFORMANCE INDEX

The outdoor air change during the year, but the performance index considers only the temperature at the nominal conditions. Moreover, a refrigerator doesn't' work at the nominal condition all the season, but usually it work partially.

However, is more important evaluate the performance in the entire cooling season, also considering the performance of the refrigerator when it works not in the nominal condition.

For this reason, in Europe has been introduce the European Seasonal Energy Efficiency Ratio (ESEER), an index calculated by combining full and part load operating EER, for different seasonal air or water temperatures, and including for appropriate weighting factors.

The formula of the ESEER is the 1.5.

$$ESEER = \frac{C_{100\%} * EER_{100\%} + C_{75\%} * EER_{75\%} + C_{50\%} * EER_{50\%} + C_{25\%} * EER_{25\%}}{100} \quad [1.5]$$

The values to consider for the formula are reported in the table 1.1, considering the temperature of the air or water at the inlet of the condenser:

Table 1.1 : Weighting coefficients value

	Partial load ratio			
	100 %	75 %	50 %	25 %
Weighting coefficients (C)	3 %	33 %	41 %	23 %
Air temperature	35 °C	30 °C	25 °C	20 °C
Water temperature	30 °C	25 °C	20 °C	20 °C

This index can be very indicative of the real consumption of the refrigerator, and it is very important to be able to compare different types of machines.

The document that specifies the terms and definitions for the classification and performance of chillers is the UNI EN 14511-1:2022, which generally deals with "*Air conditioners, liquid chilling packages and heat pumps for space heating and cooling and process chillers, with electrically driven compressors*".

A similar index is the Seasonal efficiency Ratio (SEER) defined by the Air Conditioning, Heating, and Refrigeration Institute in the United States, and represent ratio between the cooling output during a typical cooling season divided by the total electric energy input during the same period.

1.1.3. Refrigerants

The refrigerant is the working fluid used in the vapour compression cycle and the choice of it, has an important impact on the performance of the refrigeration cycles.

The internationally recognized classification of refrigerants is based on the American standard ASHRAE Standard 34 "*Designation and safety classification of refrigerants*", in which the refrigerants are identified with the letter "R", followed by a numerical code, that gives information about the components and the type of substance.

The numbers of the code are different depending on whether the refrigerant is a pure fluid or a mixture, a natural or synthetic compound.

A **pure refrigerant** is a refrigerant that consists of only one substance, as R134a.

A **mixture refrigerant** is a refrigerant that consists of more substances, like binary or ternary mixture, considering the number of substance; considering the saturation behaviour, it can be distinguish the mixture into **zeotropic** mixture, mixture in which the saturate liquid temperature is different from the saturated vapor temperature at a given pressure, and **azeotropic** mixture, mixture in which the saturated liquid temperature is equal to the saturated vapor temperature at a given pressure.

The **natural refrigerant** is any substance that can be found spontaneously in nature, like R717 (ammonia), R290 (propane) or R744 (carbon dioxide).

The **synthetic refrigerant** is a substance that is "built" by man thought chemical process, like R134a, R410A or R32. In this case the substance is usually a hydrocarbon, where the atoms of H are removed and the F or Cl atoms are added; as a result, the 3 subgroups that can be considered are: **CFC** (chlorofluorocarbons), **HFCs** (hydrofluorocarbons) and **HCFCs** (hydrochlorofluorocarbons). This type of refrigerant can also be referred to as **F-gases**, *Fluorinated gases*.

The ASHRAE 34 is also a reference for the safe use of refrigerants fluids, taking into account the flammability and toxicity aspects. The standard used a code composed by two alphanumeric digits: the first is a capital letter (A or B), which represents the toxicity, the second is a number (1, 2, 2L, 3), which represents the flammability.

About the toxicity:

- **Class A**: the occupational exposure limit is higher than or equal to 400 pm, the refrigerant is not toxic.
- **Class B**: the occupational exposure limit is lower than 400 pm is considered toxic.

About the flammability, reference is made to the lower flammability limit (LFL) which is the lowest vapour volume concentration of the mixture below which no ignition. The following classes can be distinguished, considering the refrigerant in air at P=1 atm and T=21°C:

- **Class 1**: refrigerant with no flame propagation.
- **Class 2**: refrigerant with a lower flammability, in particular the LFL > 0.10 kg/m^3 and the heat of combustion lower than 19 kJ/kg.
- **Class 2L**: refrigerant with a lower flammability and a lower burning velocity, in particular less than or equal to 10 cm/s.
- **Class 3:** refrigerant with a higher flammability**,** in particular the LFL ≤ 0.10 kg/m^3 and the heat of combustion lower than 19 kJ/kg.

ENVIRONMENTAL IMPACT

During the normal operation of a vapor compression system, after a period of time, some refrigerant can leak out of the machine and goes to the environment. Therefore, it's essential to consider the environmental impact of the refrigerant.

Firstly, refrigerants are Ozone Depleting Substances (ODP) and for each of them is identify the *Ozone Depletion Potential* (ODP). The ODP indicates how much a substance degrades the ozone layer compared to refrigerant R-11, which has an ODP of 1,0. Since the degradation of ozone by the refrigerant comes exclusively from its chlorine atoms, it follows that a chlorine-free refrigerant (HFC) has zero ODP.

Thanks to the Montreal Protocol on *Substances that deplete the ozone layer,* signed in 1987, now the CFC, Halon and HCFC are banned.

Another environmental problem is the earth global warming and so the Intergovernmental Panel on Climate Change (IPCC) has defined the Global Warming Potential (GWP), an indicator that measures the contribution to the absorption of solar thermal radiation by a greenhouse gas over a certain period of time, compared to the absorption of an equal amount of CO2. By definition, the reference is the R744 (CO_2) to which is assigned GWP equal to 1. Only the natural refrigerants have a GWP equal to 0.

About the greenhouse effect there is also the Halocarbon Global Warming Potential (HGWP), an indicator that measures how much the refrigerant impacts on the global warming compared to the R-11, which has and HGWP equal to 1,0.

The ratio between the GWP and HGWP is about 4.000, so 1 kg of R-11 produces an effect 4000 times greater than 1 kg of CO2.

The GWP is a good indicator for the greenhouse effect but does not consider the impact that takes place in a thermal power station, upstream and downstream of the refrigerant. Therefore, the AFEAS (Environmental Acceptability Study of Alternative Fluids) has studied the Total Equivalent Warming Impact (TEWI) an index that takes into account both the direct effects, like the leakages of refrigerant in the circuit, and the indirect effects, caused by the consumption of electricity to power the plant.

Regarding the environmental impact of refrigerant gases, EU regulation 517/2014, known as the **European F-Gas Regulation**, is very important, which aims to reduce greenhouse gas emissions by 79% by the year 2030, preventing:

- a gradual reduction in the quantities of F-gases placed on the European market.
- from 2020 the ban on the use of refrigerant gases with GWP>2500 in new systems.
- Ban from 2020 on the use of refrigerants with GWP>2500 for the maintenance of installations with a refrigerant charge > 40 Ton CO2 equivalent.
- Modification of limits for loss control as a function of GWP content.

Therefore, from 2020, the restrictions and bans on new plants will be more and more restrictive and restrictions will also be set on existing plants.

Italy transposed this European legislation with DPR 146/2018 (F-Gas Decree) which entered into force on 9 January 2019.

Table 1.2 shows the most common refrigerant gases, with respective GWP values.

Table 1.2: GWP value for the most common refrigerant gases

Refrigerant	GWP
R-407C	1.774
R-410A	2.088
R-134A	1.430
R-32	675
R-404	3.922
R-427A	1.430
R-507A	3.985
R-448A	1.387
R-407F	1.824
R-422D	2.729

1.1.4. Process of condensation

An important point to study for the refrigeration machine, is the condensation process.

Condensation occurs on a surface when saturated vapor comes into contact with a temperature below its saturation point; the heat-transfer process is composed by three phases, figure 1.6:

1. Desuperheating of the hot vapor: the heat-transfer coefficient is lower, but the difference of temperature between the hot vapor and the cooling water is higher.
2. Condensation of the vapor into a liquid with release of the latent heat.
3. Subcooling of the liquid refrigerant: it occupies only a small part of the condenser's surface area.

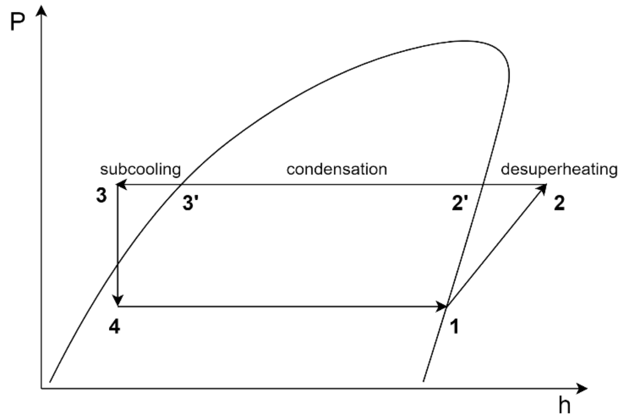


Figure 1.6 : Representation on P-h diagram of the three phases

By approximation the heat-transfer coefficient is considered to be an average value for the entire area, considering also that the refrigerant's condensation occurs at the condensing temperature.

The total heat rejection Q_{cond} $[W]$, formula 1.6, allows to define the capacity of a condenser, considering the total heat rejected during the three stages listed above.

$$Q_{cond} = 60\dot{m}_r(h_2 - h_3) \qquad [1.6]$$

The $\dot{m}_r$ is the mass flow rate of a refrigerant condenser expressed in $[kg/60s]$ and the $(h_2 - h_3)$ is the difference of enthalpy between the hot vapor that enter in the condenser and the subcooled liquid that leaves the condenser, and it is expressed in $[J/kg]$.

Another way to express the heat rejected by the condenser, formula 1.7, considers the following parameters:

- the refrigeration load at the evaporator $Q_{evap}[W]$
- the power input and the efficiency of the compressor, respectively P_{comp} $[W]$ and η_{mot}: in fact the refrigerant absorbs the heat released by the compressor and if the compressor is an hermetic one, the efficiency is lower that 1 (η_{mot} is equal to 1 for a open compressor).
- The water heat gain factor $F_{w,h}$: this factor is higher than one for the refrigerant that used chilled water like cooling medium (water-cooled condensers), because the chilled water pump power and piping heat gain should be considered.

$$Q_{cond} = \frac{F_{w,h}Q_{evap} + 2545P_{comp}}{\eta_{mot}} \qquad [1.7]$$

Another parameter that can be defined is the heat rejection factor F_{rej} or HRF, the ratio between the total heat rejected at the condenser and the load at the evaporator, formula 1.8.

$$F_{rej} = \frac{Q_{cond}}{Q_{evap}} \quad [1.8]$$

1.1.5. Different types of secondary fluid in the condenser

In the vapor compression system can be used different type of secondary fluid, which flow into the condenser, that is an indirect-contact heat exchanger in which the heat is removed from the refrigerant by a cooling medium.

Typically, like cooling medium is used the air, like the outdoor or indoor air or any substance in gas state, and the water, for example from a lake, a river or geothermal water.

In particular, a refrigerator in which flows water at the evaporator is referred to as Chillers, and considering the different cooling medium at the condenser, can be distinguished two categories:

- Air- cooled chiller, in which the air is used to as the cooling medium to condense the refrigerant.
- Water-cooled chiller, in which the water is used to condense the refrigerant, for example with the use of a cooling tower.

Another possibility is to use the ground like heat sink, in order to cool down the refrigerant that flow into the condenser.

AIR

The air, in particular the outdoor air, can be used like heat sink, in order to cool down the refrigerator that flow in the condenser.

The air is obviously available everywhere and it has a low cost.

The temperature of the air (OAT) strongly varies during the year, in according with the season, and on a daily basis. For example, considering the OAT of Milan, it can vary in range between -10°C and 40°C. Therefore, also the efficiency is largely variable and the useful effect that can be obtained changes as a function of the heat sink temperature.

The air has a very low convective heat transfer coefficient $h_{AIR} = 100 \frac{W}{m^2K} \div 200 \frac{W}{m^2K}$.

Considering the formula of the heat transferred $Q = U * A * (\Delta T)$, a low h means a very low overall transfer coefficient U, and keeping Q and A constant, it means a high ΔT.

In the case of refrigerator $\Delta T = T_{cond} - T_{air}$ and so the temperature of condensation is high, bringing a reduction of EER.

In order to move the air through the heat exchanger a fan is used and in generally the power spent is high, about the 10÷15 % of the power used in a compressor.

WATER

A river, a sea, a lake or the deep aquifer can be used as heat sink. Also, a cooling tower can be used.

The availability of the water depends on geographical place considered. Moreover, there are some bureaucratic issues:

- The maximum amount of water that can be extracted, that leads to a maximum flow rate.
- The maximum value of the difference of temperature that the water may undergo, considering that the water, after the exchange of heat, is send bake to the lake or sea, and in the T is too high, the risk is to change the environment of the lake.

The topic is different if the water source is provided through the cooling tower.

The temperature of the heat sink varies in according with the type of water source:

- For the natural reservoir like sea, river, the temperatures are in the range [5°C; 25°C]
- For the ground water the temperatures are more constant, for example about 16 °C ± 1°C for Milan.
- For the cooling tower the temperatures are in the range [25°C; 36°C]

About the heat transfer, the water has a convective heat transfer higher compared with the air: $h_{WATER} = 1000\ \frac{W}{m^2K} \div 2000\frac{W}{m^2K}$. For this reason, doing the same analysis done above, the temperature of condensation is this case is lower, more similar to the temperature of the water, and so the COP/EER is higher.

GROUND

The ground can be used as heat sink during the refrigerator operation.

In this case the system is composed by a heat exchanger put inside a borehole into the ground; some water flows into the heat exchanger and exchange heat with the refrigerant and then with the ground, releasing the heat.

The cost to build this system is high because the ground must be drilled very deep. This because in the first part of the ground the temperature varies a lot, and so, to obtain better performance, a large deep must be reached.

The use of ground as heat sink, presents a problem in long term performance. Considering the summer season, the vapor compression system has the condenser connected with the ground and so, some heat is discharged into it. After some year the thermal drift can occur, so the temperature of the ground change, due to the operation of the refrigerator. In particular the temperature of the ground increases and so the temperature of condensation, leading a decrease of the performance.

1.1.6. Air-cooled chiller

In an air-cooled condenser the air is used to remove the heat from the refrigerant, in particular the latent heat of condensation released by it. Typically, the air-cooled condensers are made by a condenser coil in which there are a main section for the condensation, a second section of the coil for the subcooling, a propellor fan to move the air and a dumper, as it showed in the figure 1.7.

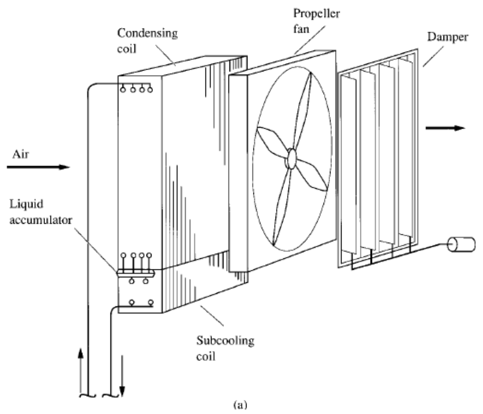


Figure 1.7 : Scheme of an air-cooled condenser

In particular, the condensing surface area of the air-cooled condenser is used for the 5 percent for the Desuperheating, 85 ÷ 90 percent for the condensation and 5 ÷ 10 percent for the subcooling. In the figure 1.8 there is an example of heat transfer diagram.

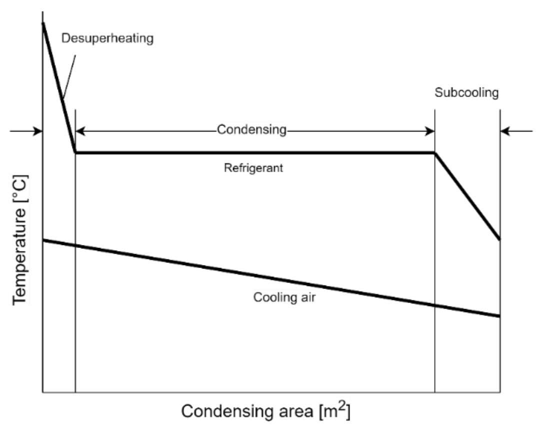


Figure 1.8 : Example of a heat transfer diagram

Considering the formula 1.7 (above), with $F_{w,h} = 1$, the total heat rejection is expressed by the formula 1.9.

$$Q_{cond} = \frac{Q_{evap} + 2545 P_{comp}}{\eta_{mot}} \qquad [1.9]$$

For a fixed Q_{rej}, the temperature rises between the air entering T_o and the air leaving T_{al} is affects by the volume flow rate of cooling air $\dot{V}_{ca}$ per unit of total heat rejection $Q_{u,rej}$. The difference $T_{al} - T_o$ is important because, if it is lower (that means that $\dot{V}_{ca}/Q_{u,rej}$ is lower), the condensing temperature T_{cond} is lower and the fans power consumption and probably the noise of them, are higher.

The total heat rejection in an air-cooled condenser is directly proportional to the condenser temperature difference CTD, formula 1.10, that is defined like the difference between the condensing temperature of the refrigerant at the pressure inlet $T_{con,i}$ and the entering air dry-bulb temperature T_0.

$$CTD = T_{con,i} - T_0 \qquad [1.10]$$

A higher condensing temperature T_{cond} brings a higher pressure and usually, and for the same location, the condensing temperature of an air-cooled is higher than the one for the water-cooled. The air-cooled condensers require a condensing temperature that is 15-20 °C over the ambient air temperature.

A higher pressure means an increase in the pressure ratio across the compressor and so an increase of the power requirement and a decrease in the compressor life. Moreover, also the efficiency of the refrigerator decreases.

However, the air-cooled condenser requires a little initial cost e a low maintenance.

EFFICIENCY OF AIR-COOLED CONDENSER

Considering a nominal operation of an air refrigeration unit (water at 7°C-12°C and air at 35°C), the condensation and evaporation temperatures are around 50 °C and 2 °C, the theoretical efficiency calculates to approximately 5,74 (utilizing formula 1.3).

$$EER_{TH} = \frac{T_{evap}}{T_{cond} - T_{evap}} = \frac{273{,}5 + 2}{50 - 2} = 5{,}74$$

Once the effects of overheating and under-cooling are taken into account, efficiency values can be elevated to about 6.

Given, for instance, a refrigeration cycle employing R707C, the exergetic efficiency is around 0,76. Assuming a compressor yield of about 0,8, the resultant real efficiency becomes 3,65 (using formula 1.4).

$$EER = EER_{TH} * \varepsilon_E * \eta_C = 6 * 0{,}76 * 0{,}8 = 3{,}65$$

In daily practice for air condensing machines the EER index that is declared also includes the electricity consumed by fans, in addition to that consumed by compressors**.**

1.1.7. Water-cooled chiller

In a water-cooled condenser the heat at the condenser is removed from the refrigerant thanks to cooling water, that is often the recirculating water from cooling tower. Also, river or lake water can be used as condenser water, and in this case an effective water filter is fundamental, in order to prevent fouling.

Considering the formula 1.7, as said above, for a water-cooled condenser, the $F_{w,h}$ is different than one; it can be equal to $F_{w,h} = 1{,}05$, for small plant-building loop or equal to $F_{w,h} = 1{,}10$ for campus-type water system.

$$Q_{cond} = \frac{F_{w,h} Q_{evap} + 2545 P_{comp}}{\eta_{mot}}$$

For air condition and refrigeration intention are commonly used as water-cooled condensers the shell-and-tube and double-tube condensers.

The last type indicated is only used in small refrigeration system, because allow a limited condensing area.

For a shell-and-tube condenser, the rate of heat transfer between the cooling water and the refrigerant Q_{cond} $[W]$, so the heat that the refrigerant rejects in the condenser, can be calculated with the formula 1.11.

$$\begin{aligned} Q_{cond} &= 60 * \dot{m}_w * c_{pw} * (T_{cl} - T_{ce}) = \\ &= 60 * \dot{m}_r * (h_{re} - h_{rl}) \quad [W] \end{aligned} \qquad [1.11]$$

Where:

- $\dot{m}_w$ is the mass flow of the cooling water $[kg/60s]$
- $\dot{m}_r$ is the mass flow rate of refrigerant $[kg/60s]$

- c_{pw} is the specific heat of the cooling water $[J/kg°C]$
- T_{ce} and T_{cl} are the condenser water entering and leaving temperatures $[°C]$
- h_{rl} and h_{re} are the enthalpy of refrigerant leaving and entering $[J/kg]$

The heat transfer can be calculated also with the NTU method, considering the condensing temperature T_{cond} $[°C]$, the NTU and the ε, as in the formulas 1.12.

$$Q_{cond} = \dot{m}_w * c_{pw} * \varepsilon * (T_{cond} - T_{ce})$$
$$\varepsilon = 1 - \exp(-NTU) \qquad [1.12]$$
$$NTU = \frac{U_o * A_o}{c_{pw}}$$

Where the U_o $[W/m^2\ °C]$ is the overall heat transfer coefficient and the A_o $[m^2]$ is the total outside surface area of the coil.

The schematic diagram of the heat transfer is showed in the figure 1.9.

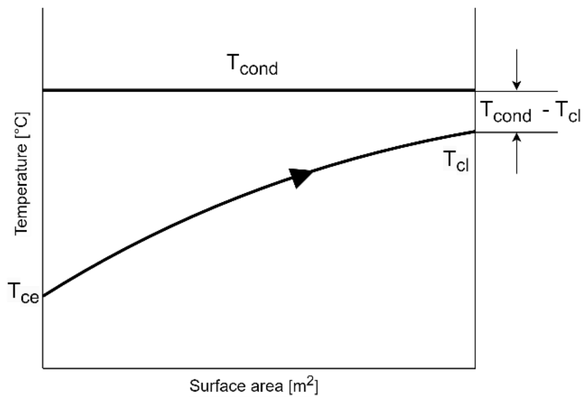


Figure 1.9 : Diagram of heat transfer

When are used cooling tower to cool the water, the temperature difference $(T_{cl} - T_{ce})$ is very important because influence the input of the compressor's power, the cooling tower fan power, and the condenser water pump power. Moreover, influence the condensing temperature T_{cond} $[°C]$.

In water-cooled condenser the condensing temperature is lower than air-cooled condenser, and so this means a lower condenser pressure and lower power requirement. Therefore, the coefficient of performance and the life of compressor increases.

Moreover, they require a lower air flow because utilize not only the sensible heat transfer but also the latent.

On the other hand, in this type of condenser the water cost, the initial cost is higher, and the water needs treatment and continuous supervision.

EFFICIENCY OF WATER-COOLED CONDENSER

Considering a nominal operation of an air refrigeration unit (7°C-12°C at the evaporator and 29,5 °C – 35 °C at the condenser), the condensation and evaporation temperatures are around 40 °C and 2 °C, the theoretical efficiency calculates to approximately 5,74 (utilizing formula 1.3).

$$EER_{TH} = \frac{T_{evap}}{T_{cond} - T_{evap}} = \frac{273{,}5 + 2}{40 - 2} = 7{,}25$$

Once the effects of overheating and under-cooling are taken into account, efficiency values can be elevated to about 7,6.

Given, for instance, a refrigeration cycle employing R707C, the exergetic efficiency is around 0,84. Assuming a compressor yield of about 0,8, the resultant real efficiency becomes 5,11 (using formula 1.4).

$$EER = EER_{TH} * \varepsilon_E * \eta_C = 7{,}6 * 0{,}84 * 0{,}8 = 5{,}11$$

1.2. Absorption refrigerator

1.2.1. Introduction

The absorption refrigerator is a thermodynamic system whose operating is based on a thermically driven cycle, instead of electrically driven cycle, like in the vapor compression system.

