UNIVERSITA' DEGLI STUDI DI PADOVA

Dipartimento di Ingegneria Industriale DII

Corso di Laurea Magistrale in Energy Engineering

Techno-economic analysis of possible implementations to the secondary circuit of an existing district heating and cooling plant

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Al gruppo "Panzerotti":

"Il mondo appartiene

a chi se lo prende"

ABSTRACT

The distribution of heat and cold from renewable sources via district heating and cooling is of significant environmental importance. Furthermore, this technology is capable of utilising waste heat, thereby ensuring optimal performance. This is due to the scaling effect, which guarantees lower emissions, and the fact that the companies responsible for its operation demonstrate a superior level of management and maintenance.

This paper will analyse an existing district heating and cooling system in order to compare some possible solutions to be implemented for the end user. In particular starting from the summer and winter thermal energy needs, some types of plant will be designed and then analysed both from technical and economical point of view, in order to give a broad view of the solutions and their relative pros and cons. Alongside this, a remote meter reading and data analysis system was also implemented, so as to have a precise description of the losses from the power station to the end user. In this way, it will be easier to understand where the system can be improved.

Through this analysis, advantages can be gained both for the plant operator, who can sample and analyse energy consumption and flows in the network in real time, and for the end user, who can decide which of the solutions is the most convenient.

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SYMBOLS AND ABBREVIATIONS

Symbol	*Description*	*Unit*
A	area	m^2
c	specific heat capacity	J/(kg K)
C	shading coefficient	-
D	diameter	m
E	energy demand	kWh
f	factor	-
G	flow rate	m^3/h
h	height	m
I	maximum incident solar radiation	kWh/m^2
L	length	m
M	thermal capacity	kg/m^2
m	mass	kg
n	air change rate	h^{-1}
p	perimeter	m
P	electric power	W
q	heat flux	W
Q	thermal energy	W
R	thermal resistance	$(m^2 K)/W$
s	thickness	m
U	transmittance	$W/(m^2 K)$
v	velocity	m/s
V	volume	m^3
α	overall heat transfer coefficient	$W/(m^2 K)$
θ	temperature	K
λ	conductivity	W/(m K)
ρ	volumetric mass	kg/m^3
ψ	linear thermal transmittance	W/(m K)

PEDIX

Pedix	*Description*
a	attenuation
adj	adjacent
amb	ambient
calc	calculated
des	design
ext	external
f	frame
fl	floor
g	glass
grad	gradient
i	i-th
ig	internal gain
inf	infiltration
int	internal
imm	inlet
occup	occupied
op	opaque
ppl	people
req	required
ret	return
se	external side
si	internal side
sol	solar
sp	specific
sum	summer
sup	supply
surf	surface
T	transmission losses
tot	overall
u	upper
V	ventilation
w	window
wat	water
win	winter
wl	wall

APEX

Apex	*Description*
*	average
#	corrected

1. INTRODUCTION

In recent years, the topic of sustainability has become very important both economically and politically. Especially after the COVID pandemic, the energy independence and green transition have become increasingly prominent. Within this global framework the air treatment of domestic, commercial and industrial environments fits in.

Prior to the problem of air treatment, which includes winter heating, summer cooling and air quality, in order to achieve energy-saving goals it is necessary to start working on building insulation so that the heat fluxes to be compensated for are reduced and with it also the energy demand.
Broadening the perspective to the energy sector the EU, through the Energy Performance of Buildings Directive (EPBD) approved in May 2024, gave the path to achieve the energetic and climatic targets by progressively reducing greenhouse gas emissions and energy consumption in the building sector by 2030 and achieving climatic neutrality in 2050.

Along the way to achieve these challenging goals, the use of district heating and cooling plays an important role. In fact, district heating usually uses renewable technologies to produce both heating and cooling such as geothermal heat pump or absorption chiller.

1.1 Purpose of the thesis

The main purpose of this master thesis is to analyse a district heating and cooling system in Padova and find some solutions for the user-side air conditioning. In particular, in a district heating network such that of Padova, which has more than 3 km, the losses in the heat distribution section are not negligible. The aim is therefore to exploit the heat reaching the end user in the best possible way.