The absorber functions thanks a mixture composed by an absorbent and a refrigerant, that undergoes some transformation in a cyclically way, thanks to particular condition of temperature and pressure.

In particular the **absorption process** is similar to a chemical reaction between vapor and liquid: the vapor is trapped, and it is absorbed by the liquid, obtaining a final mixture in a liquid state. During the absorption some heat is released.

On the contrary, by providing heat to the mixture, the desorption process is obtained, during which steam of refrigerant is produced, separating it from the absorbent.

The absorption chiller has the advantage that doesn't affect the high electrical demand because it uses gas or a heat source to work. Moreover, if the refrigerant used is water, like in the most common cases, it brings an advantage from an environmental point of view, because the water has an ODP equal to zero.

On the other hand, the initial cost of the absorption refrigerators is higher than a chiller that use a compressor.

A solution is characterized in general by:

- **Relative volatility**, the ability of the absorber in moving from liquid to vapor state.
- **Solubility**, the ability of the refrigerant plus absorbent in making a liquid when mixed together.
- **Affinity**, the ability of the absorber in absorbing the refrigerant.

Moreover, when a mixture is considered, it's important to remember that:

- At the same pressure there is an increase in the liquid-vapor saturation temperature (evaporation T) if the concentration of the absorbent increase.
- At the same pressure there is a reduction in the solid-liquid saturation temperature (freezing T) if the concentration of the absorbent increase.

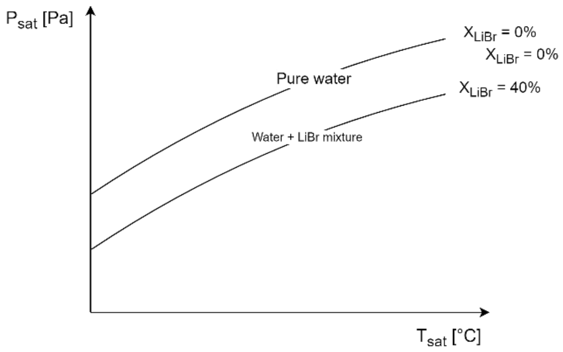


Figure 1.10 : P-T diagram of LiBr and Water mixture

1.2.2. Absorption refrigerator with aqueous lithium-bromide solution

The most common solution of refrigerant and absorber is made by:

- Water, like refrigerant
- Lithium-Bromide, like absorbent

The lithium-bromide is a salt, is not volatile and in particular it is characterized by a low solubility range and high affinity.

The composition of the solution is expressed by the mass fraction of lithium bromide, thanks to the formula 1.13, in which are considered the mass of the lithium bromide in solution m_l $[kg]$ and the mass of water in solution m_w $[kg]$.

$$X = \frac{m_l}{m_l + m_w} \qquad [1.13]$$

So, considering this formula, the mass fraction of water is $1 - X$.

Differently from a vapor compression system, an absorption system, in addition to the evaporator, condenser and expansion valve, has a thermal compressor instead of a compressor.

A thermal compressor of a single-effect cycle, is composed by:

- A generator that generates some vapor, thanks to a heat source.
- An absorber that gets the absorption of the water vapor.
- A pump, in order to increase the pressure of the liquid mixture.

So, the thermodynamic cycle can be represented in the scheme in the figure 1.11 and can be describers by the following steps and states:

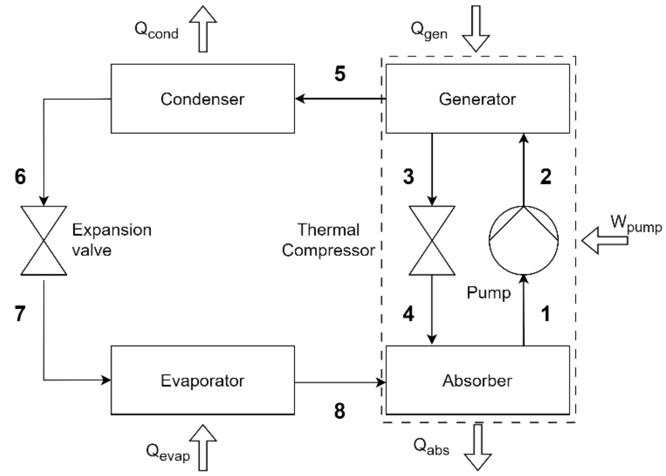


Figure 1.11: Absorption cycle

- 1 -> 2: at the end of the absorption process, that occurs in the absorber, the solution is pumped to the generator. In the point 1 the solution has a low concentration of LiBr, because it has absorbed the water; in this case is called rich solution or weak solution, because its ability to absorb additional amount of water is very low, as the water concentration is high.
- 3 -> 4: at the end of the desorption process, that occurs in the generator, the solution is send, through an expansion valve, to the absorber, to restart the cycle. At the outlet of the generator, in the point 3, the solution has a high concentration of LiBr in liquid state, because, thanks to the use of some heat Q_{gen}, some water vapor is created and sent to the condenser, instead the LiBr stay in liquid form (it is a non-volatile component). In this case the solution is called poor solution or rich solution because the concentration of water is very low, and so the ability of absorb vapor is high.
- 5 -> 6: the condensation of water vapor, produced in the generator, occurs.
- 6 -> 7: expansion of the water in liquid state
- 7 -> 8: the evaporation of the water occurs, producing the useful effect of the absorption refrigerator.

Generally, from a practical point of view, the condenser and the absorber reject heat in the same source, for example with the use of a cooling tower.

1.2.3. Performance indexes

The performance of the absorption system for the cooling operation is expressed by an energy efficiency ratio, like for the vapor compression system, in which can be considered the useful effect of the cycle, so the heat transfer at the evaporator, and the energy provided to the system in order to achieve the useful effect.

In the absorption system the energy provided is for the most, the energy provided in the generator to allow the desorption process; even if it is a value less than 3-4 order of magnitude compared to the Q_{gen} , also the power input for the pump should be added.

The performance of the absorption system is then expressed by the formula 1.14.

$$EER = \frac{Q_{evap}}{Q_{gen} + W_{pump}} \qquad [1.14]$$

The four operating temperature that influence the performance are:

- The average temperature at the generator $\bar{T}_{gen}$: it has a minimum, under of it the system can't work, because it is impossible to start the operation.
- The evaporation temperature T_{evap}
- The condensing temperature T_{cond}
- The average temperature at the absorber $\bar{T}_{abs}$

In particular the efficiency increases if the evaporation temperature increases and condensing temperature decreases.

About the other two temperature, in any absorption system must be respected the relation 1.15.

$$\bar{T}_{gen} > \bar{T}_{abs} \qquad [1.15]$$

If the $\bar{T}_{abs}$ decreases or if the $\bar{T}_{gen}$ increases, the efficiency increases.

1.2.4. Dühring plot

To describe the thermodynamic of the cycle in a diagram, are usually used the Dühring plot, figure 1.12.

Th abscissa of the plot is the saturated temperature of the mixture [$°C$]. The ordinate indicates the saturated vapor pressure of the refrigerant [kPa] (water) or also the corresponding saturated temperature [$°C$].

In the inclined lines, that are not parallel to each other, represented the mass fraction or concentration lines for the lithium bromide [%]: going from the left to the right the concentration of LiBr increases.

In the bottom part of the plot there is the crystallization line, so if state of the solution is below this line, the solidification of the LiBr salt that exceed the saturation condition occurs, creating solid crystals.

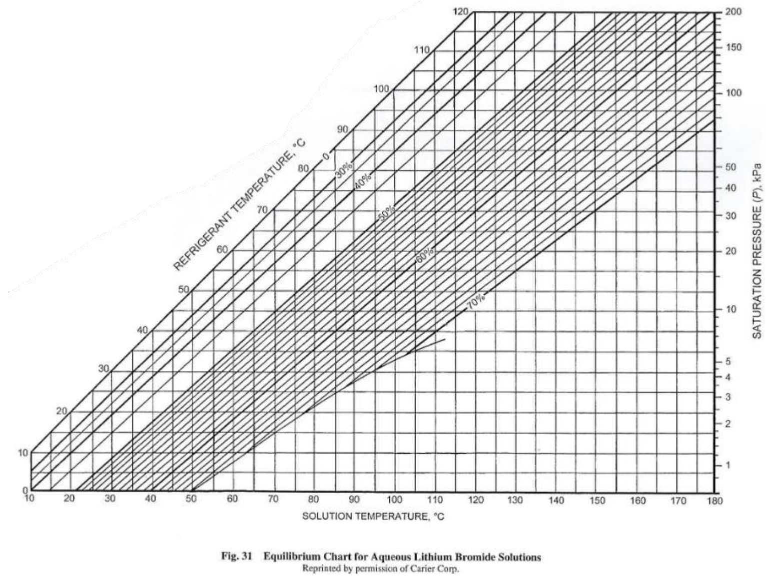


Figure 1.12 : Dühring plot of Aqueous Lithium Bromide Solutions

The process made by an absorption refrigerator, at single effect, can be drawn on the Dühring diagram, remembering than only the state of saturated liquid of the mixture can be represented in the figured 1.3, so not all the point of the figure 1.11 are presented.

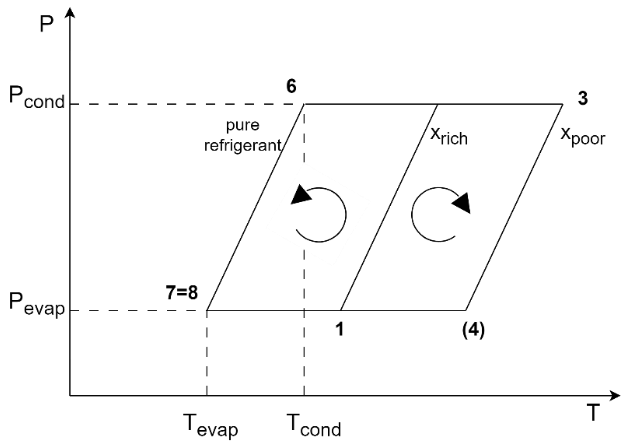


Figure 1.13 : Absorption cycle on Dühring plot

To describe this process can be considered that the evaporator has the same pressure of the absorber, the lower pressure P_{evap}, and that the condenser has the same pressure of the generator, the higher pressure P_{cond}.

In the right part of the plot is represented the mixture path, and in the left part, the refrigerant path, so the water path.

Starting from the mixture path, the points are:

- Point 1, at the outlet of the absorber, is a saturated liquid at the pressure P_{evap} and with a low concentration of LiBr, so it is rich solution.
- Point 2, after the pumping, is not in the plot because it's a subcooled liquid; the concentration is not changed.
- Point 3, at the outlet of the generator, is a saturated liquid at the pressure P_{cond} and with a high concentration of LiBr, so it is poor solution.
- Point 4, after the expansion valve, is not represented because is a two-phase mixture; the concentration is not changed.

About the water path, the points are:

- Point 5, is superheated water vapor, at the outlet of the generator, and so is not represented in the plot.
- Point 6, after the condensation, can be considered saturated liquid.
- Point 7, after the lamination, is two-phase; in this case it is represented on the plot because is pure water and not a solution.
- Point 8, at the end of the evaporation, is saturated vapor, because the superheating is not needed.

It's important that the point 4, so the mixture at the inlet of the absorber is not under the crystallization line, because if some solid crystals are formed, they can create solid blocks in the absorption system, causing a dangerous situation, figure 1.14. For this reason, the average temperature at the generator has also a maximum value below which there are crystallization, figure 1.15.

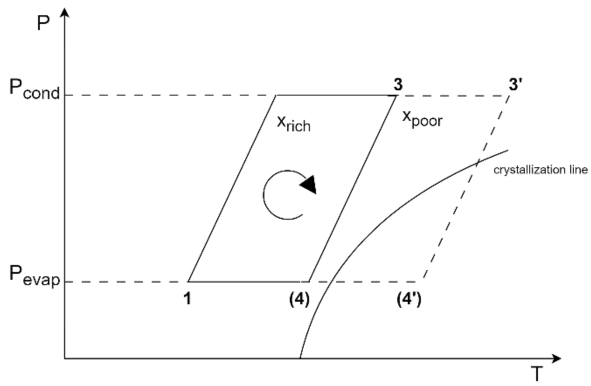


Figure 1.14 : Dühring plot with focus on crystallization

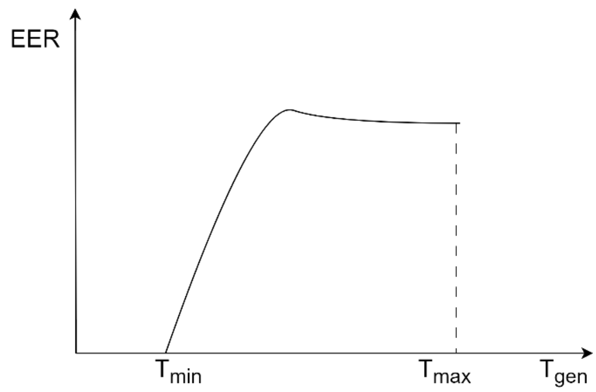


Figure 1.15 : EER as a function of generator's temperature

The value of the efficiency of an absorber system is in the range of $0{,}6 \div 0{,}75$ for the cooling.

This value is not directly comparable with the EER of the vapour compressor refrigeration treated above, because from a qualitative point of view there is a difference between the denominator of the definition of EER for a vapour compression system and an absorption system, in fact in the first it is considered electricity, while in the second a thermal energy.

In order to compare the two values, it is necessary to consider the EER values of the VCS in terms of primary energy, multiplying the index by the electrical efficiency, which according with ARERA is equal to 0,46, formulas 1.16 and 1.17.

$$EER_{water-cooled,pe} = EER_{water-cooled,pe} * 0{,}46 = 5{,}11 * 0{,}46 = 2{,}35 \qquad [1.16]$$

$$EER_{air-cooled,pe} = EER_{air-cooled,pe} * 0{,}46 = 3{,}65 * 0{,}46 = 1{,}68 \qquad [1.17]$$

1.3. Cooling towers

To condense the water-to-water refrigerator and the condenser and absorber of the absorption system, can be used as the final thermal source, the air cooling the water through a cooling tower or through a dry-cooler.

The difference between the two solutions is the temperature that the water can reach:

- **cooling towers** allow to reach water outlet temperatures of 4-5 °C above the wet bulb temperature.
- **dry coolers** allow to reach water outlet temperatures of 10-14 °C above the dry bulb temperature.

A cooling tower is a direct contact heat exchanger which is exclusively used to cool the hot stream that flow in the condenser of the refrigerator. It's largely used in HVAC system or in industry.

The dry cooler, figure 1.16, is a unit consisting mainly of an air/water heat exchanger that allows cooling a flow of water, which flows through the pipes of the exchange battery, through the use of ambient air. It is a very simple and easy to install system.

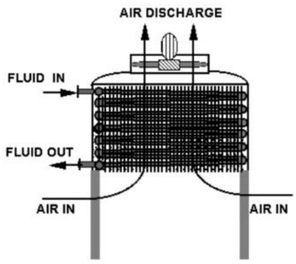


Figure 1.16 : Scheme of a dry cooler

The only energy engaged by these systems is that used by the fans that move the air lapping the pipes of the heat exchange battery.

When the external temperature or the thermal demand is reduced compared to the design conditions, the fans, if they are electronically controlled, can do modulation. modulation allows to lower the absorbed energy, modulating the rotation speed of the fan, thus increasing the economic saving and, at the same time, reducing the noise.

The initial cost and energy consumption of the dry coolers fan is high and the required floor area can be quite high.

In addition, these devices can economically cool the water to within about 11 K the dry bulb ambient temperature, which is too high a temperature for the cooling water requirement of most cold-water production plants.

For these reasons the cooling towers will be the only topic discussed in this paragraph.

1.3.1. Principle of operation

The hot water is cooled by a cooling tower through a combination of heat and mass transfer. Spay nozzles distribute the water in the tower, form the top part, through a packing fill, while the air passes upwards.

In the packing fill the largest part of heat and mass transfer occurs because there are many small and narrow channels, placed horizontally, that retard the falling water and so the contact surface area between the water to be cooled and the atmospheric air increases, increasing also the performance of the cooling tower. An example of this element is a splash-type fill, as showed in the figure 1.17.

Figure 1.17 : Splash-type fill packing

While the water passes through the filling material, some of it evaporates which causes the cooling of the remaining water: this is the **evaporative cooling** effect. The cooled water that didn’t evaporate falls into the base of tower due to gravity and here it is collected and send to the condenser. At the same time the air continues its path upwards and then exits through the top of the tower.

The air that left the tower is moisture, specifically it is the entrained water molecules that evaporating within in the cooling tower and carried by the air stream and so the water losses associated to the evaporation and drift can be quite large: in order to reduce these losses there is a drift eliminator, which consist of parallels blades arranged on the air discharge side of the tower to remove and drain water droplets from the airstream. This process is illustrated in figure 1.18.

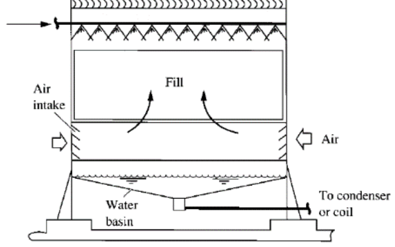


Figure 1.18 : Schematic process of evaporative cooling

The phenomenon of evaporative cooling depends on the absorption capacity of water by unsaturated atmospheric air. The maximum cooling limit temperature is equal to the wet bulb temperature of the air, which in practice can never be obtained. For this reason, the thermal performance of a cooling tower strongly depends on the entering air wet-bulb temperature.

A relation between the water and the air inside a cooling tower, in particular a counter flow cooling tower, is represented in the figure 1.19.

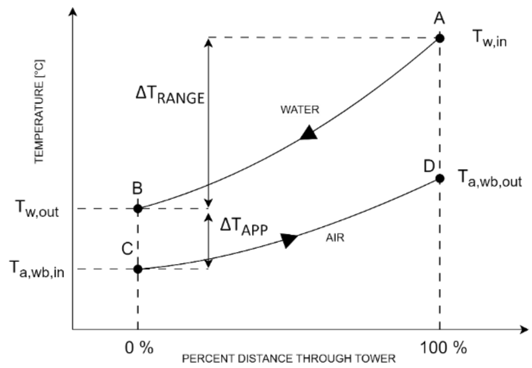


Figure 1.19 : Relation between the water and the air temperature inside a counterflow cooling tower.

The water goes from the point A to the point B, decreasing is temperature, and the air goes from the point C to the point D, increasing is temperature. The point C represent the entering wet-bulb temperature of the air.

In the graphic can be distinguish some important operating parameters, that affect the performance and the size of the tower:

- The **wet-bulb temperature** of the air $T_{A,WB,in}$
- The **range** is defined, formula 1.18, like the difference between the temperature of the water at the inlet and outlet of the tower $(A - B)$, and it do not depend by the thermal capability or size of the cooling tower, but by the heat load and the water flow rate:

$$\Delta T_{RANGE} = T_{W,in} - T_{W,out}\ [K] \qquad [1.18]$$

  A high ΔT_{RANGE} means a high volume, so a big cooling tower, because the amount of heat that are needed to be transfer is high.

  The ΔT_{RANGE} is in the range of $4°C \div 20°C$:

  - The upper limit is mandatory in order to not have a high evaporation, as a consequence of a high cooling.

- The lower limit is for a practical reason, that means, in order to cool the water, the difference between the water inlet and the outlet temperature must be significant.

▪ The **approach** is defined, formula 1.19, as the difference between the outlet of the water and the inlet of the air ($B - C$) and it depends by the cooling tower capability; for example, for a fixed thermal load, flow rate and entering air condition, a larger cooling tower produces a closer approach, which means a cooler outlet water.

$$\Delta T_{APP} = T_{W,out} - T_{A,WB,in}\ [K] \quad [1.19]$$

A low ΔT_{APP} means a high volume because the driving force for cool the water is reduced and so the volume must be higher.

The ΔT_{APP} is in the range of $2°C \div 15°C$:

- The upper limit is because, if the $T_{W,out}$ is significantly higher than the wet-bulb temperature of the inlet air, a dry-cooler may be more convenient to use.
- Below the lower limit the cooling tower would be very big and not economically sustainable.

In the figure 1.20 there is the psychrometric chart that showed the air passing through the cooling tower.

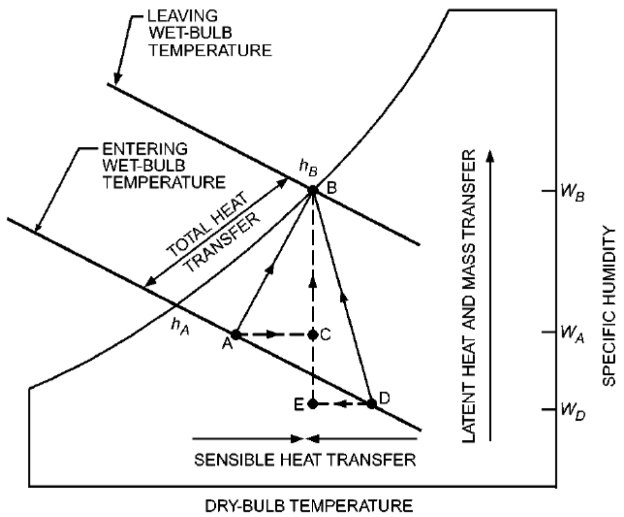


Figure 1.20 : Psychrometric analysis of air passing through cooling tower

The segment AB represents the transformation of the air: the air at the ambient conditions enters in the tower, it absorbs heat and moisture from the water, and exits saturated. The enthalpy and so the wet-bulb temperature of the air, changes during the transformation and the difference of enthalpy $h_B - h_A$ represents the heat transferred from the water to the air.

The segment DB represent another transformation for the air, considering than the air enters at the same wet-bulb temperature, but at a higher dry-bulb temperature.

In order to analyse the water consumption in a cooling tower, the ratio of latent to sensible heat is very important, in fact the mass transfer, so the evaporation, occurs in the latent portion and not in the sensible. For example, for the case AB, the sensible portion of heat is the component AC and the latent is the CB.

In the graph, the rate of evaporation is represented by the segment $W_B - W_A$ in the case first case, and by the segment $W_B - W_D$ in the case DB. So, in the first case the rate of evaporation is higher than the second case, because the latent heat transfer represents a bigger part of the total.

1.3.2. Cooling towers performance

To sum up, there are many factors that can affect the performance of a cooling tower and the main are:

- The outdoor wet bulb temperature of air: this is the first parameter for the thermal design of any cooling tower.
- The range
- The thermal load to be dissipated.
- The approach

In the following graphics 1.21 are showed the distribution of Wet bulb temperature for the "Comune di Bologna".

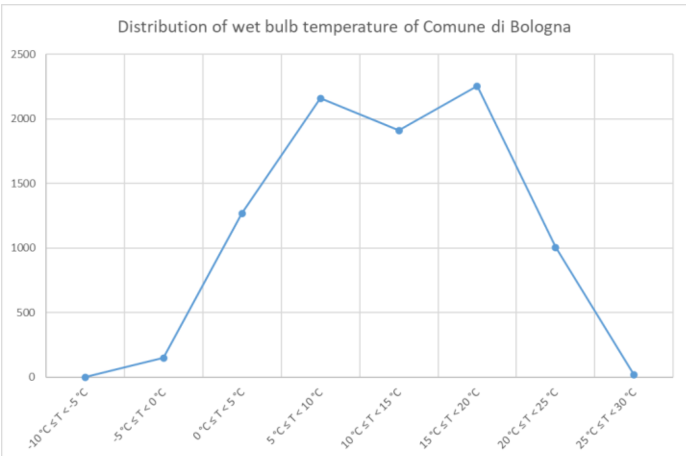


Figure 1.21: Distribution of wet bulb temperature of Comune of Bologna

1.3.3. Types of cooling towers

Cooling towers can be classified according to various criteria:

- according to the arrangement, so according to the relationship between the air flow and water flow circulation (parallel-flow, counter-flow, crossflow)
- according to the air flow circulation (natural draft or mechanical draft)
- according to water flow circuits (open circuit or closed circuit)

Considering the different arrangements, the cooling tower can be in parallel flow, in counter flow or in crossflow.

Counter-flow cooling tower is a cooling tower in which the air and the hot water flow in the opposite direction: the hot water flows from top to bottom, air flows from bottom to top.

Instead in a **Parallel-flow cooling tower** the hot water and the air flow in the same direction.

The last configuration is the **Cross-flow cooling tower** in which the hot water flows vertically from the top to the bottom of the tower, and the air flows horizontally, so it is orthogonal to the water flow.

Usually are use the counter-flow or cross-flow configuration, because in them the heat transfer is better and so the energy efficiency.

Another classification is based on the air flow circulation, in particular the two types are the natural draft cooling tower and the mechanical draft cooling towers.

In a **natural draft cooling tower**, the air flow starts thanks to the Stack effect, that means the air flow rate is induced by density variation. Inside the tower the heat transfer between the hot water flow and the cold air occurs. The air is heated and so the density variation of the air occurs, its density decreases: low density air is lighter than high density air and moves from the bottom to the top due to the gravitational effect.

In a **mechanical draft cooling tower**, the air flow rate is created by power-driven fan motors. In according with the position of the fan the draft can be:

- forced draft: in this configuration the air is forced by the fan at the base of the tower.
- induced draft: in this configuration the fan is at the discharge of the tower and the air is drawn.

The advantages of mechanical draft cooling tower are many: for example, compact structure, small volume, not affected by external wind, less loss of cooling water blowing and higher cooling efficiency.

The last classification is based on the water flow circuit, that can be open or closed.

In an **open circuit cooling tower**, figure 1.22, the hot water that has to be cooled, is in direct contact with the air.

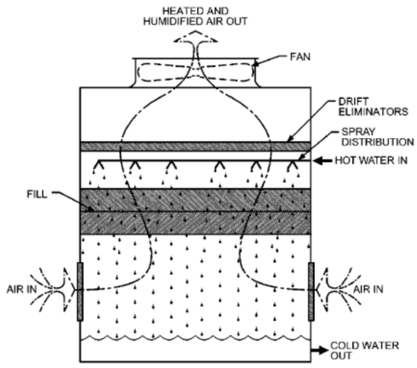


Figure 1.22: Open circuit cooling tower

Instead in a **closed circuit cooling tower**, figure 1.23, the hot water and the air used to cool it, aren't in direct contact. In this configuration there is a third fluid, an intermediate fluid which is spray over a coil and mixed with the air, while in the coil flows the hot water.

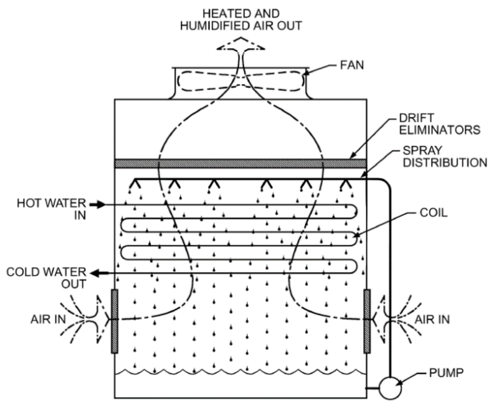


Figure 1.23: Closed circuit cooling tower

So, in this case the heat is transferred from the hot fluid in the coil to the spray water, and then to the atmosphere as a portion of the water evaporates.

This last configuration keeps the system cleaner and contaminant-free, instead of the open circuit in which any pollutant in the air goes in contact with the hot water.

1.3.4. Water losses and make-up water

During the operation of a cooling tower, an important aspect to evaluate is the consumption of water related to reintegration, called Make-Up water.

The amount of Make-Up water that are added to the circulating water system, depends on the total water losses due to:

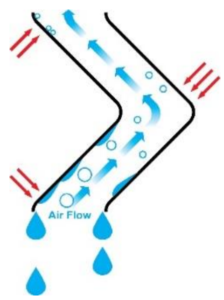
- Drift
- Evaporation
- Blowdown

WATER LOSSES DUE TO THE DRIFT

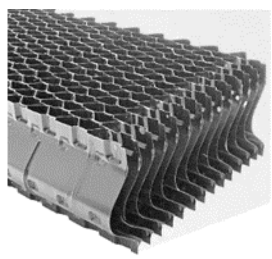
Below the spay nozzles there are a large amount of water droplets. Typically, the diameter of those droplets isn't' constant, but there is a distribution of water droplets diameter.

Considering a threshold value of diameter, the droplets with a diameter higher than the threshold fall and are collected into the basis of the tower. Instead, the droplets with a diameter lower than the threshold are carried by the air flow air and exits from the discharge of the cooling tower.

In order to reduce this phenomenon as much as possible, a **drift eliminator** is put below the fan, just before the air outlet section. The drift eliminator, figure 1.24a and 1.24b, is a component that has a large number of channels, causing a change of direction in the air flow. In this way, thanks to centrifugal force, the drops of water are separated from the air, depositing them on the eliminator surface and then they flow back into the tower.



(a) Scheme of drift eliminator



(b) Example of a drift eliminator

Figure 1.24 : Drift eliminator

Usually, an efficient eliminator design can reduce the drift water losses to a range of 0,001 to 0,2% of the water circulation rate.

Another important reason to reduce the drift is related to the fat that the composition and the quality of the drift is the one of the process waters, that flows into the condenser so is not clean water.

WATER LOSSES DUE TO EVAPORATION

In the cooling tower, during the heat transfer between the hot water, spayed on the packing fill, and the air that flows upwards, some water evaporates.

WATER LOSSES DUE TO THE BLOWDOWN

The process water that flows in the system, that is cooled in the tower, is not pure water because contains salts, particles and other impurities. During the normal operation of the cooling tower, the water evaporates, but not the impurities, that increase their concentration in the water.

The concentration of impurities contained in water must not exceed a certain limit to have a good quality of water and to not have problems in the components, so a small percentage of water is often purposely discharged: this process is called **blowdown**.

The periodicity of the purge must be in line with the composition of the make-up water, in particular its salt content.

CALCULATION OF THE MAKE-UP WATER

In order to understand the quantity of the make-up water for the right functioning of the cooling tower, the quantity of water losses due to each loss has to be calculated.

Considering the drift losses, if in the tower there is an efficient drift eliminator, the losses is around the 0,005 % of the circulating water, as said before.

To calculate the quantity of water that evaporates D_v $[m^3/h]$, can be considered:

- The enthalpy of evaporation Δh $[kJ/m^3]$, that depends on the temperature of the water.
- The quantity of heat discharge by the condenser Q_c $[kJ/h]$

So, the water that evaporates is equal to the equation 1.20.

$$D_v = \frac{Q_c}{\Delta h} \quad \left[\frac{m^3}{h}\right] \qquad [1.20]$$

About the water for the blowdown D_p, it depends on two factors: the concentration of allowable salts in circulating water (hardness) and the quality of the feed water.

To have a balance between the initial hardness of the make-up water and the maximum permissible hardness for the cooling circuit water, the graph 1.25 can be used.

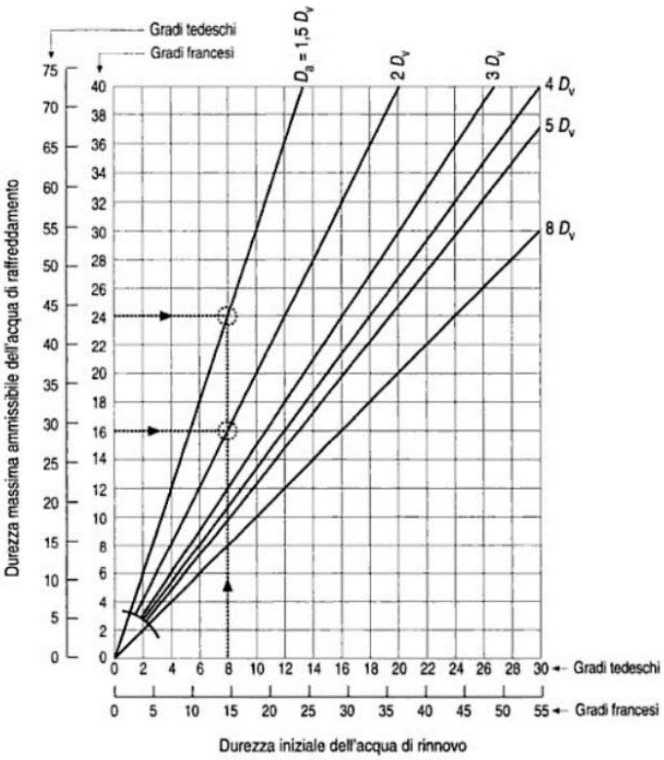


Figure 1.25: Graph for the calculation of make-up water due to blowdown

Starting from an initial hardness of the make-up (horizontal axis) water and from a maximum permissible hardness (vertical axis), at the intersection of the corresponding coordinates, the flow rate of make-up water to be provided D_a as a function of the evaporated water flow rate D_v is obtained.

So, the quantity of water for the blowdown is equal to formula 1.21.

$$D_p = D_a - D_v \left[\frac{m^3}{h}\right] \quad [1.21]$$

At the end, if a quantity of water equal to D_a has provided to the cooling tower, it can be assured that on the one hand, all the evaporated water to ensure cooling will be replaced, and, on the other, that the salt content of the circulating water will remain within reasonable limits.

Generally, the water that evaporates is in the range of 1% ÷ 2% of the circulating water and the blowdown water is in the range of 2% ÷ 3% of the circulating water.

The water hardness is expressed in French degrees (°F), where one degree represents 10 mg of calcium carbonate per liter of water.

About the hardness of the make-up water, not all the range of French grades is considered. Considering, for example, the territory of Bologna, the water used has hardness in the range of $26 \div 30\ °F$ and it is usually good practice to use a tower water with hardness of about $10 \div 15\ °F$.

It is therefore necessary add to the system, a **pre-treatment** of the water, such as **softening or osmosis processes**, to reduce its hardness and so to avoid problems like scaling, corrosions and algae and biofilm development, which lower exchange efficiency and can clog the passages.

1.3.5. Energy consumption in cooling towers

In order to calculate the total electrical energy absorbed by cooling tower for the cooling of the water, must be calculated the electric energy absorbed by the pumps and by the fans, and then use the equation 1.22.

$$En_{el} = En_{el,pump} + En_{el,fan}\ [kWh] \quad [1.22]$$

Regarding the pumping is used the equation 1.23.

$$En_{el,pump} = \sum_i f_i * h_i * P_{nom,i}\ \ [kWh] \quad [1.23]$$

Where:

- the summatory is referred to the total number of pumps.
- the factor f_i represents the real current absorbed by the i-th pump respect the nominal value.
- the number h_i indicated the real operating hours of the i-th pump.
- the $P_{nom,i}\ [kW]$ is the nominal electrical power of the i-th pump.

Regarding the fans is used the equation 1.24.

$$En_{el,fan} = \sum_i \left(\sum_j h_j * P_{nom,j} \right) [kWh] \quad [1.24]$$

Where:

- the summatory with index i is referred to the total number of towers.

- the summatory with index j is referred to the number of fans for each tower.
- the number h_j indicated the real operating hours of the j-th fan of the i-th tower.
- the $P_{nom,j}$ $[kW]$ is the nominal electrical power of the i-th fan of the i-th cooling tower.

2 Case study presentation

2.1. High efficiency trigeneration plant

2.1.1. Description of the plant

In this thesis is considered as a case study the central high efficiency trigeneration technology of a district heating and cooling system serving the "Ex-Mercato Navile" of Bologna, in figure 2.1. The power station thus guarantees the simultaneous production of hot water, for the domestic hot water and heating of the district, the cold water for the cooling and the electricity, used for the auxiliary systems of the plant or sold to the network.

Figure 2.1: View from maps of the Trigeneration plant

Today the plant serves 8 substations, at full capacity will be 17, between the streets Carracci to the south, Fioravanti to the east, Gobetti to the northeast and the railway to the west.

In the Figure 2.2 is showed a satellite image of the area served by the power plant and in the Figure 2.3 is showed the development of the network plan with highlighted utilities to date served.

Figure 2.2: Satellite image of the area served by the power plant

Figure 2.3: Development of the network

The project began in 2013, with a first construction in 2014-2015 and a second expansion in the plant in 2018-2019.

The table 2.1 shows some project data concerning the district heating (TLR) and cooling (TRF) network.

Table 2.1: Project data of district heating and cooling

	U. M.	TLR	TRF
Nominal network's power	MW	10	4
Operating temperature of the network	°C	82 ÷ 90	9 ÷ 14
Nominal pressure	bar	16	16
Effective pressure	bar	2,5 ÷ 4,5	2,5 ÷ 4,5

The thermal group for the production of thermal energy and therefore hot water, consists in a cogeneration system and two boilers with sliding temperature, as a support.

Both boilers are powered by methane gas and have a power of 1,9 MW and 1,1 MW.

Also, the cogeneration unit is powered by natural gas and is characterized by:

- an endothermic motor of 366 kW_{el} and 284 kW_{th}
- firebox capacity of 0,9 MW

- actual electrical efficiency of 37-38%
- actual thermic efficiency of 50-51%

The cogeneration system is set to heat tracking, so the engine modulates according to the heat needed by the network.

About the refrigeration plant, to produce cold water there are two chillers with a cooling capacity of 1,2 MW, which use R134a as a refrigerant. The chillers are tower-water cooled chillers. The production of chilled water will be dealt in more detail in the following paragraph. To produce cold water there is also a Lithium-bromide water absorption chiller powered by thermal energy from the cogeneration system. Also, the absorber is cooled by the cooling tower. In the year under consideration in the analysis the apparatus was not used.

The simplified structure of the system as a whole is shown in Figure 2.4.

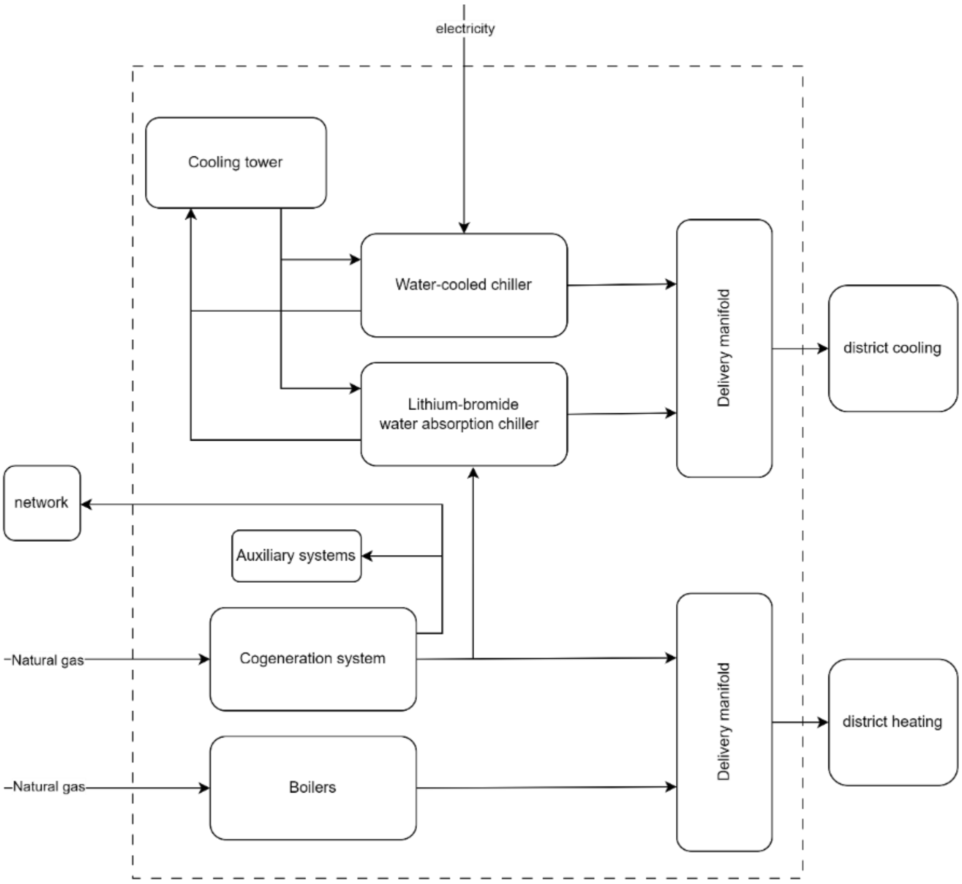


Figure 2.4: Block diagram of the plant connected to the district heating and cooling network

2.1.2. Focus on refrigeration plant

As mentioned above, for the production of cold water for the district cooling network, in the technology plant there are two refrigerator groups and an absorber.

The condensers in the machine are water-cooled and the characteristics of the two refrigerators are the following in table 2.2 and table 2.3.

Table 2.2: Chiller 01

TECHNICAL DATA CH01 BlueBox - OMEGA V ECHOS /A	
Unit size	124,2
Refrigerant	$R134A$
Main power supply	$400\ V\ /3 \sim 50\ Hz$
Cooling (EN 14511 values)	
Nominal cooling capacity	1296 kWf
EER	5,06
ESEER	5,1
Evaporator	
Water flow rate	220 mc/h
Pressure drop	6,4 m.c.a.
Condenser	
Water flow rate	263 mc/h
Pressure drop	3,5 m.c.a.

The water flow rate at the condenser is equal to: $263\ mc/h \rightarrow 73{,}06\ kg/s$

So, the heat that the cooling towers must remove from the fluid can be calculated with the formula 2.1:

$$Q_{cond,CH01} \quad = \dot{m}_{w,cond} * cp_w * \Delta T = \\ = 73{,}06 \frac{kg}{s} * 4{,}186 \frac{kJ}{kgK} * 5\ K = 1.529\ kW \qquad [2.1]$$

Table 2.3: Chiller 02

TECHNICAL DATA CH02 **Trane - RTHF-360-XE**	
Unit size	360
Refrigerant	R134A
Main power supply	400 V /3 ~50 Hz
Cooling (EN 14511 values)	
Nominal cooling capacity	1245 kWf
EER	5,63
ESEER	6,72
Evaporator	
Water flow rate	214 mc/h
Pressure drop	2,6 m.c.a.
Condenser	
Water flow rate	250 mc/h
Pressure drop	1,5 m.c.a

The water flow rate at the condenser is equal to: $250\ mc/h \rightarrow 69{,}4\ kg/s$

So, the heat that the cooling towers must remove from the fluid can be calculated with the formula 2.2:

$$Q_{cond,CH02} \quad = \dot{m}_{w,cond} * cp_w * \Delta T = \\ = 69{,}4 \frac{kg}{s} * 4{,}186 \frac{kJ}{kgK} * 5\ K = 1.453\ kW \qquad [2.2]$$

The characteristics of absorber, that is a single effect with hot water, are in table 2.4.

Table 2.4: Absorber

TECHNICAL DATA Trane – Thermax 5G 2E C	
Nominal cooling capacity	325 kW
COP	0,719
Refrigerated water circuit	
Refrigerated water flow rate	55,9 mc/hr
Pressure drop	37,4 kPa
Temperature in/out water	12°C/7°C
Water cooling circuit	
Cooling water flow rate	135,0 mc/hr
Pressure drop	53,6 kPa
Temperature in/out water	30°C/35°C
Thermal power to reject	777,1 kW
Hot water circuit	
Hot water flow rate	20,1 mc/h
Pressure drop (by friction)	7,6 kPa
Temperature in/out water	90°C/70°C
Thermal power input	452,1 kW

The water exchanging heat in the condenser is cooled by the use of two evaporative towers, placed on the roof of the plant, with the following characteristics, table 2.5 and 2.6.

Table 2.5: Cooling tower 01

TECHNICAL DATA SCAM TPE – TASF–RB10-LP-SS-LN2-VDI-S	
Nominal cooling capacity	2407 kW
Water flow rate	414 mc/h
Air flow rate	17768 mc/h
Fans consumption	2 x 7,5 kWe
Temperature in/out	35/30 °C
Main power supply	400 V /3 ~50 Hz

Table 2.6: Cooling tower 02

TECHNICAL DATA DECSA – TMR-41-319	
Nominal cooling capacity	2500 kW
Water flow rate	430 mc/h
Air flow rate	200000 mc/h
Fans consumption	2 x 30 kWe
Temperature in/out	35/30 °C
Main power supply	400 V /3 ~50 Hz

2.1.3. Energy analysis and energy carriers used

The time period that is taken into account in this thesis, on which the analysis carried out later will be based, is one year, in particular from September 2022 to August 2023.

Considering the data provided by the operators of the power plant, the district heating network requires about 3.151 MWh, while the district heating network requires about 1.405 MWh.

The energy carriers that are used in the plant to produce hot and cold water are:

- natural gas to power the burners of the boilers and the cogeneration unit.
- electricity to power the compressors of the refrigerators, for the circulation pumps and fans of the cooling towers.

Table 2.7: Data of natural gas

NATURAL GAS [Sm^3]	
PCI [1]	9,98 kWh/Sm3
Primary energy conversion [Tep][2]	1 Sm3 = 0,000825 Tep
Total primary energy factors f_p	1

Table 2.8: Data of electricity from the grid

ELECTRICITY [kWh_e]	
Primary energy conversion [Tep][3]	1 kWh = 0,000187 tep
Total primary energy factors f_p	2,174

The conversion factors into primary energy are defined at national and European level. At the national level, reference will be made to the most recent document drawn up by the *Comitato Termotecnico Italiano (CTI)* on primary energy factors, which is called Raccomandazione CTI 14: 2013.

Moreover, must be considered the consumption of the water for the reintegration of the cooling tower and, even if only in small part, for maintenance.

The total annual turnover consumption of electricity, gas and water is shown in table 2.9.

[1] The value of the PCI results from an average made on the PCI of each month of the year analysed.

[2] Source conversion factor: *All.A "Linee guida per la preparazione, esecuzione e valutazione dei progetti di cui all'art.5 comma1, del DM 20 luglio 2004"*

[3] Source conversion factor: *AEEG, Delibera EEN 3/08*

Table 2.9: Total annual consumption

Total consumption			
	Water [m3]	**Electricity [kWh]**	**Natural gas [Sm3]**
September 2022	2.333	21.121	51.484
October 2022	1.303	1.531	68.525
November 2022	710	15.695	40.553
December 2022	733	3.218	98.614
January 2023	475	165	105.624
February 2023	429	95	88.912
March 2023	0	2.658	80.076
April 2023	184	1.095	54.371
May 2023	289	8.081	49.710
June 2023	432	10.881	47.253
July 2023	475	83.493	49.414
August 2023	3.964	26.461	39.158
Total	**11.327**	**174.494**	**773.694**

The graph 2.5 represents the share of each energy carrier in total costs and it can be seen that natural gas is most expensive.