In this thesis, some possibilities for the end user will be examined with their pros and cons. The goal is to present at the customer possible solutions in terms of costs and benefits.

Concurrently, a network will be established to remotely read customers’ meters, thereby facilitating the acquisition of a comprehensive and dynamic consumption profile. Then, through the analysis of this trend it will be possible to understand where and how much losses are related to the customer's actual consumption. This provides a comprehensive overview of the plant, which in turn enables the identification of potential avenues for loss reduction and the optimisation of efficiency.

1.2 Forgreen spa benefit company

Forgreen Spa is a company headquartered in Verona that was established in 2009 and is engaged in the field of renewable energy and development in projects of sustainability. It passed from energetic operator 100% renewable to promoter of energy communities models for enterprises and people.

In 2019 it became a benefit company based on environmental and social sustainability, ethic, and responsibility in doing business. The main goal is to develop energy communities in such a way to create responsible consumption patterns. The paradigm of the society is the one of "*Civil Economy*": it considers the shared well-being as a cornerstone and the direction of the economic action [1].

The vision of the company is to develop business model based on the review of the value at stake:

- Sustainability
- Sharing
- Ethic
- Innovation

The mission of the company can be resumed in the following sentence:

> *"The energy is the instrument for growth, innovation and development new sustainable lifestyle"*

Forgreen spa itself is not defined as energetic operator but a developer and promotor of models for enterprises and people, where the energy is a vehicle for starting a sustainability path. The energetic communities of Forgreen are made up of people and companies that wish to consume clean energy they know where it comes from.

The main fields in which the company works are:

- Supply and withdrawing energy produced by shared facilities, traceable and certified
- Spread in the Italian market innovative e renewable energy models

- Develop the best solution of photovoltaic plant that feed and will feed energy communities of the model following also the technic and administrative management to maintain efficiency at high standard

Although Forgreen is involved photovoltaic energy, it also manages a district heating and cooling plant in Padova. This plant produces both heating in winter and cooling in summer and through a cogeneration engine it also produces electricity. This plant has been designed as a high-efficiency plant with the aim of reducing emissions to the environment serving hundreds of users.
The district heating management is not so simple because of several reasons that will be treated in the following part of this thesis.
The future perspectives of Forgreen are:

- Energetic auto-production
- Give the possibilities to produce energy at the highest possible number of people
- Reach an organization model with:
    - Exponential growth
    - Agility and adaptability
    - Innovation
    - Efficiency
    - Attraction of talents

1.3 Structure of the thesis

The first section of the thesis will present the overarching concept of district heating and cooling. Subsequently, the Padova plant will be explained in comprehensive detail, with a view to grasping the operational principle and the critical issues.

The second part of the thesis is devoted to an examination of the metering system. In particular, the present metering system and its potential implementation will be examined in order to obtain an accurate and intelligible remote reading of the plant's operation.

Subsequently, the thermal energy demand of a typical non-connected user is estimated and potential solutions for final users are presented. Each type of plant is studied in detail to gain a comprehensive understanding of their characteristics and limitations.

Finally, the advantages of remote reading are outlined, and an overview of the advantages and disadvantages of the different types of systems is provided.

2. DISTRICT HEATING AND COOLING: PRINCIPLES AND TECHNOLOGIES

This chapter presents a general historical overview of district heating and its developments with an analysis of the advantages and disadvantages. Subsequently, the particulars of the Padova plant (named TELEZIP) will be elucidated, accompanied by an explanation of each component, the configuration, and the current mode of operation and consumption reading.
Then, a brief overview of European and Italian regulations will be provided.

2.1 Historical overview of district heating, principle of operation, main concepts and possible issues

District heating (DH) can be defined as a system whereby heat is generated in a central location and then distributed to residences, businesses and industries in the surrounding area. District heating offers a substantial opportunity for the efficient, cost-effective and flexible large-scale utilisation of low-carbon energy for heating and cooling purposes [2].
A district heating and cooling system is a networked and localised system that includes many users and provides both heating and cooling, usually through the exploitation of renewables sources and/or waste heat. As a public system, it requires political action to facilitate its implementation.