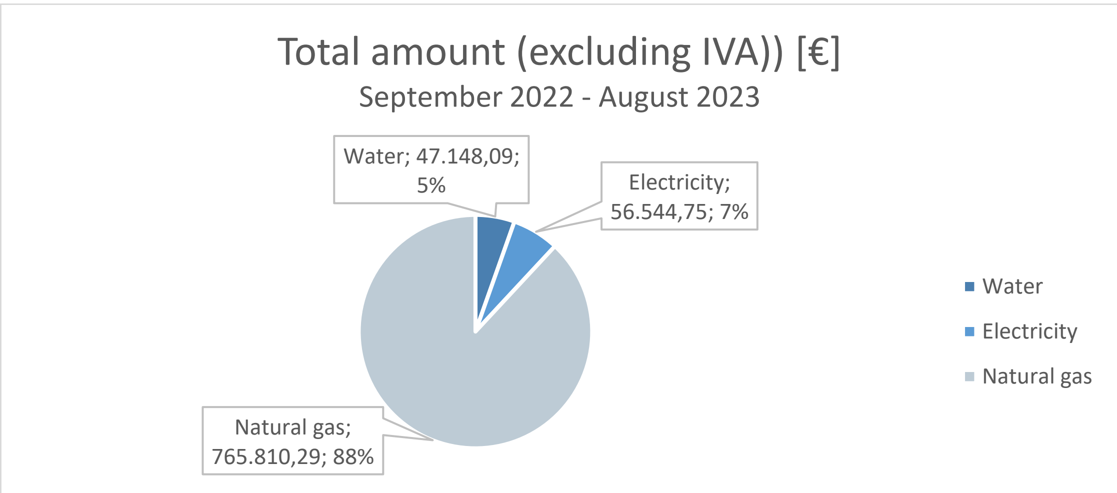


Figure 2.5 : Total amount of energy carrier

2.1.4. Focus on consumption for production of chilled water

The focus of this thesis is the production of chilled water for the district cooling, so in the tables 2.10 below are shown, in details, the needs of the network about this year and the sum of the cooling energy produced by the two chillers.

Table 2.10: Cooling production by chiller and needs of the district

Period	Cooling needs of the district [kWh$_f$]	Cooling production by the chillers [kWh$_f$]
September 2022	201.380	221.790
October 2022	84.010	107.694
November 2022	37.130	34.736
December 2022	16.830	34.930
January 2023	9.200	19.079
February 2023	8.900	19.521
March 2023	23.800	32.518
April 2023	26.500	32.746
May 2023	65.020	78.197
June 2023	219.970	267.889
July 2023	413.080	518.381
August 2023	299.520	307.939
Total	**1.405.340**	**1.675.420**

The difference between the quantity of thermal energy delivered by the chillers and that actually requested and received by the district cooling network is equal to **16%.**

It is a rather high difference that is not however attributed to the total energy losses. Usually, due to the losses, the two values differ by 5 -7%.

The additional difference may be due to many factors, such as measurement errors related to the instruments: in fact, the instrument itself has related measurement errors, to which are added those due to the installation itself. For this analysis, the cooling production by the chiller's units, and not the needs of the district cooling network, will be taken as a reference.

2.2. Different modes for the production of refrigerant water

The purpose of this thesis is to study different solutions for the production of cooling water for the district cooling.

In this section will be introduced the analysis that will be carried out in the next chapter, describing the three solutions analysed, in order to outline an energy model for each.

The description will be made through a rather simple block diagram, that it goes to identify the main members, the flows and the taken in consideration capacities.

For each machine and component, a reference identification code will be established.

A first solution with air condenser will be studied, a second with water condensate and finally a solution considering an absorption machine.

For each type of solution will be defined a size, considered in the energy and economic analysis made later.

The size of the machine and other associated features were chosen taking into account as a model a machine really present in the market, with a cooling capacity of 1 MW. For ease of calculation then some data have been approximated.

For all three solutions, having a cooling capacity of 1 MW, can be calculated, formula 2.3, the water flow rate $\dot{m}_p$ necessary at the evaporator, in the district side (called primary circuit).

$$\dot{m}_p = \frac{Q_{evap}}{c_{p,water} * \Delta T_{water}} \; [m^3/h] \tag{2.3}$$

Considering a specific heat capacity of 4,186 kJ/kgK and that the water enters in the chiller at 12°C and it is cooled until 7°C, the water flow rate is equal to:

$$\dot{m}_p = \frac{1000\ kW}{4{,}186 \frac{kJ}{kgK} * 5K} = 47{,}8 \frac{kg}{s} = 172 \frac{m^3}{h} \tag{2.3}$$

2.2.1. Air-cooled chiller

The first solution that will be described concerns the use of an air condensing machine, that is, a vapour compression chiller that, to reject heat, uses fans and so the secondary fluid at the condenser is air.

Air cooled chillers are characterized by a simple structure, easy to install and easy to maintain, compared to other types of chillers. The system condenses depending on the dry bulb temperature of the ambient.

In this thesis the machine will be identified with the code **AW-C01**, figure 2.6.

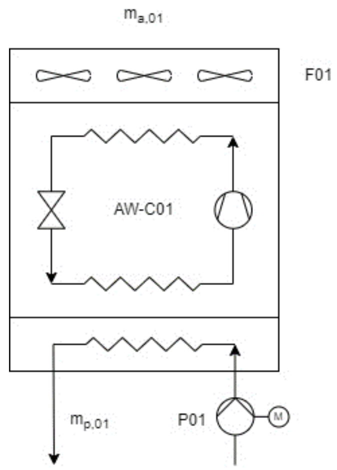


Figure 2.6: Scheme of air-cooled solution, AW-C01

In the schematic representation of the solution is draw the typical Carnot cycle of vapour compression machines.

The evaporator consists of a refrigerant-water exchanger. In particular, water is the one that will then be used in the district cooling network, and then flows into the primary circuit of the plant. The evaporative capacity is indicated with $\boldsymbol{Q_{evap,AW,C01}}$ $[\boldsymbol{MW}]$.

In the primary circuit there is the **P01** pump, controlled by an inverter, which processes a water flow $\boldsymbol{\dot{m}_{p,01}}$ $[\mathbf{m^3/h}]$.

The condenser instead is composed of a refrigerant-air exchanger: the refrigerant flows inside the machine and is cooled by the external air that is pushed against the exchanger thanks to the use of F01 fans. The air flow required to cool the refrigerant is identified by $\boldsymbol{\dot{m}_{a,01}}$ $[\mathbf{m^3/s}]$ and the heat extracted by refrigerant is $\boldsymbol{Q_{cond,AW,C01}}$ $[\boldsymbol{MW}]$.

The energy carrier to consider in this situation is only one: electricity.

Electricity consumption is related to:

- the refrigeration cycle compressor: $\boldsymbol{P_{AW,C01}}$ $[\boldsymbol{kW}]$
- the pump of the primary circuit $\boldsymbol{P_{P01}}$ $[\boldsymbol{kW}]$

- the motor of the fans in the condenser $\boldsymbol{P_{F01}}$ $[\boldsymbol{kW}]$

The electricity consumption for the compressor and the fans will be considered together, $\boldsymbol{P_{AW,C01+F01}}$ $[\boldsymbol{kW}]$. There is no water consumption in this case.

After defining the scheme of the machine, the size was chosen.

Based on the fact of wanting to have a machine that processed **1 MW** of cooling capacity, a chiller of **TRANE** was chosen, in particular an air-cooled chiller **Sintesis Prime RTAF**, figure 2.7.

Figure 2.7 : Air-cooled chiller of Trane / Sintesis Prime RTAF

Table 2.11 summarizes the main characteristics of the model chosen, which is the **RTAF 275 HSE G XLN.**

Table 2.11: Data of chiller on the market - Trane

MACHINE REFERENCE ON THE MARKET	
TRANE - RTAF 275 HSE G XLN	
Cooling capacity at evaporator [MW](4)	1,0031
EER[4]	3,16
SEER[5]	4,62
Power absorbed by the compressor and fans [kW]	317,4
Heat rejected at condenser [MW]	1,3205
Refrigerant	R1234ze(E)

[4] From EN 14511:2022 where are considered: external air temperature of 35 ºC and chilled water temperature of 12 ºC/ 7 ºC.

[5] As defined in the Ecodesign design requirements for Comfort Chillers with a maximum capacity of 2000 kW - REGULATION (EU) No. 2016/2281 of 20 December 2016.

Therefore, considering the reference machine, the air condenser chiller considered in the analysis has the data shown in table 2.12.

Table 2.12: Data of AW-C01 for the analysis

AIR – COOLED CHILLER /AW – C01	
$\mathbf{Q_{evap,AW,01}}$ [MW]	1
$\mathbf{Q_{cond,AW,C01}}$ [MW]	1,3
SEER	3,7
$\mathbf{P_{AW,C01+F01}}$ [kW]	317

The seasonal efficiency value that will be considered in the consumption analysis, is that given by the manufacturer of the reference machine reduced by a safety factor of 20%.

About the pump of the primary circuit, the characteristics are in table 2.13.

Table 2.13: Data of P01 for the analysis

PUMP P01	
$\mathbf{P_{P01}}$ [kW]	11
Pressure difference [m. c. a.]	10
Water flow rate $[\mathbf{m^3/h}]$	172

The sizing was done thanks to the use of a software provided by the *Grundfos* site; for the calculation of the pressure losses and therefore of the head that the pump must provide, was considered the circuit really present in the Navile's technical plant.

This pump is the same for all three solutions.

2.2.2. Water-cooled chiller

The second solution analysed is a water-cooled chiller; in this case, the secondary fluid used to cool the condenser and allow the refrigeration cycle to continue is water.

Specifically, a configuration that includes the chiller together with a cooling tower that takes care of removing heat from the refrigerant through the water, which is then cooled by the external air.

The condensation of the system depends on the ambient bulb temperature, which is lower than the dry bulb temperature, and so the efficiency can increase. However, in the water-cooled chillers that use cooling tower, there is a large consumption of water.

In this thesis the machine will be identified with the code **WW-C02**, figure 2.8.

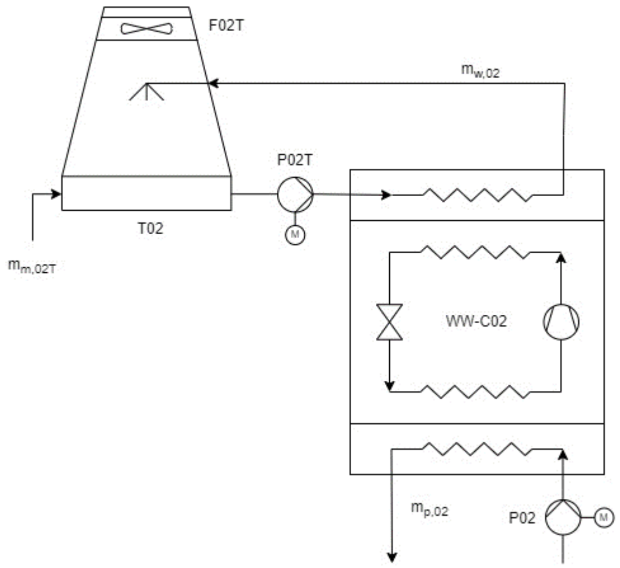


Figure 2.8 : Scheme of water-cooled solution, WW-C02

The typical Carnot cycle of vapor compression machines is illustrated in the schematic representation of the solution.

The evaporator is made in the same way of the air-cooled chiller, so there is a heat exchanger refrigerant-water, in which the water of the district cooling, that flows in the primary circuit, is cool down.

The evaporative capacity is indicated with $\boldsymbol{Q_{evap,W,C02}}$ $[\boldsymbol{MW}]$ and the pump **P02**, controlled by an inverter, of the primary circuit, processes a water flow $\boldsymbol{\dot{m}_{p,02}}$ $[\mathbf{m^3/h}]$.

The difference from the air-cooled chiller lies in the condenser. It consists of a refrigerant-water exchanger, connected to the cooling tower, which precisely cools the water that has extracted heat $\boldsymbol{Q_{cond,WW,C02}}$ $[\boldsymbol{MW}]$to the refrigerant.

The cooling tower circuit specifically consists of the cooling tower **T02** and the tower pump **P02T**, controlled by inverter.

In the cooling tower is highlighted the presence of the fans **F02T** to move the air inside the tower.

Two different water flows are then identified:

- the water flow rate flowing through the exchanger $\dot{m}_{W,02}$ $[\mathrm{m^3/h}]$, elaborated by the tower pump P02T.
- the make-up water flow rate $\dot{m}_{m,02T}$ $[\mathrm{m^3/h}]$, that comes directly from the water supply thanks only to the network pressure.

The last defined flow rate is the one that involves the large consumption of water associated with the use of this solution to produce chilled water.

The reasons for this make-up water are those described in the preceding chapter, so the evaporation of the water during the process, the blowdown, and the drift.

In this situation the energy carrier to consider in the analysis are two: the water and the electricity.

About the electricity, the consumption is related to:

- the refrigeration cycle compressor $P_{WW,C02}$ $[kW]$
- the pump of the primary circuit P_{P02} $[kW]$
- the motor of the fans in the tower P_{F02T} $[kW]$
- the pump of the tower circuit P_{02T} $[kW]$

After defining the scheme of the machine, the size was chosen.

Also in this case, has chosen a size, in order to have a machine that processed **1 MW** of cooling capacity. A chiller of **TRANE** was chosen, in particular a water-cooled **XStream RTWF**, figure 2.9.

Figure 2.9 : Water-cooled chiller of Trane / XStream RTWF

Table 2.14 summarizes the main characteristics of the model chosen, which is the **RTWF 290 HE.**

Table 2.14: Data of chiller on the market - Trane

MACHINE REFERENCE ON THE MARKET	
TRANE – RTWF 290 HE	
Cooling capacity at evaporator [MW][6]	1,005
EER[6]	5,19
ESEER[6]	6,52
Power absorbed by the compressor and fans [kW]	197
Heat rejected at condenser [MW]	1,202
Refrigerant	$R134A$

Therefore, considering the reference machine, the water condenser chiller considered in the analysis has the data shown in table 2.15.

Table 2.15: Data of WW-C02 for the analysis

WATER – COOLED CHILLER /WW – C02	
$Q_{evap,WW,02}$ [MW]	1
$Q_{cond,WW,C02}$ [MW]	1,2
ESEER	5,2
$P_{WW,C02}$ [kW]	197

The seasonal efficiency value that will be considered in the consumption analysis, is that given by the manufacturer of the reference machine reduced by a safety factor of 20%.

About the pump of the primary circuit, the characteristics are in table 2.16.

[6] According to the EN14511:2013 where they are considered: the evaporator at 12/7 ºC, the condenser water temperature at 30/35 ºC.

Table 2.16: Data of P02 for the analysis

PUMP P02	
$\mathbf{P_{P02}}$ **[kW]**	11
Pressure difference [m. c. a.]	10
Water flow rate $\dot{\mathbf{m}}_{\mathbf{p,02}}$ $[\mathbf{m^3/h}]$	172

Considering the condensing capacity, the water flow rate $\dot{\boldsymbol{m}}_{\boldsymbol{W},\mathbf{02}}$ $[\mathbf{m^3/h}]$ can be calculated, considering the specific heat capacity of the water and that the water leaves the condenser at 35°C and, by the cooling tower, it is cooled until 30°C, formula 2.4.

$$\dot{m}_{W,02} = \frac{1200\ kW}{4{,}186\,\frac{kJ}{kgK} * 5K} = 57{,}4\,\frac{kg}{s} = 206{,}7\,\frac{m^3}{h} \qquad [2.4]$$

Then, in order to size the pump of the tower, has been calculated the pressure drop of the circuit, table 2.17.

Table 2.17: Calculation of pressure drop

Calculation of pressure drop		
Distributed	3	m.c.a.
Concentrated	3	m.c.a.
Nozzles	6	m.c.a.
Hydrostatic depression	8	m.c.a.
Condenser	3,5	m.c.a.
Totale	23,5	m.c.a.

Thanks to the Grundfos's software, the pump was sized, and the characteristics are in table 2.18.

Table 2.18: Data of P02T for the analysis

PUMP P02T	
$\mathbf{P_{P02T}}$ **[kW]**	22
Pressure difference [m. c. a.]	24
Water flow rate $\dot{\mathbf{m}}_{\boldsymbol{W},\mathbf{02}}$ $[\mathbf{m^3/h}]$	207

The pumps of the tower, also for the following solution, are in particular, end suction pumps.

Now, the cooling tower must also be chosen.

As mentioned above, the cooling tower must be able to cool the water to the condenser, so it will be chosen in according with the Q_{cond} of the chiller, taking into account an appropriate oversize.

An open circuit cooling tower from **DECSA** has been selected as the reference, considering an oversize of about 60%. The chosen model, **TMA-EU 19-211**, has the characteristics present in table 2.19, and figure 2.10.

Table 2.19: Data of cooling tower on the market - Decsa

MACHINE REFERENCE ON THE MARKET	
DECSA – TMA – EU 19 – 211	
Heat rejection [MW]	1,91
Air flow rate [m^3/s]	38,63
Tower fan consumption [kWe]	22
Oversize respect to the condenser	59%

Figure 2.10 : Cooling tower of Decsa

Therefore, considering the reference machine, the cooling tower considered in the analysis has the data shown in table 2.20.

Table 2.20: Data of the T02 for the analysis

COOLING TOWER – T02	
$Q_{rej,T02}$ [MW]	2
$\dot{m}_{a,T02}$ [m^3/s]	39

$\mathbf{P}_{F02T}$ **[kWe]**	22

2.2.3. Absorption chiller

The last solution studied concerns the production of chilled water thanks to an absorption chiller that uses an aqueous lithium-bromide solution. Also in this case, like the previous one, the condenser is cooled by water, which is then cooled in the cooling tower.

The difference with the two previous solutions, explained in more detail in the previous chapter, is that the chiller is not powered by electricity, but by thermal energy.

In this case is assumed that the thermal energy needed by the absorber derives from the cogeneration system present inside the thermal plant.

The machine will be identified with the code **ABS-C03**, figure 2.11.

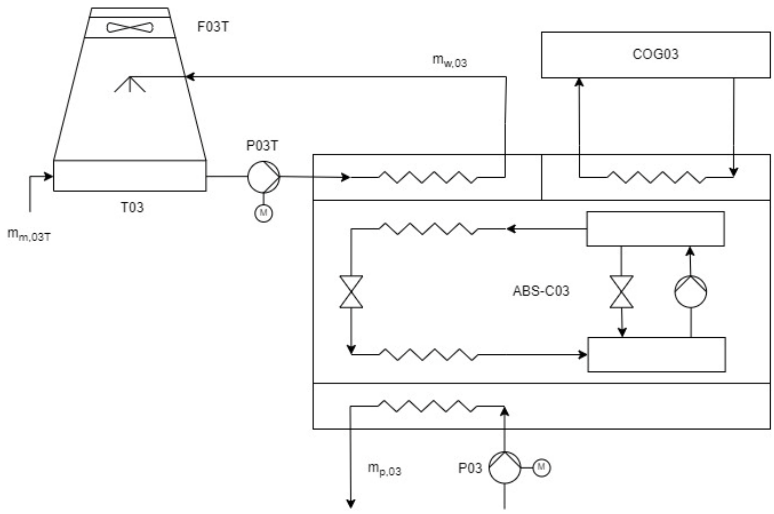


Figure 2.11 : Scheme of absorption solution, ABS-C03

In the schematic representation of the solution is draw the absorption cycle with the generator and absorber in which flows the solution, and usual condenser, evaporator and expansion valve in which flows the water vapour. In this case there is pump instead of a compressor.

So, in the same way of the previous solution, the evaporator is a refrigerant-water heat exchanger, with a pump **P03** for the primary circuit, that processes a water flow $\dot{\boldsymbol{m}}_{\boldsymbol{p},\mathbf{03}}$ $[\mathbf{m}^3/\mathbf{h}]$**;** the evaporative capacity is indicated with $\boldsymbol{Q}_{\boldsymbol{evap,ABS,C03}}$ $[\boldsymbol{MW}]$.

The generator of the absorption cycle is fed by the thermal power produce by the cogeneration unit $\boldsymbol{Q}_{\boldsymbol{COG03}}$ $[\boldsymbol{MW}]$; in the heat exchanger flows a water flow rate $\dot{\boldsymbol{m}}_{\boldsymbol{cog},\mathbf{03}}$ $[\mathbf{m}^3/\mathbf{h}]$, that is send from the cogeneration system to the chiller, thank to a three-way valve.

The condenser with a capacity $\boldsymbol{Q_{cond,ABS,C03}}$ $[\boldsymbol{MW}]$**,** is connected with the cooling tower, in which circuit are identified:

- The water flow rate $\boldsymbol{\dot{m}_{W,03}}$ $[\boldsymbol{m^3/h}]$ elaborated by the pump **P03T, that has an inverter.**
- The make-up water flow rate, $\boldsymbol{\dot{m}_{m,03T}}$ $[\boldsymbol{m^3/h}]$**,** for the losses that occurs during the cooling process.
- The fan **F03T** to move the air inside the tower.

The consumption of electricity in this case is related to:

- the pumps of the solution cycle and absorber: $\boldsymbol{P_{ABS,C03}}$ $[\boldsymbol{kW}]$
- the pump of the primary circuit $\boldsymbol{P_{P03}}$ $[\boldsymbol{kW}]$
- the motor of the fans in the tower $\boldsymbol{P_{F03T}}$ $[\boldsymbol{kW}]$
- the pump of the tower circuit $\boldsymbol{P_{03T}}$ $[\boldsymbol{kW}]$

In order to have a machine of **1 MW** of cooling capacity, was chosen an absorption chiller, with aqueous Li-Br solution, of **LG**, figure 2.12, in particular the model **WCMH 030**.

Figure 2.12 - Absorption chiller of LG / WCMH

The table 2.21 sum up the characteristics of the absorption chiller selected.

Table 2.21: Data of absorption chiller on the market - LG

MACHINE REFERENCE ON THE MARKET	
LG – WCMH 030	
Cooling capacit at evaporator [MW]	1,055
Heat rejected at condenser [MW]	2,143
Hot heat input [MW]	1,302
COP	0,8
Cooling water flow rate $[\mathrm{m^3/h}]$	368,60
Chilled water flow rate $[\mathrm{m^3/h}]$	181,40

Hot water flow rate $[m^3/h]$	48,70
Refrigerant pump [kW]	0,4
Absorbent pump 1 [kW]	2,2
Absorbent pump 2 [kW]	1,5
Purge pump [kW]	0,4

Therefore, considering the reference machine, the absorption chiller considered in the analysis has the data shown in table 2.22.

Table 2.22: Data of the ABS-C03 for the analysis

ABSORPTION CHILLER / ABS – C03	
$Q_{evap,ABS,03}$ [MW]	1
$Q_{cond,ABS,C03}$ [MW]	2
Q_{COG03} [MW]	1,3
COP	0,72
$\dot{m}_{W,03}$ $[m^3/h]$	370
$\dot{m}_{p,03}$ $[m^3/h]$	181
$\dot{m}_{cog,03}$ $[m^3/h]$	50
$P_{ABS,03}$ [kW]	4,5

The consumption of the 4 pumps present in the absorption chiller, are considered together in $\mathbf{P_{ABS,03}}$ **[kW]**, and for the efficiency of the machine, has been considered the COP decreases of a safety factor equal to 10%.

About the pump of the primary circuit, the characteristics are in table 2.23.

Table 2.23: Data of the P03 for the analysis

PUMP P03	
P_{P03} [kW]	11
Pressure difference [m. c. a.]	10

Water flow rate $\dot{m}_{p,03}$ [m^3/h]	172

Considering the condensing capacity, the water flow rate $\dot{m}_{W,03}$ [m^3/h] can be calculated, considering the specific heat capacity of the water and that the water leaves the condenser at 35°C and, by the cooling tower, it is cooled until 30°C, formula 2.5.

$$\dot{m}_{W,03} = \frac{2000\ kW}{4{,}186\frac{kJ}{kgK} * 5K} = 102{,}4\frac{kg}{s} = 368{,}6\frac{m^3}{h} \quad [2.5]$$

The pressure drop has been calculated, in order to size the pump of the tower, table 2.24.

Table 2.24: Calculation of pressure drop

Calculation of pressure drop		
Distributed	3	m.c.a.
Concentrated	3	m.c.a.
Nozzles	6	m.c.a.
Hydrostatic depression	8	m.c.a.
Condenser	8,1	m.c.a.
Totale	28,1	m.c.a.

Thanks to the Grundfos's software, the pump was sized, and the characteristics are in table 2.25.

Table 2.25: Data of the P03T for the analysis

PUMP P03T	
P_{P03T} [kW]	45
Pressure difference [m.c.a.]	28
Water flow rate $\dot{m}_{W,03}$ [m^3/h]	369

Also, for this solution, in order to choose the cooling tower, was considered the condenser capacity, with an oversize about 50-60%.

The condenser capacity for the absorption chiller, is higher than the other two solution, so, considering the cooling tower of **DECSA,** it has been chosen a bigger model **TMA-EU 31-392**, with the characteristics in table 2.26.

Table 2.26: Data of the cooling tower on the market - Decsa

MACHINE REFERENCE ON THE MARKET	
DECSA – TMA – EU 31 – 392	
Heat rejection [MW]	3,38
Air flow rate [m^3/s]	76,20
Tower fan consumption [kWe]	67,5
Oversize respect to the condenser	58%

Therefore, considering the reference machine, the cooling tower considered in the analysis has the data shown in table 2.27.

Table 2.27: Data of the T03 for the analysis

COOLING TOWER – T03	
$\mathbf{Q_{rej,T03}}$ [MW]	3,38
$\dot{\mathbf{m}}_{\mathbf{a,T03}}$ [m^3/s]	70
$\mathbf{P_{F03T}}$ [kWe]	67,5

3 Energy and economic analysis

In this third chapter will be treated first of all the energy analysis of the three different solutions described above, calculating the consumption of each energy carrier involved and making a rough analysis of the environmental impact that they have.

Then an economic analysis is made, following the UNI ISO/TS 50044, at the end of which some indices will be calculated in order to compare the three different solutions.

3.1. Energy analysis

The consumption of electricity, natural gas and water of the three different solutions will be analysed below, starting from the calculation of the working hours of each machine.

Initially a monthly analysis will be made, in particular the period from September 2022 to August 2023, and then it will lead to an annual comprehensive analysis, the starting point for the economic study.

3.1.1. Cooling energy production and equivalent hours

The solutions studied must meet the demand of the district cooling network served by the thermal power plant of the Navile; then, the cooling energy produced by the refrigerator groups E_f $[kWh_f]$ present is taken as a reference, graphic 3.1, as indicated above.

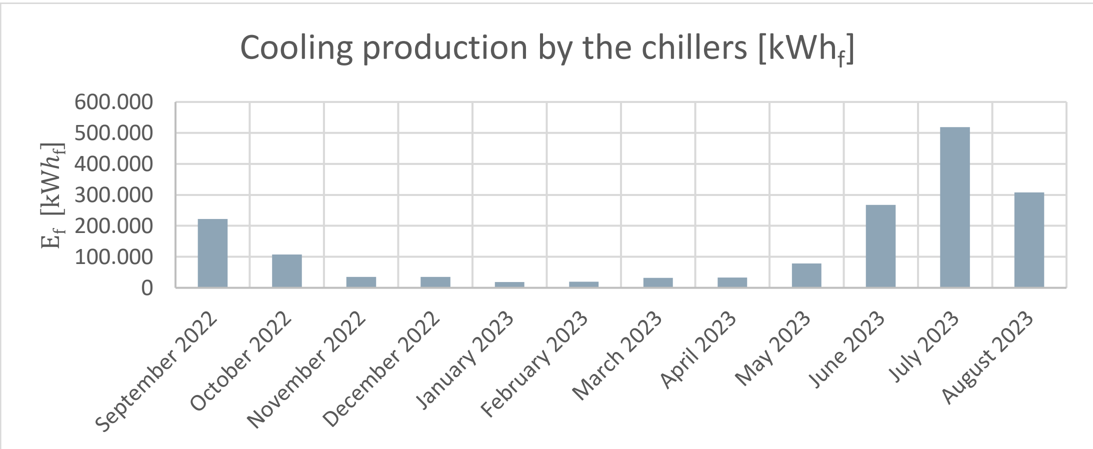


Figure 3.1 : Cooling production by the chillers of the trigeneration plant

Based on this energy, the equivalent hours h_{eq} $[h]$ of each solution are calculated, which will be the same, since the machines have all 1 MW of cooling power Q_{evap}.

The formula that is applied for each single month m is 3.1.

$$h_{eq_m} = \frac{E_{f_m}}{Q_{evap}} \; [h] \qquad [3.1]$$

The operating hours at full load of the three solutions are therefore indicated in table 3.1.

Table 3.1: Operating hours of machine during the year

Period	Equivalent hours [h]
September 2022	222
October 2022	108
November 2022	35
December 2022	35
January 2023	19
February 2023	20
March 2023	33
April 2023	33
May 2023	78
June 2023	268
July 2023	518
August 2023	308
Total	1.675

It can be seen, in graphic 3.2, that machines work much more in summer than in winter, during which cooling energy is required due to the presence in the district cooling network of a building with a server room, which requires cooling even during the heating period.

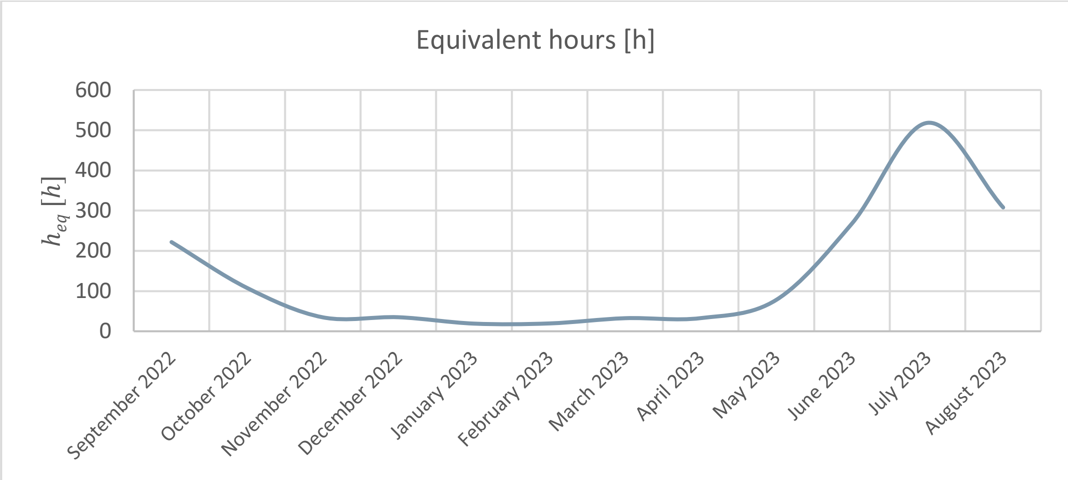


Figure 3.2 : Trend of the equivalent hours during the year

3.1.2. Electricity consumption

The following step concerns the calculation of the electricity consumed by each device in the single solution, as described in the previous chapter.

The calculation was carried out separately for each solution s, but in common they have the fact that for each device d was considered a load factor f_d [%]. In fact, the machines do not work all the time at the maximum load, but some follow a variable load curve, and therefore in function of it, sometimes works partializing. The load factors that have been considered are shown in table 3.2 have been chosen considering the typical operation of the equipment actually present in the thermal plant.

Table 3.2: Load factor of the components

Load factor f_d [%]	
Pumps P01, P02, P03	100%
ABS, C03	70%
Pumps of tower P02T, P03T	75%
Fans of tower F02T, F03T	60%

For almost all devices, for each solution, the formula used for calculating the electrical energy absorbed $E_{a,s}$ $[kWh]$, monthly, is 3.2. The power of each device and solution, is indicated with $P_{d,s}$ $[kW]$.

$$E_{a,s_m} = f_d * h_{eq_m} * P_{d,s} \ [kWh] \qquad [3.2]$$

Instead, the electricity consumed by the air condensing and water condensing refrigerating machine is calculated differently, using the seasonal efficiency SEER and the demand for cooling energy, with the formula 3.3.

$$E_{AW-WW_m} = \frac{E_{f_m}}{SEER} \ [kWh] \qquad [3.3]$$

For all three solutions, the first consumption of electricity is that associated with the pumps of the primary circuit P_{01}, P_{02} e P_{03}; because the pump has been chosen equal for all three solutions, for some assessments made previously, also the electricity absorbed by them E_{P0s} $[kWh]$ will be the same for all three solutions, and is that indicated in table 3.3.

Table 3.3: Energy absorbed by the primary pumps during the year

Period	$\mathbf{E_{P01} - E_{P02} - E_{03}}$ $[kWh]$
September 2022	2.440
October 2022	1.185
November 2022	382
December 2022	384
January 2023	210
February 2023	215
March 2023	358
April 2023	360
May 2023	860
June 2023	2.947
July 2023	5.702
August 2023	3.387
Total	18.430

For the AW-C01 solution, the other electricity consumption is associated only with the compressor of the refrigeration unit and the fans of the condenser that exchanges heat with the external air $E_{AW,C01}$ $[kWh]$, shown in table 3.4.

Table 3.4: Energy absorbed by the compressor of the chiller during the year AW-C02

Period	$E_{AW,C01+F01}$ [kWh]
September 2022	60.008
October 2022	29.138
November 2022	9.398
December 2022	9.451
January 2023	5.162
February 2023	5.282
March 2023	8.798
April 2023	8.860
May 2023	21.157
June 2023	72.481
July 2023	140.255
August 2023	83.317
Total	453.306

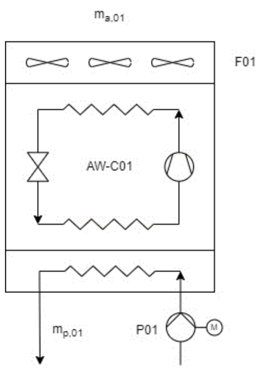


Figure 3.3 : Scheme of air-cooled solution

For the WW-C02 solution, other electricity consumption is associated with the compressor of the refrigeration unit $E_{WW,C02}$ $[kWh]$, the pump E_{P02T} $[kWh]$ and the fans of the evaporative tower E_{F02T} $[kWh]$, shown in table 3.5.

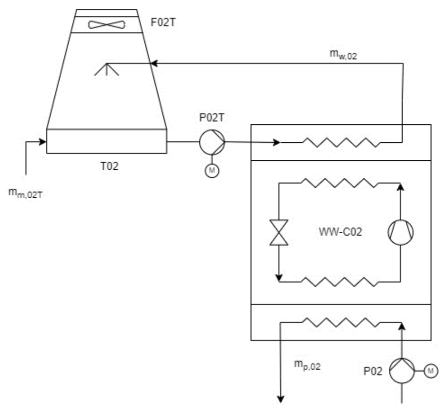


Figure 3.4 : Scheme of water-cooled solution

Table 3.5: Energy absorbed by the compressor of the chiller, and by the cooling tower during the year WW-C01

Period	$E_{WW,C02}$ [kWh]	E_{P02T} [kWh]	E_{F02T} [kWh]
September 2022	42.521	3.660	2.928
October 2022	20.647	1.777	1.422
November 2022	6.660	573	459
December 2022	6.697	576	461
January 2023	3.658	315	252
February 2023	3.743	322	258
March 2023	6.234	537	429
April 2023	6.278	540	432
May 2023	14.992	1.290	1.032
June 2023	51.359	4.420	3.536
July 2023	99.383	8.553	6.843
August 2023	59.037	5.081	4.065
Total	321.208	27.644	22.116

For the ABS-C03 solution, other electricity consumption is associated with the absorption unit pumps $E_{ABS,C03}$ $[kWh]$, the pump E_{P03T} $[kWh]$ and the evaporative tower fans E_{P03T} $[kWh]$, shown in table 3.6.

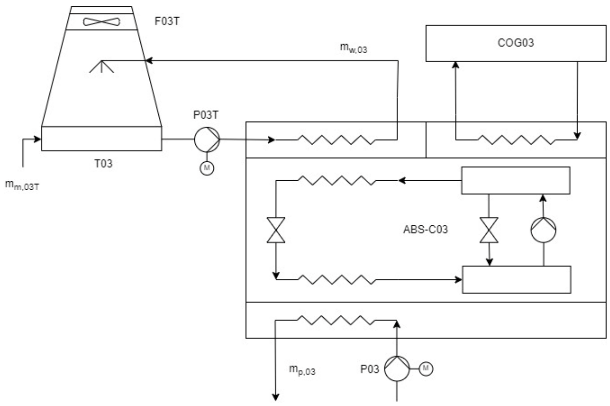


Figure 3.5 - Scheme of absorption solution

Table 3.6: Energy absorbed by the compressor of the chiller, and by the cooling tower during the year ABS-03

Period	$E_{ABS,C03}$ [kWh]	E_{P03T} [kWh]	E_{F03T} [kWh]
September 2022	699	7.485	8.982
October 2022	339	3.635	4.362
November 2022	109	1.172	1.407
December 2022	110	1.179	1.415
January 2023	60	644	773
February 2023	61	659	791
March 2023	102	1.097	1.317
April 2023	103	1.105	1.326
May 2023	246	2.639	3.167
June 2023	844	9.041	10.850
July 2023	1.633	17.495	20.994
August 2023	970	10.393	12.472
Total	5.278	56.545	67.855

For solution 2 (WW-C02) in which there are evaporative tower, SEER performance indices can be calculated also taking into account the presence of the electricity consumption given by the pumps and tower fans.

The new global SEER, that is lower than the previous, is calculated with the formula 3.4 and the values are those given in table 3.7.

$$SEER_{WW-C02} = \frac{E_f}{E_{WW,C02} + E_{P02T} + E_{F02T}} \quad [3.4]$$

Table 3.7 - New SEER of the water-cooled chiller

	$\boldsymbol{SEER}$ $[-]$
$\boldsymbol{WW-C02}$	4,5

3.1.3. Natural gas consumption and electrical energy production

In the ABS-C03 absorber solution, it's necessary to account for an additional energy demand—specifically, the thermal energy sourced from natural gas—and the simultaneous generation of electricity. The absorber relies on thermal energy supplied by a cogeneration unit, which utilizes natural gas for its operation, concurrently producing electricity.

For this analysis, will be considered a cogeneration system COG03 that can reliably fulfil the absorber's monthly thermal requirements. The goal is to calculate the energy consumption and production associated with this specific thermal energy quantity. It's worth noting that the cogeneration unit could potentially be oversized, designed to power additional machinery, that are not included in this case study.

By taking into account the rated thermal power Q_{COG03} $[kW]$ required for the absorber from the cogeneration system and considering the absorber's operational hours, the monthly thermal energy $E_{th_{03,m}}$ $[kWh]$ needed can be calculated using the formula 3.6.

$$E_{th_{03,m}} = Q_{COG03} * h_{eq_m} \ [kWh] \quad [3.6]$$

The results are reported in the table 3.8.

Table 3.8: Thermal energy produced by the cogeneration unit during the year

Period	$E_{th_{03,m}}$ $[kWh]$
September 2022	288.865
October 2022	140.264
November 2022	45.241
December 2022	45.494
January 2023	24.849
February 2023	25.425
March 2023	42.352
April 2023	42.649
May 2023	101.846
June 2023	348.906
July 2023	675.154
August 2023	401.068
Total	2.182.113

In order to calculate the thermal energy entering in the cogeneration unit E_{fuel} $[kWh]$, the amount of natural gas V_{fuel} $[Sm^3]$ and the electricity produced $E_{el,prod}$ $[kWh]$, must be considered the average seasonal electrical $\eta_{el,seas}$ [%] and thermal $\eta_{th,seas}$ [%] efficiency associated with the cogeneration system, table 3.9.

Table 3.9: Efficiency of the cogeneration unit

$\boldsymbol{\eta_{el,seas}}$	36,5%
$\boldsymbol{\eta_{th,seas}}$	47,5%

Then, month by month, was calculated the thermal energy input to the cogeneration unit, with the formula 3.7, and accordingly, considering a PCI of natural gas equal to 9,98 $\frac{kWh}{Sm^3}$ as indicated in the paragraph 2.1.2., also the volume of fuel consumed, with the formula 3.8.

$$E_{fuel} = \frac{E_{th_{03,m}}}{\eta_{th,seas}} \quad [kWh] \qquad [3.7]$$

$$V_{fuel} = \frac{E_{fuel}}{PCI} \quad [Sm^3] \qquad [3.8]$$

The values are shown in table 3.10.

Table 3.10: Energy entering in the cogeneration unit and volume of fuel consumed

Period	E_{fuel} [kWh]	V_{fuel} [Sm^3]
September 2022	608.138	60.936
October 2022	295.292	29.588
November 2022	95.244	9.544
December 2022	95.776	9.597
January 2023	52.314	5.242
February 2023	53.526	5.363
March 2023	89.163	8.934
April 2023	89.788	8.997
May 2023	214.412	21.484
June 2023	734.539	73.601
July 2023	1.421.376	142.422
August 2023	844.354	84.605
Total	4.593.922	460.313

Then considering the average seasonal electrical efficiency, can be derived the amount of electricity produced by the cogeneration system, formula 3.9 and table 3.11.

$$E_{el,prod} = E_{fuel} * \eta_{el,seas} \quad [kWh] \qquad [3.9]$$

Table 3.11: Electricity produced by the cogeneration unit

Period	$E_{el,prod}$ [kWh]
Setember 2022	221.970

October 2022	107.782
November 2022	34.764
December 2022	34.958
January 2023	19.095
February 2023	19.537
March 2023	32.544
April 2023	32.773
May 2023	78.261
June 2023	268.107
July 2023	518.802
August 2023	308.189
Total	1.676.781

Not all the energy produced by the cogenerator is sold on the grid, because part of it is self-consumed on site. Month by month, the electricity produced by the cogeneration unit is greater than the electricity requirement of the solution in general; it is assumed that 60% of the need is covered by the electricity produced by the cogeneration unit. 40% of electricity is then taken from the grid and the surplus of cogenerated energy is sold on the grid. In table 3.12 Are reported the value of the self-consumed energy, the electrical energy sold on the grid and, consequently, the electrical energy taken from the grid.

Table 3.12 : Value of electricity self-consumed, sold to the grid and taken from the grid

Period	$E_{self-consumed}$ [kWh_e]	E_{sold} [kWh_e]	$E_{from\ grid}$ [kWh_e]
September 2022	13.724	208.246	5.882
October 2022	6.664	101.117	2.856
November 2022	2.149	32.615	921
December 2022	2.161	32.797	926
January 2023	1.181	17.914	506
February 2023	1.208	18.329	518

March 2023	2.012	30.532	862
April 2023	2.026	30.746	868
May 2023	4.839	73.422	2.074
June 2023	16.577	251.530	7.104
July 2023	32.077	486.725	13.747
August 2023	19.055	289.134	8.167
Total	103.675	1.573.106	44.432

3.1.4. Water consumption

Another very important aspect to evaluate is the water consumption inside the plant.

The solution AW-C01, being air cooled, does not have a water consumption.

Instead, in the solutions WW-C02 and ABS-C03 water is used as reintegration in evaporative towers, to compensate for the reduction of water due to evaporation, blowdown and drift.

The calculation of the make-up water flow rate, for each of the two solutions, was made following the procedure described in paragraph 1.3.3., going to calculate separately the 3 components:

- make-up water flow rate for evaporation: $\dot{m}_{m,evap}$ $[m^3/h]$
- repeat water flow rate for blowdown: $\dot{m}_{m,bd}$ $[m^3/h]$
- repeat water flow rate for drift: $\dot{m}_{m,drift}$ $[m^3/h]$

The make-up water flow rate due to evaporation, it was calculated with formula 3.10.

$$\dot{m}_{m,evap} = \frac{Q_{cond}}{\Delta h_{vap}} \left[\frac{m^3}{h}\right] \tag{3.10}$$

Where the Δh_{vap} $[kJ/m^3]$ is equal to $2260000\ kJ/m^3$, considering the water at the atmospheric pressure and at the boiling point and the Q_{cond} is the heat exchanges in the condenser.

For the water due to the blowdown, the graph 1.25 was used, whereas it has a reintegration water with a hardness of 15 °F and a maximum admissible hardness for the cooling water of 20 °F. It should be noted that the mains water that reaches the towers, has undergone before the treatments, as the net water in Bologna, has a hardness in the range of $26 \div 30\ °F$.

Then, on the basis of the graph and the input data, an average quantity of water flow rate $\dot{m}_a$ is obtained equal to the formula 3.11.

$$\dot{m}_a = 4 * D_v = 4 * \dot{m}_{m,evap} \left[\frac{m^3}{h}\right] \tag{3.11}$$

And then, formula 3.12, can be found the make-up water flow rate for the blowdown, equal to the difference between water $\dot{m}_a$ and evaporated water $\dot{m}_{m,evap}$.

$$\dot{m}_{m,bd} = \dot{m}_a - \dot{m}_{m,evap} \left[\frac{m^3}{h}\right] \tag{3.12}$$

About the make-up water flow rate due to the drift, has been considered that is equal to 0,005% of the circulation water flow rate, formula 3.13.

$$\dot{m}_{m,drift} = 0{,}005\% * \dot{m}_W \left[\frac{m^3}{h}\right] \tag{3.13}$$

It can then be verified that the % indicated in paragraph 1.3.3 are respected, in comparison to the value of circulating water $\dot{m}_W$. The percentages are given in table 3.12 and are the same for the two solutions.

Table 3.13: Percentage of make-up water respect the circulating water in the cooling tower

$\dot{m}_{m,evap}$	0,926 %
$\dot{m}_{m,bd}$	2,778 %
$\dot{m}_{m,drift}$	0,005 %

So, the total make-up water $\dot{m}_m$ $[m^3/h]$ is the sum of the three component, formula 3.14 and for the two solution the value are reported in the table 3.13.

$$\dot{m}_m = \dot{m}_{m,evap} + \dot{m}_{m,bd} + \dot{m}_{m,drift} \left[\frac{m^3}{h}\right] \tag{3.14}$$

Table 3.14: Make-up water flow rate of T02 and T03

$\dot{m}_{m,02T}$	$7{,}67\ m^3/h$
$\dot{m}_{m,03T}$	$13{,}67\ m^3/h$

Then can be calculated, for the two solutions, month by month, the volume of water V_m $[m^3]$ consumed for the reintegration of the cooling tower, con la formula 3.15.

$$V_{m,m} = h_{eq_m} * \dot{m}_m \; [m^3] \qquad [3.15]$$

The values are reported in the table 3.14.

Table 3.15: Volume of water consumed in the T02 and T03

Period	$V_{m,02T}\,[m^3]$	$V_{m,03T}\,[m^3]$
September 2022	1.701	3.033
October 2022	826	1.472
November 2022	266	475
December 2022	268	478
January 2023	146	261
February 2023	150	267
March 2023	249	445
April 2023	251	448
May 2023	600	1.069
June 2023	2.054	3.663
July 2023	3.976	7.088
August 2023	2.362	4.210
Total	12.849	22.908

3.1.5. Environmental analysis and CO2 emissions

This paragraph outlines a basic environmental analysis focusing on CO_2 emissions associated with electricity from the grid and the usage of natural gas.

The CO_2 equivalent emission factors utilized are presented in table 3.15.