The application of district heating technology has undergone a transformation in terms of both the heat transfer fluid employed and the temperature at which is utilised.
In particular the historical development can be summarised in five generations [3]:

- *1st generation*: in New York and Paris between the late 19th and early 20th centuries. The heat carrier fluid was steam at high temperature and the reason for the utilization of this technology was aimed to replace polluting coal boilers in big cities. The 1st generation was characterized by steam leakages, huge heat losses and corrosion
- *2nd generation*: in URSS between 20’s and 70’s. The DH was integrated in the planned economy programme, and it was “Production-driven” technology. The network was made by oversized pipes with no thermal insulation and the heat carrier fluid was superheated water (>100°C)
- *3rd generation*: in Scandinavian countries between 70’s and 2010. The DH was used to promote efficiency and energy security concerns. The plant was “Demand-driven”, and

the pipes were pre-insulated. The heat carrier fluid used in this generation was hot water (90/60 °C)

- *4th generation*: in Scandinavian countries, from 10's the temperature of the heat carrier fluid was reduced to address the reduction in heat demand and the growth of renewable technologies. The temperature was set at 70/40°C. Also in this generation the "Demand-driven" regulation was employed
- *5th generation*: in western Europe from 10's (parallel to the 4th gen.). Also in this case the objective was the reduction of the temperature of the heat carrier fluid under the 50°C. The salient features of the plant in question were the decentralisation of the reversible heat pumps, the possibility of simultaneous heating and cooling and the free floating temperature in the loop [4]

GEN.	*WHERE*	*WHY*	*FLUID*	*CHARACTERISTICS*
1st	New York, Paris	Replace polluting coal boilers in big cities	Steam	Steam leakages Huge heat losses Corrosion
2nd	URSS	Planned economy	Superheated water (>100°C)	Oversized pipes with no thermal insulation "Production-driven"
3rd	Scandinavian countries	Efficiency and energy security	Hot water (90/60°C)	Pre-insulated pipes "Demand-driven"
4th	Scandinavian countries	Heat demand reduction Renewables	Hot water (70/40°C)	Increased supply from renewable heat "Demand-driven"
5th	Western Europe	Heat demand reduction Renewables	Low temperature water (<50°C)	Decentralized reversible heat pumps Simultaneous heating and cooling possible

Table 2.1 – Resume of the history of the district heating

Nowadays the district heating has become a pervasive phenomenon in Central and Northern Europe, North America and in Japan. In Italy, district heating accounts for only 2,3% of the total heat demand.

In general district heating represents an alternative, environmentally friendly, safe and economical solution to produce domestic hot water (DHW) and for space heating for residential and commercial buildings. District heating is a system whereby heat is produced in a central power plant and then distributed to a neighbourhood or city. The distribution of heat within the buildings is facilitated by a network of pipes through which hot water or steam flows. It can be argued that this type of plant is only justifiable where the population density is sufficiently high to ensure that the thermal energy produced can be distributed via a small network, which is therefore likely to be inexpensive.