Table 3.16: Emission factors considered for natural gas and electricity

$fe_{CO_2,gn}$[7]	$2{,}006 * 10^{-3} \; \frac{tCO_2eq}{Sm^3}$

[7] Source natural gas emission factor of 2021: ISPRA – "Italian Greenhouse Gas Inventory 1990-2021 National Inventory Report 2023"

$fe_{CO_2,el}$[8]	$0{,}4571 * 10^{-3}\ \frac{tCO_2eq}{kWhe}$

Subsequently, the emissions in tonnes of CO_2equivalent were computed for each solution on a monthly basis, as detailed in table 3.16.

Table 3.17: Tons of emission for the three solutions

	AW – C01	**WW – C02**	**ABS – C03**	
Period	$\mathbf{CO_{2\ eq,el}}$ **[ton]**	$\mathbf{CO_{2\ eq,el}}$ **[ton]**	$\mathbf{CO_{2\ eq,el}}$ **[ton]**	$\mathbf{CO_{2\ eq,gn}}$ **[ton]**
September 2022	28,545	23,563	2,689	122,24
October 2022	13,860	11,441	1,305	59,35
November 2022	4,471	3,690	0,421	19,14
December 2022	4,496	3,711	0,423	19,25
January 2023	2,456	2,027	0,231	10,52
February 2023	2,512	2,074	0,237	10,76
March 2023	4,185	3,455	0,394	17,92
April 2023	4,214	3,479	0,397	18,05
May 2023	10,064	8,308	0,948	43,10
June 2023	34,478	28,460	3,247	147,64
July 2023	66,717	55,072	6,284	285,70
August 2023	39,632	32,715	3,733	169,72
Total	215,630	177,994	20,310	923,4

[8] Source electricity from the grid emission factor of 2022: Association of Issuing Bodies (AIB) - “European Residual Mixes: Results of the calculation of Residual Mixes for the calendar year 2022”

3.1.6. Comparison between the solutions

A comparison in energy terms of the three different solutions is made below. The annual consumption values, as detailed in the preceding paragraphs, are presented in table 3.17.

Table 3.18: Summary of the main data of the three solutions

Period: Sep – 2022 / Aug – 2023	**AW – C01**	**WW – C02**	**ABS – C03**
Cooling production by the chillers [kWh_f/y]	1.675.420	1.675.420	1.675.420
Equivalent hours [h/y]	1.675	1.675	1.675
Consumption electrical energy [kWh_{el}/y]	471.736	389.397	44.432
Water consumption [m^3/y]	–	12.849	22.908
Natural gas consumption [Sm^3/y]	–	–	460.313
Electrical energy production [kWh_{el}/y]	–	–	1.676.781
$CO_2eq_{electrical\ energy}$ [ton]	216	178	20
$CO_2eq_{natural\ gas}$ [ton]	–	–	923

Graph 3.6 shows the consumption of electricity.

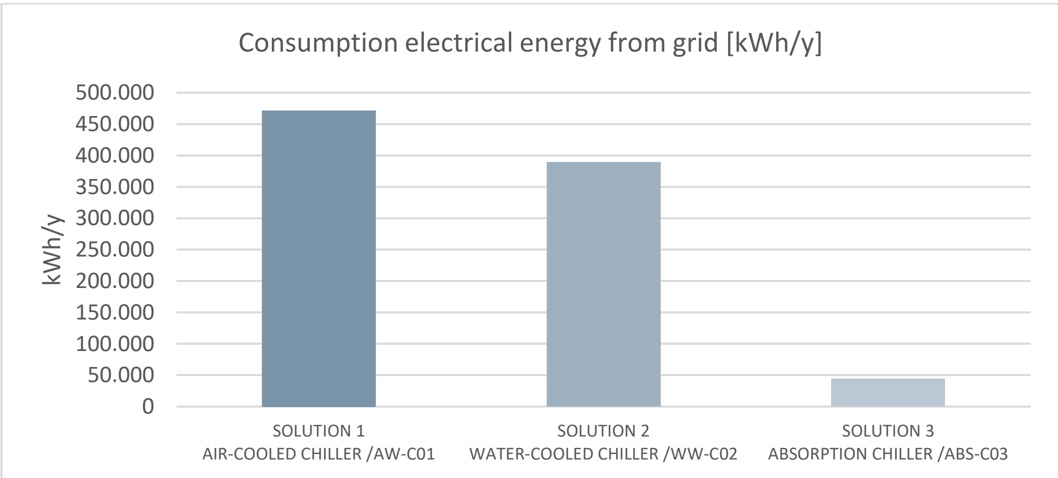


Figure 3.6 - Consumption of electrical energy

It can be noted that the lower consumption is associated with the third solution, the absorption chiller (ABS-C03), because the use of electricity is associated only with pumps, which consume less than the compressors, instead present in the solution with air condenser (AW-C01) and water (WW-C02). On the other hand, for the ABS-C03 solution, there is a consumption of natural gas, associated with the production of thermal energy that the cogenerator provides to the absorber to power it.

As for water consumption, it is present only in the solution WW-C02 and ABS-C03, which each have a cooling tower to remove the heat from the condenser, figure 3.7. In particular, the consumption is higher in the cooling tower of the absorption solution because it has a larger size than the other.

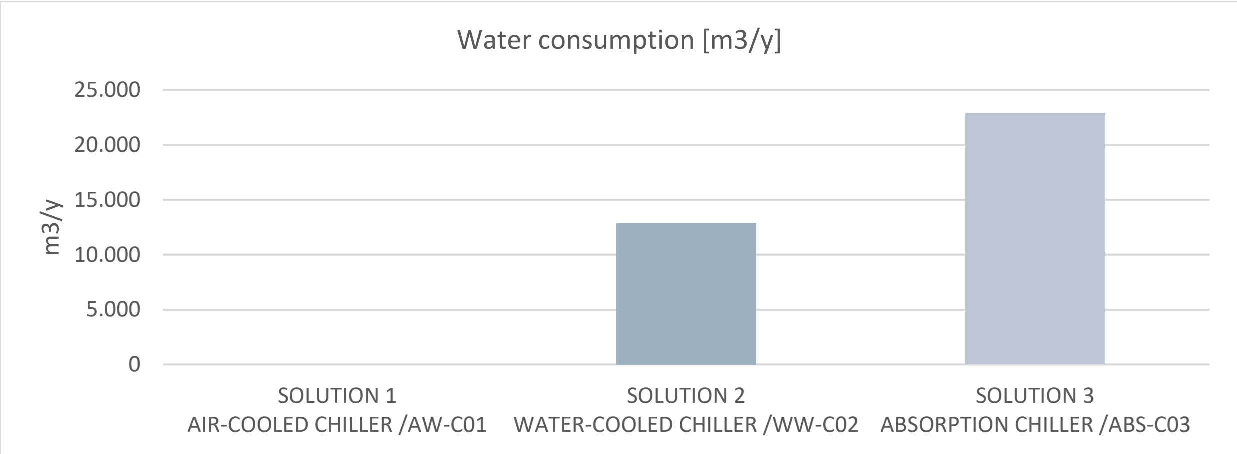


Figure 3.7 : Water consumption of the cooling tower

3.2. Economic analysis

In this second section of the third chapter, an economic-financial analysis will be carried out in order to be able to compare, from an economic point of view, as well the three different solutions studied. This will be done through the guidance of the UNI ISO/TS 50044 technical specification, which will lead, after studying the costs and revenues of each solution, to the determination of some economic indicators.

3.2.1. UNI ISO/TS 50044 – Guidelines for economic and financial evaluation

The economic analysis of these three different solutions will be based on the procedure and indices described in **UNI ISO/TS 50044**, "*Energy saving projects (EnSPs)-Guidelines for economic and financial evaluation*", the national adoption of the 2019 international technical specification ISO/TS 50044, developed by Technical Committee ISO/TC 301, "*Energy management and energy savings*."

By means of economic and financial evaluation, the following technical specification provides guidelines to be able to analyse and compare energy saving projects (EnSp), using economic and financial evaluations.

The main steps for an economic and financial evaluation of an EnSp, given in the standard and adapted for this thesis, are as follows:

1. Plan the economic and financial evaluation, describing the energy intervention, its useful life, identifying and defining the boundaries, and collecting the data needed for the evaluation.
2. Identify the revenues.
3. Identify the costs.
4. Calculate cash flows.
5. Definition and calculation of key economic and financial indices, net present value (NPV), payback period (PP) and internal rate of return (IRR).
6. Comparison of different solutions.

3.2.2. Planning of economic and financial evaluation

As previously described, the energy intervention consists of the installation of a chiller to produce chilled water for the district cooling network serving the "Ex-Mercato Navile" in Bologna.

The considered boundaries of the economic analysis, in order to evaluate all the factors involved in the energy intervention, are the same as those evaluated to do the energy analysis. Details of costs and revenues will be specified later.

Regarding the useful life of the project usually is about 15 to 20 years; for all three solutions, a useful lifetime of 20 years was considered in the economic-financial analysis.

Finally, as for the energy data that will be used to estimate costs and revenues are those shown in table 3.18, over a period of 1 year.

Table 3.19: summary of the main data of the three solutions used int he economic analysis

Period: **Sep – 2022 / Aug – 2023**	**AW – C01**	**WW – C02**	**ABS – C03**
Cooling production by the chillers $[kWh_f/y]$	1.675.420	1.675.420	1.675.420
Equivalent hours $[h/y]$	1.675	1.675	1.675
Consumption electrical energy $[kWh_{el}/y]$	471.736	389.397	148.107
Water consumption $[m^3/y]$	–	12.849	22.908
Natural gas consumption $[Sm^3/y]$	–	–	460.313
Electrical energy production $[kWh_{el}/y]$	–	–	1.676.781

3.2.3. Evaluation of revenues

Revenues from the energy intervention can be energy and non-energy. In the present case, the non-energy revenues are the White Certificates CB, associated only with the cogeneration system.

3.2.3.1. Sale of cooling energy

The first revenue taken into account is the sale of the cooling energy produced by each solution's chiller, which is then sent into the district cooling network.

In order to estimate the amount derived from it, the amount of cooling energy directly produced by the chillers is not evaluated, but the energy demanded by the utilities, in which the energy losses occurring along the network have therefore already been taken into account. The annual value of energy demanded and sold to the utilities is shown in table 3.19.

Table 3.20: Needs of the district

Period: **Sep – 2022 / Aug – 2023**	**All three solutions**
Cooling needs of the district $[\mathbf{kWh_f/y}]$	1.405.340

Based on the latest histories of the Navile technology plant, the cost of selling refrigeration energy was considered to be 0,157 $€/kWh_f$.

Thus for a single year, the revenue from the sale of refrigeration energy, which is the same for all three technologies, is equal to that shown in table 3.20.

Table 3.21: Revenue by the sale of cooling energy during a year

	All three solutions
Cooling energy sold [€/y]	221.060

3.2.3.2. Sale of electric energy produced

The second revenue to be considered is the sale to the grid of the electricity produced by the cogenerator, calculated on the basis of how much thermal energy is supplied to power the absorber. It is therefore an income to be associated exclusively with solution 3 (ABS-C03).

In the event that the cogeneration unit produces a greater amount of thermal energy, going to feed other utilities, other than the absorber, the corresponding electricity produced, is not valued among the revenues of this energy intervention.

Not all the energy produced by the cogeneration unit is sold on the grid, because part of it (60% of it, like explained in the previous chapter) is self-consumed on site. Table 3.21 shows the values of the different energies during a year.

Table 3.22 : Distribution of energy self-consumed, sold and taken from the grid.

	ABS – C03
Electricity self – consumed $[\mathbf{kWh_e}]$	103.675
Electricity sold to the grid $[\mathbf{kWh_e}]$	1.573.107
Electricity taken from the grid $[\mathbf{kWh_e}]$	44.432

Based on the latest histories of the Navile Power Plant, the value that was used to value the electricity sold to the grid is 0,161 $€/kWh_e$, about 65% of the cost of energy taken from the network.

Thus, for a single year, the revenue from the sale of electricity to the grid, produced by the cogeneration unit and not self-consumed, is equal to that shown in table 3.22.

Table 3.23: Revenue by the sale of electricity during a year

	ABS – C03
Electricity sold to the grid [€/y]	252.624

3.2.3.3. White Certificates

An important income that must be considered for solution 3 (ABS-C03) is that resulting from the annual release of white certificates (CB) associated with the cogeneration unit, imposed by *Article 4 of the DM of 5 September 2011* for all units that meet the CAR requirements, high efficiency cogeneration.

It is therefore necessary first of all to verify that the part of cogeneration associated with the solution in question is considered high-efficiency cogeneration.

To recognition of operation in high-efficiency cogeneration the unit must have a Primary Energy Saving (PES) greater than or equal to 10%. The PES allows to quantify at equal output of electricity and heat, the savings obtained in terms of inputs from the combined production of outputs compared to their potential separate production.

Annex II of the Ministerial Decree of 4 August 2011 defines the procedure of the quantities involved for the purposes of calculating the PES.

The first thing to do is to calculate the overall efficiency, η_{global} [%] of the unit: if it is greater than or equal to 75%, then the PES evaluation boundary will be the only cogeneration unit.

Overall efficiency is defined as in formula 3.16, calculated during a year.

$$\eta_{global} = \frac{E_{CHP} + H_{CHP}}{F_{CHP}} \quad [\%] \tag{3.16}$$

where:

- E_{CHP} $[kWh]$ is the electricity produced by the unit.
- H_{CHP} $[kWh]$ is the unit's cogenerated useful thermal energy.
- F_{CHP} $[kWh]$ is the thermal energy input into the unit.

In the present case, the overall return is equal to 84 %.

The PES, defined by *Annex III of the Ministerial Decree of 4 August 2011* with the formula 3.17, can now be calculated.

$$PES = 1 - \frac{F_{CHP}}{\frac{E_{CHP}}{\eta_{es}} + \frac{H_{CHP}}{\eta_{ts}}} \geq 10\% \qquad [3.17]$$

Where:

- η_{es} is the efficiency reference value for separate heat production, taken equal to 52,1 % [9]
- η_{ts} is the efficiency reference value for the separate production of electricity, taken equal to the 90% [9]

The primary energy saving of the cogeneration unit of the third solution is equal to 18,56%, so the unit is in CAR regime.

Finally, the number of white certificates of which the cogeneration unit is entitled is calculated, based on the primary energy savings achieved in the year under consideration, calculated by formula 3.18.

$$RISP = \frac{\mathrm{E_{chp}}}{\eta_{E,RIF}} + \frac{H_{chp}}{\eta_{T,RIF}} - F_{chp}\ [MWh] \qquad [3.18]$$

where:

- $\eta_{E,RIF}$ is the average conventional yield of the Italian electricity production fleet, which is 46%
- $\eta_{T,RIF}$ is the average conventional yield of the Italian thermal production park, assumed to be 90%, since the thermal energy produced by the cogeneration unit is directed to the production of hot water.

The number of white certificates is defined by the formula 3.19, where K represents a coefficient of harmonization that varies according to the power of the unit of cogeneration and that in this case will be assumed equal to 1.

$$CB = RISP * 0{,}086 * K \qquad [3.19]$$

The value of a single white certificate has been set at 250 € and in addition they are levied for the first 10 years of operation of the cogeneration unit.

The white certificates CB for a year and the respective revenue associated with the ABS-C03 solution are shown in table 3.23.

[9] Obtained from Commission Delegated Regulation (EU) 2015/2402 of 12 October 2015

Table 3.24 : Number of White Certificates

	ABS – C03
N° of CB [nr/y]	127
Revenue from CB [€/y]	31.730

3.2.4. Evaluation of cost

Another important objective of economic analysis is the calculation of costs.

There is first to consider the initial investment associated with each technology.

Then variable costs are to be considered, which are at a minimum made on an annual basis, include labour, materials, utilities, such as water and energy used, repairs, and maintenance; in general, they can be separated into operating costs and maintenance costs.

3.2.4.1. Investment costs

The investment costs associated with each technology were estimated based on a market analysis of machines of similar size and are as shown in table 3.24.

Table 3.25: Initial investment costs of the three solutions

	AW – C01	WW – C02	ABS – C03
Initial investment [€]	230.000	290.000	420.000

These costs will be considered only at the beginning of the project.

3.2.4.2. Operation and maintenance costs

Regarding operating and maintenance O&M costs, there are first of all the costs associated with machine maintenance, which in the case of solution 2 and 3, the costs also include the maintenance of the cooling towers. The values are shown in table 3.25 and are costs on an annual basis.

Table 3.26: Maintenance costs of the three solutions during a year

	AW – C01	WW – C02	ABS – C03
Maintenance of machine [€/y]	8.000	15.000	20.000

For the solutions that report water consumption, WW-C02 and ABS-C03, there is a cost associated with the treatment to be done to the make-up water that comes from the water supply, in order to reduce its hardness from about 25°F to 15 °F. The assumed cost is evaluated on m3 of make-up water and is 1 $€/m^3$.

Finally, for the last solution, ABS-C03, which includes the use of the cogenerator, there is a maintenance cost associated with the hours the cogeneration unit is working to produce the thermal energy required by the absorption unit. This cost is provided on an hourly basis and is 10 $€/h$.

Below in table 3.26 are the operational costs for one year, for each solution.

Table 3.27: Make-up water treatment and cogeneration maintenance during a year

	AW – C01	**WW – C02**	**ABS – C03**
Make – up water treatment [$€/y$]	–	12.849	22.908
Cogenerator maintenance [$€/y$]	–	–	16.754

3.2.4.3. Utilities costs

Another important cost to be evaluated is the cost of utilities, in this case electricity taken from the grid, natural gas used to power the cogeneration unit, and for make-up water used in the cooling towers.

Energy and water costs were assumed on the basis of a market survey, and the values are as shown in table 3.27 and for each solution, in table 3.28.

Table 3.28: Unit cost of the single utilities

Electricity from grid [$€/kWhe$]	0,247
Natural gas [$€/Sm^3$]	0,643
Water [$€/m^3$]	4,7

Table 3.29: Cost due to the utilities for the three solutions during a year

	AW – C01	**WW – C02**	**ABS – C03**
Electricity from grid [$€/y$]	116.547	96.205	10.977
Natural gas [$€/y$]	–	–	296.027
Water [$€/y$]	–	60.390	107.667

It should be specified, that since this analysis is done for a business and not for a residential utility for example, the energy costs are exclusive of VAT.

Finally, a convenience index for cogeneration called K_{cog} has been calculated, defined as the ratio between the cost of electricity $[€/kWh_e]$ and the cost of natural gas $[€/Sm^3]$.

In the case of the ABS-C03 solution, the index is equal to 0,38 Sm^3/kWh_e. In general, more this index is higher, more the cogeneration is convenient, because electricity cost more and therefore is also sold at a higher price.

3.2.5. Evaluation of cash flow

After determining the costs and revenues of the energy intervention, cash flow is evaluated in order then to be able to assess subsequent economic ratios.

Cash flow is calculated as the difference between revenues and costs for each year. In order to account for possible changes in energy and nonenergy costs, it is very important to go to an **inflation rate** $\boldsymbol{i}$ [%] applied to each cost and revenue. In particular, an energy and nonenergy inflation rate of 2 % was assumed.

Then considering the costs and revenues calculated from year to year, they were multiplied by a factor to account for the inflation rate, and then the costs and revenues are calculated using the following formula equal to formulas 3.20.

$$\begin{gathered} C_{y,i} = C_y * (1+i)^y \quad \left[{}^{€}/_{y}\right] \\ R_{y,i} = R_y * (1+i)^y \quad \left[{}^{€}/_{y}\right] \end{gathered} \qquad [3.20]$$

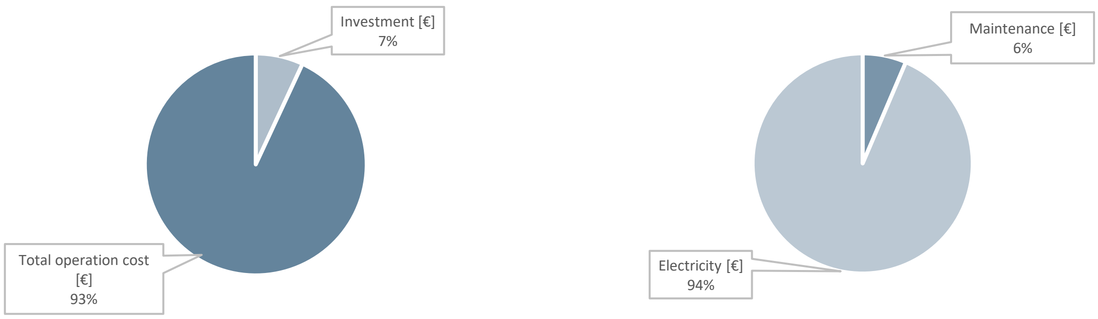
Where C_y and R_y are the costs and revenues, year by year, without considering the inflation rate and $C_{y,i}$ and $R_{y,i}$ are the costs and revenues, year by year, considering the inflation rate. The exponent y indicates the year being considered.

After calculating the costs and revenues, the annual cash flow is then evaluated using formula 3.21.

$$A_y = R_{y,i} - C_{y,i} \left[{}^{€}/_{y}\right] \qquad [3.21]$$

In the Appendix A are reported the details of the cash flow for the entire 20 years.

Below are the graphs 3.8, 3.9 and 3.10 that represent for each solution the percentage breakdown of costs during the 20 years, considering inflation. In a first case (a), it comes evidenced which is the percentage of the initial investment regarding the total of the operation costs, and it can be noticed that it is approximately the same for all the three solutions; in a second case (b) it comes instead shown the subdivision of the total costs in the single lines of cost: It can be noted that in the third solution the costs deriving from natural gas play a very important role, while for the second solution electricity has the greatest role.

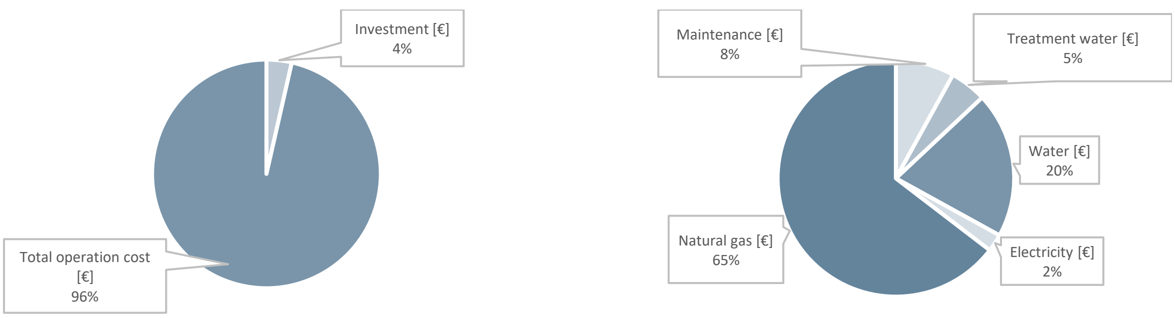


(a) (b)

Figure 3.8 : Solution air-cooled - AW-C01

(a) (b)

Figure 3.9 : Solution water-cooled - WW-C02



(a) (b)

Figure 3.10 : Solution absorption - ABS-C03

3.2.6. Economic indicators

In order to assess whether the energy intervention is worthwhile or not, thus whether the benefits brought by it exceed the costs, it is important to go to the time value of money.

To do this, the following economic indicators will be analysed in this thesis:

- Net present value NPV
- Internal rate of return IRR
- Discounted Payback period DPP

The following indicators will be used to compare different solutions.

3.2.6.1. Net Present Value – NPV

Before going to add up all the cash flows for each year, it is important to go to discount the cash flow, thus considering a **discount rate** $\boldsymbol{r}$ [%].

Discounting is important for several reasons, such as considering the fact that money received today is more valuable than money received in the future, because it can, for example, be invested. The discount rate also takes cost of the uncertainty that may exist about future cash flows, considering the risk that these expected cash flows may not be generated.

This concept is encapsulated in the calculation of $\boldsymbol{Net\ Present\ Value}$ $(\boldsymbol{NPV})$ [€], which is first evaluated for each individual year, and then throughout the 20-year time period ("n" is the useful time of the project, in years), formula 3.22 and 3.23.

$$NPV_y = \frac{R_{y,i} - C_{y,i}}{(1+r)^y} \quad [€] \qquad [3.22]$$

$$NPV = \sum_{y=0}^{n} \frac{R_{y,i} - C_{y,i}}{(1+r)^y} \quad [€] \qquad [3.23]$$

For all three solutions considered in this thesis, a discount rate r of 5% was assumed.

The calculation of cash flows, NPVs for each year, and cumulative cash flow over the 20 years taken into consideration for each solution are reported in Appendix A; in table 3.29 are shown the Net present values of the three different solutions.

Table 3.30: Net present value of the three solutions

	AW – C01	**WW – C02**	**ABS – C03**
NPV [€]	1.213.709	374.015	22.598

3.2.6.2. Internal rate of return – IRR

In order to assess the profitability of an investment, it is very important to go and calculate the $\boldsymbol{Internal\ rate\ of\ return}$ $(\boldsymbol{IRR})$ [%]. In fact, an energy intervention is attractive if $IRR > MARR$ where MARR is the minimum accetable rate of return, which value is established for the precise intervention being studied.

At the mathematical level, the IRR is the discount rate that makes the NPV equal to zero.

To calculate this value, an Excel function was used that sees as input all the discounted cash flows.

Shown below in table 3.30 are the Internal rate of return values of the three different solutions.

Table 3.31: Internal rate of return of the three solutions

	AW – C01	**WW – C02**	**ABS – C03**
IRR [%]	45%	15,6%	6%

3.2.6.3. Discounted payback period – DPP

The $\boldsymbol{Discounted\ payback\ period}$ $(\boldsymbol{DPP})$ $[\boldsymbol{years}]$ is an index used to calculate the payback time of the investment.

Discounted cash flows are considered for the calculation of this index, and in particular, if they are different during the lifetime of the project, a formula that could be used to calculate the DPP is 3.24.

$$DPP = (N-1) + \frac{C_i - Cum\ NPV_{N-1}}{NPV_N} \quad [years] \qquad [3.24]$$

Where N is the year in which the cumulative discounted cash flows exceed the initial investment C_i [€] and instead $CumNPV_{N-1}$ is the cumulative discounted cash flow at the end of the year $(N-1)$.

More simply for this analysis, the DPP is evaluated using formula 3.25, where "a" represents the last period in which the cumulative discounted cash flow was negative, "b" represents, in absolute value, the cumulative discounted cash flow at the end of period "a," and "c" is the discounted cash flow during period "a+1."

$$DPP = a + \frac{b}{c} \quad [years] \qquad [3.25]$$

Shown below in table 3.31 are the discounted payback period values of the three different solutions, also represented graphically in graphs 3.11, 3.12 and 3.13.

Table 3.32: Discounted payback period of the three solutions

	$\mathbf{AW - C01}$	$\mathbf{WW - C02}$	$\mathbf{ABS - C03}$
DPP $[year]$	3	8	19

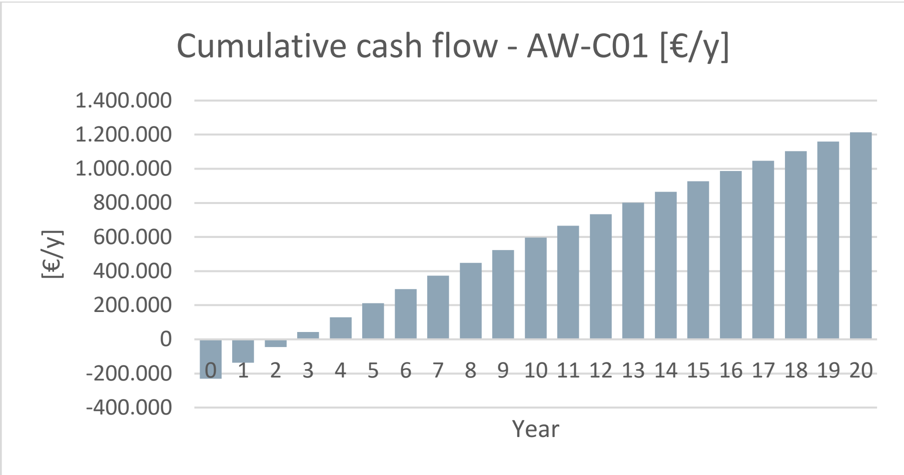


Figure 3.11 : Graphical representation of the DDP of first solution

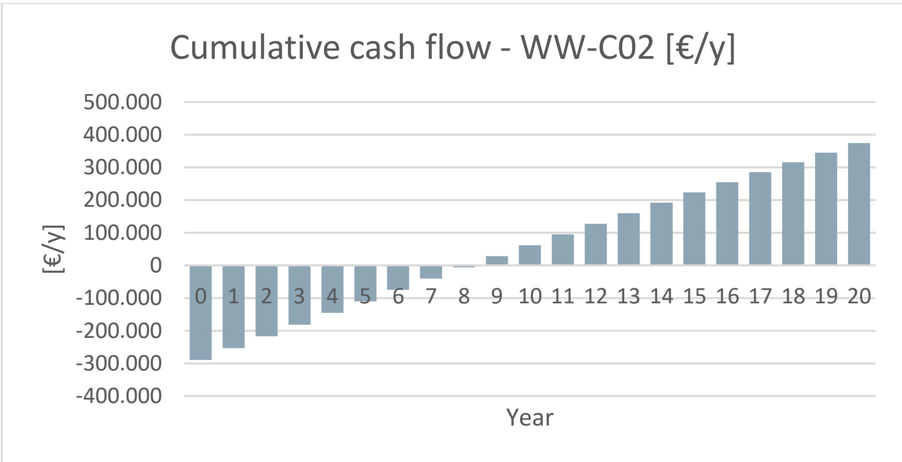


Figure 3.12 : Graphical representation of the DDP of second solution

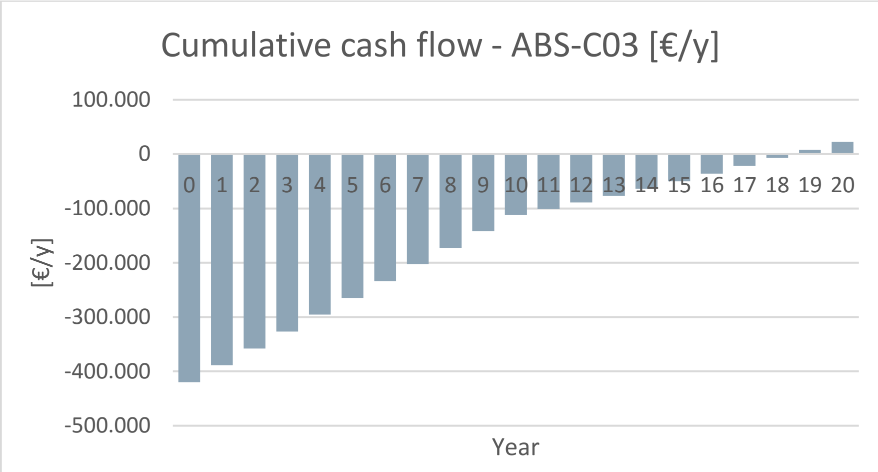


Figure 3.13 : Graphical representation of the DDP of third solution

3.2.7. Comparison of the solutions

A comparison in economic terms of the three different solutions is made below, summarising the main economic items and indices for each solution, table 3.32.

Table 3.33 : Summary of the main economic data of the three solutions

Period: 20 years	**AW – C01**	**WW – C02**	**ABS – C03**
Initial investment [€]	230.000	290.000	420.000
Total operation cost[10] [€]	3.086.690	4.300.191	11.357.274
Total revenues[10] [€]	5.478.600	5.478.600	12.056.753
Specific cost for cooling production [€/MWh_f][11]	74	110	113 (132)[12]
NPV [€]	1.213.709	374.015	22.598
DPP [year]	3	8	19
IRR [%]	45%	15,6%	6%

A more detailed comparison will be made in the conclusions chapter.

[10] For this calculation has been considered the inflation, but not the actualization.
[11] This specific cost has been calculated excluding the initial investment. For the solution with the cogeneration unit, have been subtracted from the costs the profits deriving from the sale in net of the electricity and the white certificates.
[12] The value 132 €/MWh_f is considering that the white certificates are not more considered.

3.3. Sensitivity analysis

In this section, a sensitivity analysis will be carried out for each of the three solutions. The sensitivity analysis is a very important tool to evaluate how the results of a model vary in response to changes in the most important variables, in order to understands the sensitivity of certain output parameters, changing input parameters. This also allows to understand where it is most important to bring attention to optimize the solution overall.

Values will generally vary from -60% to +60%, with an interval of 20%. In some specific cases other percentages will be used instead.

Moreover, it is considered as a parameter to see the changes, the return on investment time; this value, for a mathematical question of how the analysis was carried out, cannot go beyond the 20 years, which represent the useful life of the investment, Thus, all DPPs equal to 20, represent even greater return times.

3.3.1. Cooling energy

For all the three solutions an analysis is made, table 3.33, table 3.34 and table 3.35, going to estimate as the time of return of the investment varies, varying the **cost of sale of the cooling energy**. The rate of energy inflation applied to the cost of electricity has been also varied.

Table 3.34 : Variation of DPP as a function of cooling energy cost - Solution AW-C01

	DPP [years]	Cooling energy cost [€/kWhf]						
	2,51	0,0629	0,0944	0,1258	**0,1573**	0,1888	0,2202	0,2517
Inflation [%]	0,80%	20,00	20,00	4,99	2,57	1,73	1,31	1,05
	1,20%	20,00	20,00	4,92	2,55	1,72	1,30	1,04
	1,60%	20,00	20,00	4,85	2,53	1,71	1,30	1,04
	2,00%	20,00	20,00	4,78	**2,51**	1,70	1,29	1,03
	2,40%	20,00	20,00	4,72	2,49	1,69	1,28	1,03
	2,80%	20,00	20,00	4,65	2,47	1,68	1,28	1,03
	3,20%	20,00	20,00	4,59	2,45	1,67	1,27	1,02

Table 3.35 : Variation of DPP as a function of cooling energy cost - Solution WW-C02

	DPP [years]	Cooling energy cost [€/kWhf]						
	8,18	0,0629	0,0944	0,1258	**0,1573**	0,1888	0,2202	0,2517
Inflation [%]	0,80%	20,00	20,00	20,00	10,10	3,99	2,50	1,82
	1,20%	20,00	20,00	20,00	9,30	3,90	2,47	1,80
	1,60%	20,00	20,00	20,00	8,68	3,82	2,44	1,79
	2,00%	20,00	20,00	20,00	**8,18**	3,75	2,41	1,77
	2,40%	20,00	20,00	20,00	7,76	3,68	2,38	1,76
	2,80%	20,00	20,00	20,00	7,40	3,61	2,36	1,74
	3,20%	20,00	20,00	20,00	7,09	3,55	2,33	1,73

Table 3.36 : Variation of DPP as a function of cooling energy cost - Solution ABS-C03

	DPP	Cooling energy cost [€/kWhf]						
	18,50	0,0629	0,0944	0,1258	**0,1573**	0,1888	0,2202	0,2517
Inflation [%]	0,80%	20,00	20,00	20,00	20,00	6,73	3,94	2,79
	1,20%	20,00	20,00	20,00	20,00	6,42	3,85	2,75
	1,60%	20,00	20,00	20,00	20,00	6,15	3,77	2,72
	2,00%	20,00	20,00	20,00	**18,50**	5,91	3,69	2,68
	2,40%	20,00	20,00	20,00	14,53	5,70	3,62	2,64
	2,80%	20,00	20,00	20,00	12,26	5,51	3,56	2,61
	3,20%	20,00	20,00	20,00	10,75	5,34	3,49	2,58

It can be seen that the time of return of investment is little dependent on the inflation rate, while it is very dependent on the cost of thermal energy, especially in solutions with higher DPPs. In fact, in solution 2 and 3 we can respectively from values of DPP equal to 8,18 and 18,5 to values of 3,75 and 5,91, are increasing by 20% the sales cost of refrigerating energy.

3.3.2. Electricity cost

For the first solution the most involved parameter is the cost of electricity, being the only energy carrier in the solution AW-C01.

As can been seen in table 3.36, where the change in the discounted return time DDP is evaluated, the increase in the cost of electricity leads to a significant lengthening of the return on investment. The analysis also varied the rate of energy inflation applied to the cost of electricity, and as can be seen, the PBP is little affected by it.

Table 3.37 : Variation of DPP as a function of electricity cost - Solution AW-C01

	DPP [years]	Electricity cost [€/kWhe]						
	2,51	0,0988	0,1482	0,1976	**0,2471**	0,2965	0,3459	0,3953
Inflation [%]	0,80%	1,46	1,70	2,05	2,57	3,45	5,26	11,26
	1,20%	1,45	1,69	2,03	2,55	3,42	5,18	10,84
	1,60%	1,44	1,68	2,02	2,53	3,38	5,10	10,47
	2,00%	1,43	1,67	2,00	**2,51**	3,35	5,03	10,13
	2,40%	1,43	1,66	1,99	2,49	3,32	4,96	9,83
	2,80%	1,42	1,65	1,98	2,47	3,28	4,89	9,55
	3,20%	1,41	1,64	1,97	2,45	3,25	4,82	9,29

Also for the solution two (WW-C02) was carried out an analysis going to vary the cost of electricity and its inflation, table 3.37. As for solution 1, inflation does not lead to a significant change in the DPP, on the contrary of the change in the cost of energy that leads the DPP to fluctuate in just over 3 years to a time over 20 years.

Table 3.38 :Variation of DPP as a function of electricity cost - Solution WW-C02

DPP [years]		Electricity cost [€/kWhe]						
	8,18	0,0988	0,1482	0,1976	**0,2471**	0,2965	0,3459	0,3953
Inflation [%]	0,80%	3,38	4,33	6,04	10,10	20,00	20,00	20,00
	1,20%	3,32	4,22	5,81	9,30	20,00	20,00	20,00
	1,60%	3,26	4,13	5,61	8,68	17,98	20,00	20,00
	2,00%	3,21	4,04	5,43	**8,18**	15,38	20,00	20,00
	2,40%	3,15	3,95	5,26	7,76	13,68	20,00	20,00
	2,80%	3,11	3,87	5,12	7,40	12,45	20,00	20,00
	3,20%	3,06	3,80	4,98	7,09	11,50	20,00	20,00

3.3.3. Water cost

For the second and third solutions, which involve the use of evaporative towers, the variation of the DPP has been analysed, changing the specific cost of water and the inflation associated with it, table 3.38 and table 3.39.

Table 3.39 : Variation of DPP as a function of water cost - Solution WW-C02

DPP [years]		Water cost [€/m3]						
	8,18	1,88	2,82	3,76	**4,70**	5,64	6,58	7,52
Inflation [%]	0,20%	4,23	5,05	6,23	8,03	10,93	15,81	20,00
	0,30%	4,23	5,06	6,25	8,08	11,06	16,15	20,00
	0,40%	4,24	5,07	6,27	8,13	11,19	16,52	20,00
	0,50%	4,24	5,08	6,29	**8,18**	11,32	16,91	20,00
	0,60%	4,25	5,09	6,32	8,23	11,47	17,33	20,00
	0,70%	4,25	5,10	6,34	8,28	11,62	17,79	20,00
	0,80%	4,25	5,10	6,36	8,34	11,78	18,30	20,00

Table 3.40 : Variation of DPP as a function of water cost - Solution ABS-C03

DPP		Water cost [€/m3]						
	18,50	1,88	2,82	3,76	**4,70**	5,64	6,58	7,52
Inflation [%]	0,20%	4,75	6,13	8,50	16,55	20,00	20,00	20,00
	0,30%	4,76	6,15	8,56	17,13	20,00	20,00	20,00
	0,40%	4,76	6,17	8,62	17,78	20,00	20,00	20,00
	0,50%	4,77	6,18	8,68	**18,50**	20,00	20,00	20,00
	0,60%	4,78	6,20	8,74	19,30	20,00	20,00	20,00
	0,70%	4,78	6,22	8,81	20,00	20,00	20,00	20,00
	0,80%	4,79	6,24	8,87	20,00	20,00	20,00	20,00

Also in this case, the change in inflation does not affect the return time; on the contrary, it is very important the change that leads to an increase or reduction in the cost of water on the DPP. For solution 3, which provides for an evaporative tower with the highest water consumption, a single 20% increase in the cost of water leads, even with an inflation of 0,20%, to a return time of capital that goes beyond the useful life of the solution.

However, it should be stressed that the cost of water should not fluctuate over the years, as it could be for electricity, for example. The cost of water should not increase too much and at the same time, low costs can only be found in situations where we have a rainwater recovery system.

3.3.4. Electricity and natural gas cost

Finally, for the third solution, an analysis is made considering the variation of the two most involved costs: the cost of electricity and the cost of natural gas.

It should be remembered that in this solution an increase in the cost of electricity taken from the grid, also leads to an increase in the cost of selling electricity.

Looking in table 3.40 as the time of return of the investment varies, it can be noticed that the variation of the natural gas has the greater weight.

It is also important to see how for the assumed conditions of energy cost, the investment is not at all convenient, while, with a fluctuation of -20% on the cost of natural gas, at the same cost of electricity, the investment is immediately much more advantageous.

Table 3.41 : Variation of DPP as a function of natural gas cost and electricity cost - Solution ABS-C03

	DPP [years]	**Natural gas cost [€/Sm3]**						
	18,50	0,2572	0,3858	0,5144	**0,6431**	0,7716	0,9002	1,0288
Electricity cost [€/kWhe]	0,0988	6,99	20,00	20,00	20,00	20,00	20,00	20,00
	0,1482	3,94	8,41	20,00	20,00	20,00	20,00	20,00
	0,1976	2,74	4,38	10,98	20,00	20,00	20,00	20,00
	0,2471	2,09	2,94	4,91	**18,50**	20,00	20,00	20,00
	0,2965	1,69	2,21	3,17	5,60	20,00	20,00	20,00
	0,3459	1,42	1,77	2,34	3,45	6,49	20,00	20,00
	0,3953	1,22	1,48	1,85	2,49	3,78	7,71	20,00

The coefficient of convenience of cogeneration is then evaluated, table 3.41.

It can be noted that both parameters greatly affect the convenience, in fact cogeneration is convenient both in the event that, at the same cost of natural gas, the cost of electricity rises, and both in the case where, at the same cost of electricity, the cost of gas was low.

Table 3.42 : Variation of K_{cog} as a function of natural gas cost and electricity cost - ABS-C03

	K_{cog}	**Natural gas cost [€/Sm3]**						
	0,38	0,2572	0,3858	0,5144	**0,6431**	0,7716	0,9002	1,0288
Electricity cost [€/kV	0,0988	0,38	0,26	0,19	0,15	0,13	0,11	0,10
	0,1482	0,58	0,38	0,29	0,23	0,19	0,16	0,14
	0,1976	0,77	0,51	0,38	0,31	0,26	0,22	0,19
	0,2471	0,96	0,64	0,48	**0,38**	0,32	0,27	0,24
	0,2965	1,15	0,77	0,58	0,46	0,38	0,33	0,29
	0,3459	1,34	0,90	0,67	0,54	0,45	0,38	0,34
	0,3953	1,54	1,02	0,77	0,61	0,51	0,44	0,38

3.3.5. Maintenance cost

Another parameter that has been studied in the sensitivity analysis is the cost associated with **maintenance**. For all the three solutions the variation that it involves on the time of return of the capital is not relevant, table 3.42, 3.43 and 3.44.

Table 3.43 : Variation of DPP as a function of maintenance - Solution AW-C01

	DPP [years]	Maintenance cost [€/y]						
	2,51	2.000	4.000	6.000	**8.000**	10.000	12.000	14.000
Inflation [%]	0,80%	2,36	2,41	2,45	2,51	2,56	2,62	2,67
	1,20%	2,36	2,41	2,46	2,51	2,56	2,62	2,68
	1,60%	2,36	2,41	2,46	2,51	2,56	2,62	2,68
	2,00%	2,36	2,41	2,46	**2,51**	2,57	2,62	2,68
	2,40%	2,36	2,41	2,46	2,51	2,57	2,63	2,69
	2,80%	2,36	2,41	2,46	2,51	2,57	2,63	2,69
	3,20%	2,36	2,41	2,46	2,52	2,57	2,63	2,69

Table 3.44 : Variation of DPP as a function of maintenance - Solution WW-C02

	DPP [years]	Maintenance cost [€/y]						
	8,18	9.000	11.000	13.000	**15.000**	17.000	19.000	21.000
Inflation [%]	0,80%	6,92	7,22	7,55	7,90	8,28	8,69	9,14
	1,20%	6,97	7,28	7,62	7,99	8,38	8,82	9,29
	1,60%	7,02	7,35	7,70	8,08	8,50	8,96	9,46
	2,00%	7,08	7,41	7,78	**8,18**	8,62	9,11	9,65
	2,40%	7,13	7,48	7,86	8,28	8,75	9,27	9,85
	2,80%	7,19	7,55	7,95	8,40	8,89	9,45	10,07
	3,20%	7,25	7,63	8,05	8,52	9,04	9,64	10,33

Table 3.45 : Variation of DPP as a function of maintenance - Solution ABS-C03

	DPP	Maintenance cost [€/y]						
	18,50	14.000	16.000	18.000	**20.000**	22.000	24.000	26.000
Inflation [%]	0,80%	11,61	12,50	13,47	14,52	15,64	16,85	18,15
	1,20%	12,19	13,20	14,31	15,53	16,85	18,28	19,82
	1,60%	12,89	14,06	15,36	16,80	18,40	20,00	20,00
	2,00%	13,73	15,13	16,71	**18,50**	20,00	20,00	20,00
	2,40%	14,82	16,55	18,56	20,00	20,00	20,00	20,00
	2,80%	16,30	18,58	20,00	20,00	20,00	20,00	20,00
	3,20%	18,53	20,00	20,00	20,00	20,00	20,00	20,00

3.3.6. Initial investment

Another sensitivity analysis is carried out by varying the **initial investment cost** of ± 25%. Indeed, the cost was established on the basis of assumptions and therefore errors may have been introduced. Looking at table 3.45, 3.46 and 3.47 can been seen that the DPP varies, not significantly for the first and second solution, significantly for the third solution, where the investment is higher.

Table 3.46 : Variation of DPP as a function of initial investment - Solution AW-C01

DPP [years]		**Initial investment [€]**	
2,51	172.500	**230.000**	287.500
	1,86	2,51	3,17

Table 3.47 : Variation of DPP as a function of initial investment - Solution WW-C02

DPP [years]		**Initial investment [€]**	
8,18	217.500	**290.000**	362.500
	6,08	8,18	10,33

Table 3.48 : Variation of DPP as a function of initial investment - Solution ABS-C03

DPP [years]		**Initial investment [€]**	
18,50	315.000	**420.000**	525.000
	10,64	18,50	20,00

3.3.7. SEER of air-cooled and water-cooled chiller

The last sensitivity analysis that has been carried out, concerns the seasonal efficiency coefficients SEER associated with the vapor compression machines. In the modelling of the two systems, have already been considered as SEER less than those declared by the producers, in order to consider a more truthful condition, but also, machine efficiencies could be reduced over lifetime or due to unforeseen climatic changes.

With this analysis is evaluated what is the threshold within which it is convenient to have an air-cooled or water-cooled chiller.