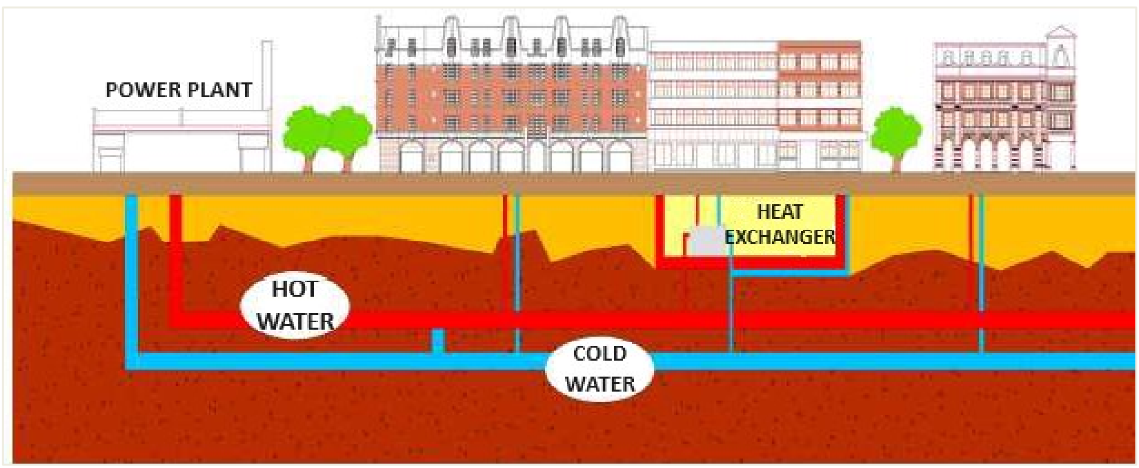

Figure 2.1 – General scheme of the plant [5]

The heat is generated at the power plant.
The distribution network is responsible for the transportation of the fluid to the substation, where the heat exchanger facilitates the transfer of heat from the primary circuit (from the power centre) to the secondary circuit (serving the user). The two circuits are separated, with one fluid allocated to each to prevent mixing. After the heat exchanger, the water in the primary circuit returns to the power centre to be brought back to the desired temperature.
The cycle then restarts.
The district heating network eliminates the need for the user to utilise methane, substituting the traditional boiler with the thermal substation, which only allows water to flow.

In this way, the customer's logic shifts from the fuel purchase to the heat purchase, together with maintenance, management and assistance.
Very often, in order for district heating to fully exploit its energy advantages, it is necessary to utilise a combined system that generates electricity and heat simultaneously. This technology is referred to as cogeneration.
Furthermore, the district heating service can be enhanced by the provision of a summer cooling service. This system is designated a "*trigenerative system*", as it is capable of providing three distinct forms of energy: heating, cooling and electricity. This configuration permits the cogeneration plant to be employed for the majority of the year, as the heat generated by the engine during the summer is utilised to feed the absorption chiller, thereby preventing its dissipation. Furthermore, the economic aspect is optimised since a vapour compression cycle with a high electricity requirement is not employed to produce the cool.

The energy assessment is conducted through the implementation of inspections and the consideration of specific parameters that ascertain the required thermal power. These parameters are typically:

a) climatic characteristics of the area
b) surface area served
c) type of buildings
d) state of the buildings

It is responsibility of the district heating operator to guarantee the service and save energy when the demand is low by ensuring that the pressure differential (Δp) is greater than the minimum pressure differential (Δp_{min}) for each customer.

One of the main parameters employed in the economic assessment of a district heating plant is the *linear heat density* (d) defined as the ratio between the heat demand and the length of the transmission pipes:

$$d = \frac{Q_T}{l_{net}} \quad (1)$$

The following table presents a series of illustrative examples of linear heat density, with the objective of providing an order of magnitude for the existing plant in Italian cities.

City	*Energy delivered to the buildings [MWh]*	*Overall network length [km]*	*Linear heat density [MWh/m]*
Asiago	9711	13.47	0.72
Brescia DH	981194	379.8	2.58
Brescia DC	32122	7.91	4.06
Ferrara	134816	82.58	1.63
Forni di sopra	1614	3.08	0.52
Telezip	1565	3.50	0.45
Torino	1790025	598.66	2.99
Verona	260395	80.63	3.23
Vicenza	38967	23.15	1.68

Table 2.2 – Linear heat density of some Italian cities

It is of the fundamental importance that an economic feasibility study be conducted to guarantee that the heat tariff (€/MWh) for the final user is identical (or lower) than that of alternative individual heat supply solutions (e.g. gas boilers). It is imperative that the heat generation cost for the utility be less than that of domestic users.

In order to determine the appropriate diameter for the district heating network pipes, the subsequent procedure should be followed [3]:

1. Estimate the target heat demand and peak load of the connected buildings + heat losses (in kW)
2. Use nominal ΔT (e.g. 30K) to find corresponding mass flow rate
3. Calculate diameter with either constant velocity or constant pressure losses

The distribution network may be constructed from steel or PE and HDPE. The two alternatives present a series of advantages and disadvantages, which are summarised in the following table.