A $\Delta DDP\ [years]$ parameter is defined which represents the difference between the DDP of the WW-C02 solution (water-cooled chiller) and the DPP of the AW-C01 solution (air-cooled chiller); when the difference is positive, then solution 2 is more convenient, whereas if it turns out negative solution 1 is convenient. To define the convenience, the SEERs of the two machines are varied.

The start is a situation where solution 1 is more convenient, since, as also shown above in table 3.48, it has a lower DPP.

Table 3.49 : Variation of DPP as a function of SEER

ΔDDP [$years$]		Air-water chiller - SEER				
	5,67	2,22	2,96	**3,70**	4,44	5,17
Water-cooled chiller - SEER	3,13	7,22	16,41	17,49	17,91	18,13
	4,17	2,99	12,19	13,27	13,69	13,91
	5,22	-4,61	4,59	**5,67**	6,09	6,31
	5,74	-5,90	3,30	4,38	4,80	5,02
	6,26	-6,71	2,48	3,56	3,99	4,21

As can be seen from the table 3.48, there is a SEER threshold value of solution 1, below which the most cost-effective solution is not reused. Through the Goal Seek of Excel Software, keeping the SEER of the WW-C02 solution unchanged, it turns out that under a $SEER_{AW} = 2{,}35$ the water-cooled solution is convient, while keeping the SEER of the AW-C01 solution unchanged, the solver does not find a solution, which means that WW-C02 will never be cheaper than solution 1.

The same scenario occurs if the difference between the operating costs of the two solutions (1 - 2) varies, parameter $\Delta Cost$ [k€], table 3.49. When it turns out positive, it means the solution 1 is that with the costs minor and almost in all the cases, when the operating costs are greater, then that solution is the less convenient. At the $SEER_{AW}$ equal to 2,22 and $SEER_{WW}$ equal 4,17, looking at the ΔDDP, solution 1 is always the most convenient, while looking at the $\Delta Cost$, the higher operating cost has the solution 1; Solution 1 therefore remains the most convenient in this case, because it has lower investment costs.

Table 3.50 : Variation of cost as a function of SEER

$\Delta Cost$ [k€]		Air-water chiller - SEER				
	1.214	2,22	2,96	**3,70**	4,44	5,17
Water-cooled chiller - SEER	3,13	674	1831	2525	2987	3318
	4,17	-145	1011	1705	2168	2498
	5,22	-637	520	**1214**	1676	2007
	5,74	-816	341	1035	1497	1828
	6,26	-965	192	886	1348	1679

4 Conclusion and future developments

In this last chapter will be drawn the conclusions of this thesis going to analyse in particular two aspects. In a first paragraph the models of the three solutions studied will be compared, while in a second chapter some considerations will be made in merit to possible future developments deriving from this analysis.

4.1. Results and considerations

During this thesis, three different solutions for the production of cold water were analysed: an air-cooled chiller, a water-cooled chiller and an absorption chiller.

For each of them an ideal model was created and for each component in play, an energy consumption was associated and subsequently an economic analysis was carried out.

The first can be a comparison of the specific costs for cold water production, graph 4.1. It can be noted that the first solution is the most cost-effective, while the second and third solutions involve a very similar specific cost. It is recalled that electricity sold on the grid was considered as a cost discount for the calculation of the specific cost.

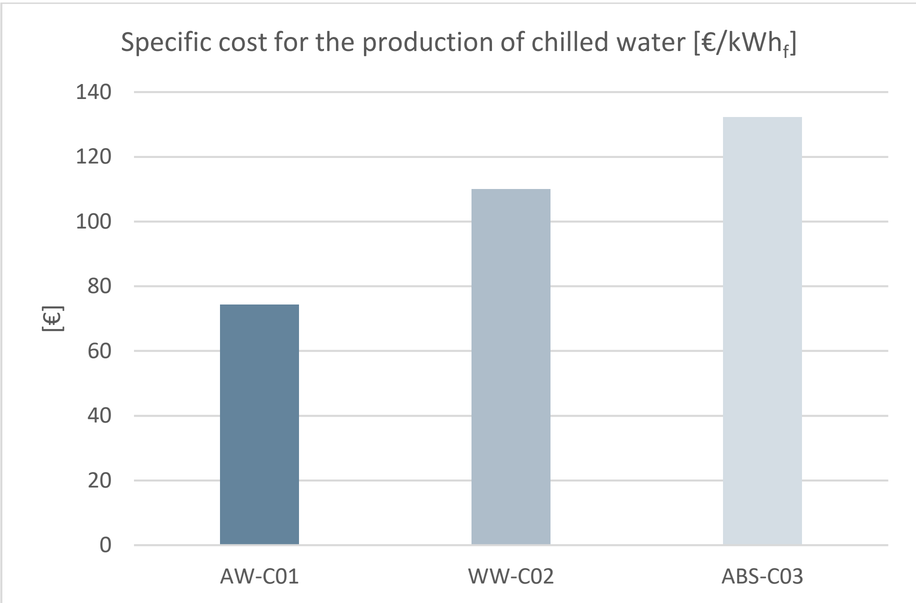


Figure 4.1 : Specific cost for the production of chilled water for the three solutions

The AW-C01 solution is more cost-effective by also evaluating operating and maintenance costs alone, graphic 4.2.

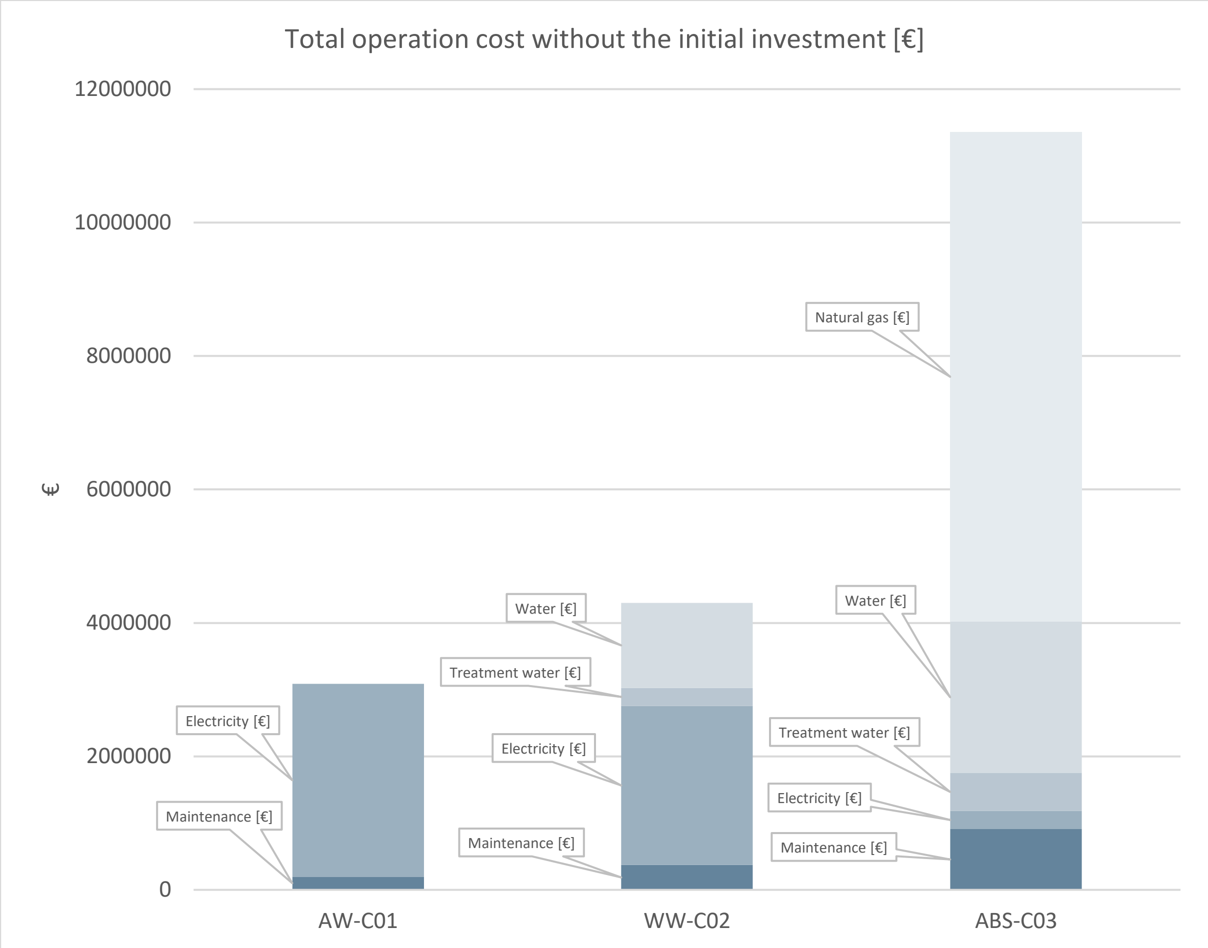


Figure 4.2 : Total operation cost of the three solutions

In fact, for the two solutions that involve the use of evaporative towers, the cost of water plays a very important role, significantly increasing costs; Moreover, for the third solution there is also all the cost derived from the use of natural gas as a source of power.

The convenience of the air-cooled solution is also reflected in economic indices such as the Net present value [€] and the Discounted payback period [year], graph 4.3.

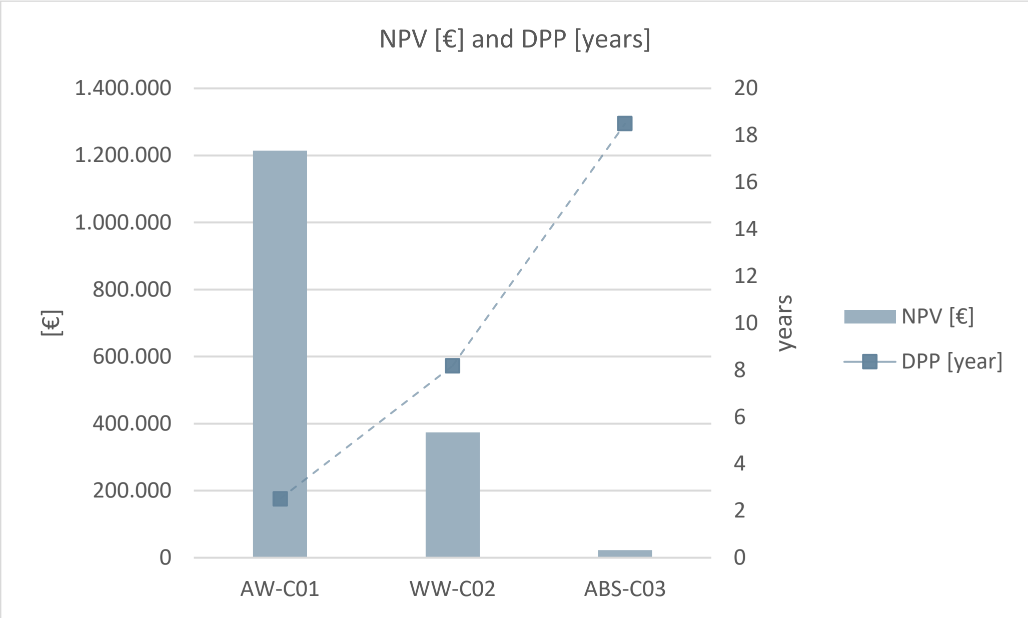


Figure 4.3 : Net present value and payback period of the three solutions

From the analysis of sensitivity instead, what can be drawn is that the cash flows and therefore the convenience or not of the investment, are strongly dependent on the cost of energy and water.

In particular, in the case of solution 1 there is a dependence exclusively on the price of electricity, for the solution 2 there is also a dependence on the cost of water. The third solution sees the worst case, where the investment depends on the cost of electricity, water and natural gas. Having much more price uncertainty makes investments less attractive and more risk-prone, especially at a time when energy prices are highly variable.

For the third solution is also very important the relationship between electricity and natural gas, which represents the convenience of cogeneration: in this solution, compared to the others, it is convenient to increase the cost of electricity, as it is also sold.

Indeed, considering the sensitivity analysis for the third solution, looking at how DPP and K_{cog} vary according to the cost of electricity and natural gas, it can be noted that there are smaller return times at K tending to or greater than 1, which means when the cost of electricity tends or exceeds that of natural gas.

In general, the solution with the absorption unit is never convenient to cover all the load required by the network.

4.2. Future developments

The drafting of this thesis was born from the study and evaluations made on the high efficiency trigeneration plant serving the district cooling and cooling network of the Navile district. Currently, as for the production of cold water, there are two water-cooled chillers, with two evaporative towers.

By modelling and studying the existing solution and further possibilities for cold water production, in addition to comparing them, the objective is also to go to understand what the possible developments regarding the technological plant of the Navile could be, based on the evaluations made.

The aspects that will be analysed are two:

- an extension of the plant
- a revamping of the plant

The district heating and cooling network has been designed to provide 17 users and at the moment the plant provides 8; it is therefore a network in development and in the future, most likely, there will be the need to expand the refrigeration plant, adding a new fridge group. On the basis of the economic and energy study done, it can be said that it is definitely not convenient to go and install an absorption group, given the high initial investment and the associated costs and also, given the high cost associated with cooling towers, deriving from the cost and the consumption of water for reintegration, it is not convenient to install an additional water-cooled cooling circuit.

A valid option in case of expansion of the refrigeration plant will therefore be the installation of air-cooled chillers, which has lower investment costs and a shorter return time of capital; moreover, it does not involve the installation of an additional structure, like the cooling tower.

Another aspect that can be evaluated is a possible revamping of the system, whether for a reason of technological evolution or because the refrigeration unit or cooling towers reach the end of their useful life.

For example, if a refrigeration unit were to be replaced, looking at the evaluations made previously, it might not be convenient to replace a water-cooling unit with the same one, risking then a few years after also having to replace an evaporative tower, since the power plant was built about 10 years ago. Instead, it might be more convenient to replace the refrigeration units with an air-cooled chiller.

The absorption group would not be convenient here either. Moreover, with regard to the use of the absorber, it is convenient only if the thermal energy comes from a cogeneration unit, because otherwise, without a revenue from the electricity sold and white certificates, and given the high cost given by natural gas, would never shift investment costs.

To summarize, the solution with an air-cooled chiller is always convenient both from the point of view of the initial investment and for the specific costs related to the production of chilled water.

The negative side brought by the air condenser is the oscillation of the condensation temperature, which is more stable if water is considered as a means of cooling. Greater stability in condensation temperature results in higher efficiency. This aspect is already considered in the seasonal efficiency index SEER that has been used for the calculation of the electricity associated with each refrigeration group. The SEER is in fact lower for the air condenser than the water condenser.

Moreover, through the sensitivity analysis carried out in the previous chapter we can see how, despite the lower values of seasonal efficiency, the air condensing machine is in most cases convenient.

In conclusion, many economic and technical assessments have been carried out in this thesis and considerations have been made on the basis of them.

However, it should be borne in mind that when someone really has to make decisions about the construction of a new plant or about the revamping of an existing plant, energy and economic issues alone cannot be considered, but together with them is linked to environmental issues, assessments of the available space, on the site of the plant and many other aspects, but that are outside the study in question.

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A Appendix A

In this appendix are reported the annual results of costs, revenues, and cash flow for each solution.

A.1. Air-cooled chiller

Below the calculations for the first solution AW-C01, the air-cooled chiller. In table A.1 the calculation of costs and the revenues, yearly; in table A.2 the calculation of cash flow, NPV yearly and cumulative cash flow.

Table A. 1: Calculation of costs and revenues for the solution AW-C01

		Maintenance	Utilities	Revenues
Year	**Initial investment [€]**	**AW-C01 [€]**	**Electricity [€]**	**Cooling energy [€]**
0	−230.000	0	0	0
1	0	−8.000	−116.547	221.060
2	0	−8.000	−116.547	221.060
3	0	−8.000	−116.547	221.060
4	0	−8.000	−116.547	221.060
5	0	−8.000	−116.547	221.060
6	0	−8.000	−116.547	221.060
7	0	−8.000	−116.547	221.060
8	0	−8.000	−116.547	221.060
9	0	−8.000	−116.547	221.060
10	0	−8.000	−116.547	221.060
11	0	−8.000	−116.547	221.060
12	0	−8.000	−116.547	221.060
13	0	−8.000	−116.547	221.060
14	0	−8.000	−116.547	221.060
15	0	−8.000	−116.547	221.060
16	0	−8.000	−116.547	221.060
17	0	−8.000	−116.547	221.060
18	0	−8.000	−116.547	221.060
19	0	−8.000	−116.547	221.060
20	0	−8.000	−116.547	221.060

Table A. 2: Calculation of cash flow and NPVj for the solution AW-C01

Year	Cash flow [€]	NPVj	Cumulative cash flow
0	−230.000	−230.000	−230.000
1	98.443	93.755	−136.245
2	100.412	91.077	−45.168
3	102.420	88.474	43.307
4	104.469	85.947	129.253
5	106.558	83.491	212.744
6	108.689	81.106	293.850
7	110.863	78.788	372.638
8	113.080	76.537	449.175
9	115.342	74.350	523.526
10	117.649	72.226	5959.752
11	120.002	70.162	665.914
12	122.402	68.158	734.072
13	124.850	66.210	800.283
14	127.347	64.319	864.601
15	129.894	62.481	927.082
16	132.492	60.696	987.778
17	135.141	58.962	1.046.740
18	137.844	57.277	1.104.017
19	140.601	55.641	1.159.658
20	143.413	54.051	1.213.709

A.2. Water-cooled chiller

Below the calculations for the second solution WW-C02, the water-cooled chiller, table A.3 and A.4.

Table A. 3: Calculation of costs and revenues for the solution WW-C02

		Maintenance		Utilities		Revenues
Year	Initial investment [€]	WW-C02 [€]	Water treatment [€]	Electricity [€]	Water [€]	Cooling energy [€]
0	−290.000	0	0	0	0	0
1	0	−15.000	−12.849	−96.205	−60.390	221.060
2	0	−15.000	−12.849	−96.205	−60.390	221.060
3	0	−15.000	−12.849	−96.205	−60.390	221.060
4	0	−15.000	−12.849	−96.205	−60.390	221.060
5	0	−15.000	−12.849	−96.205	−60.390	221.060
6	0	−15.000	−12.849	−96.205	−60.390	221.060
7	0	−15.000	−12.849	−96.205	−60.390	221.060

8	0	−15.000	−12.849	−96.205	−60.390	221.060
9	0	−15.000	−12.849	−96.205	−60.390	221.060
10	0	−15.000	−12.849	−96.205	−60.390	221.060
11	0	−15.000	−12.849	−96.205	−60.390	221.060
12	0	−15.000	−12.849	−96.205	−60.390	221.060
13	0	−15.000	−12.849	−96.205	−60.390	221.060
14	0	−15.000	−12.849	−96.205	−60.390	221.060
15	0	−15.000	−12.849	−96.205	−60.390	221.060
16	0	−15.000	−12.849	−96.205	−60.390	221.060
17	0	−15.000	−12.849	−96.205	−60.390	221.060
18	0	−15.000	−12.849	−96.205	−60.390	221.060
19	0	−15.000	−12.849	−96.205	−60.390	221.060
20	0	−15.000	−12.849	−96.205	−60.390	221.060

Table A. 4: Calculation of cash flow and NPVj for the solution WW-C02

Year	**Cash flow [€]**	**NPVj**	**Cumulative cash flow**
0	−290.000	−290.000	−290.000
1	38.255	36.433	−253.567
2	39.930	36.218	−217.349
3	41.644	35.973	−181.376
4	43.396	35.702	−145.674
5	45.188	35.406	−110.268
6	47.020	35.087	−75.181
7	48.894	34.748	−40.433
8	50.810	34.390	−6.042
9	52.769	34.015	27.973
10	54.772	33.625	61.598
11	56.820	33.221	94.820
12	58.913	32.805	127.624
13	61.053	32.378	160.002
14	63.240	31.941	191.943
15	65.477	31.495	223.438
16	67.762	31.043	254.481
17	70.099	30.584	285.065
18	72.487	30.120	315.184
19	74.927	29.651	344.836
20	77.422	29.179	374.015

A.3. Absorption chiller

Below the calculations for the third solution ABS-C03, the absorption chiller, table A.5 and A.6.

Table A. 5: Calculation of costs and revenues for the solution ABS-C03

		Maintenance			Utilities			Revenues		
Year	Initial investment [€]	ABS-C03 [€]	COG03 [€]	Water treatment [€]	Electricity [€]	Water [€]	Natural gas [€]	Electricity [€]	Cooling energy [€]	CB [€]
0	-420.000	0	0	0	0	0	0	0	0	0
1	0	-20.000	-16.754	-22.908	-10.977	-107.667	-296.027	252.624	221.060	31.730
2	0	-20.000	-16.754	-22.908	-10.977	-107.667	-296.027	252.624	221.060	31.730
3	0	-20.000	-16.754	-22.908	-10.977	-107.667	-296.027	252.624	221.060	31.730
4	0	-20.000	-16.754	-22.908	-10.977	-107.667	-296.027	252.624	221.060	31.730
5	0	-20.000	-16.754	-22.908	-10.977	-107.667	-296.027	252.624	221.060	31.730
6	0	-20.000	-16.754	-22.908	-10.977	-107.667	-296.027	252.624	221.060	31.730
7	0	-20.000	-16.754	-22.908	-10.977	-107.667	-296.027	252.624	221.060	31.730
8	0	-20.000	-16.754	-22.908	-10.977	-107.667	-296.027	252.624	221.060	31.730
9	0	-20.000	-16.754	-22.908	-10.977	-107.667	-296.027	252.624	221.060	31.730
10	0	-20.000	-16.754	-22.908	-10.977	-107.667	-296.027	252.624	221.060	31.730
11	0	-20.000	-16.754	-22.908	-10.977	-107.667	-296.027	252.624	221.060	0
12	0	-20.000	-16.754	-22.908	-10.977	-107.667	-296.027	252.624	221.060	0
13	0	-20.000	-16.754	-22.908	-10.977	-107.667	-296.027	252.624	221.060	0
14	0	-20.000	-16.754	-22.908	-10.977	-107.667	-296.027	252.624	221.060	0
15	0	-20.000	-16.754	-22.908	-10.977	-107.667	-296.027	252.624	221.060	0
16	0	-20.000	-16.754	-22.908	-10.977	-107.667	-296.027	252.624	221.060	0
17	0	-20.000	-16.754	-22.908	-10.977	-107.667	-296.027	252.624	221.060	0
18	0	-20.000	-16.754	-22.908	-10.977	-107.667	-296.027	252.624	221.060	0
19	0	-20.000	-16.754	-22.908	-10.977	-107.667	-296.027	252.624	221.060	0
20	0	-20.000	-16.754	-22.908	-10.977	-107.667	-296.027	252.624	221.060	0

Table A. 6: Calculation of cash flow and NPVj for the solution ABS-C03

Year	Cash flow [€]	NPVj	Cumulative cash flow
0	-420.000	-420.000	-420.000
1	32.682	31.126	-388.874
2	34.324	31.133	-357.741
3	36.007	31.104	-326.637
4	37.732	31.042	-295.595
5	39.500	30.949	-264.646
6	41.311	30.827	-233.819
7	43.166	30.678	-203.142

8	45.068	30.503	-172.638
9	47.015	30.306	-142.332
10	49.010	30.088	-112.244
11	19.323	11.298	-100.946
12	21.415	11.925	-89.021
13	23.558	12.493	-76.528
14	25.753	13.007	-63.521
15	28.000	13.468	-50.053
16	30.300	13.881	-36.172
17	32.655	14.247	-21.925
18	35.066	14.571	-7.354
19	37.534	14.854	7.500
20	40.060	15.098	22.598

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