Material	*Pros/cons*	*Description*
STEEL	Advantages	High strength Good flexibility Can be joined by welding Widely available in all sizes Familiar material to most workforces
	Disadvantages	Relatively high cost Highly susceptible to corrosion Skilled labour forces required for welding Slower installation
PE AND HDPE	Advantages	Low weight Very flexible Can be fusion welded Available in sizes up to 1.6 m Leak free and fully restrained
	Disadvantages	Lower strength that results in high thickness and thus large diameters Higher pressure losses Larger diameters fusion-welding machines may be of limited availability Cost fluctuates with oil price

Table 2.3 – Advantages and disadvantages of two materials for the distribution pipes

The configuration of the distribution line may vary in accordance with the design selected by the engineer:

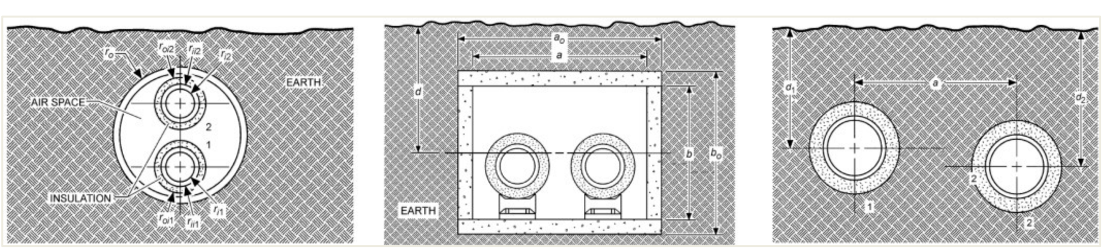

Figure 2.2 – Possible configuration of the distribution line

2.2 Case study: the TELEZIP district heating and cooling plant

The plant was built in the early years of the 21st century with the aim of redeveloping the area of the former train factory, "*Officine Meccaniche Stanga*", located on Corso Stati Uniti, 3 Padova. The primary objective was to ensure compliance with the requirements of sustainable development in economic, environmental and social terms. The specific location within the Veneto region has rendered it an increasingly optimal destination for the establishment of advanced manufacturing, logistics and service activities, which have attracted both Italian and international business operators to the city.

Figure 2.3 – Location of the plant

The system consists of an industrial district heating and cooling plant.
The main functions of the plant are as follows:

- The system provides district heating in winter and district cooling in summer. It produces and distributes heat transfer fluid via an underground network for the air conditioning of industrial buildings.
  In the winter mode, the system produces hot water through the use of cogenerators and/or condensing boilers; in the summer mode, the production of chilled water is obtained through electric chillers and/or absorption machines that recover the heat produced by cogenerators
- The system is equipped with high-efficiency power generation via cogenerators, with LV/MV elevation in the power station, with discharge to the MV cabin

The complete floor plan can be found in Appendix A.

The plant is constituted of three principal subsystems [5]:

1. *Central heating plant*

   The central heating plant encompasses all machinery and ancillary systems employed in the generation of thermal and cooling energy, as well as the production and distribution of electrical energy. This section includes: boilers, cogenerators, pumping systems and absorber. The entire control system of the plant is located here. The chiller, evaporative towers and the heat sink of the cogenerators are designed for outdoor use and they are placed on the roof

2. *Distribution line*

   The distribution line is completely insulated and buried between 1,7 and 0,5 meters below ground level. The pipes are made of steel for the primary sections and plastic for the final branches

3. *Substations*

   The distribution substations are situated in close proximity to the building units. They are made entirely of steel and housed in dedicated technical compartments. In the substation, a heat exchange process occurs between the heat transfer fluid coming from the power plant (primary) and that of the customer's air conditioning system (secondary) through a plate heat exchanger

The system comprises two principal manifolds for the collection of the heat transfer medium (water):

- One that feeds the supply network and from which the fluid will be sent to the substations
- One that receives the water returning from the substations

The network is of the branched type. This solution is the most straightforward to implement, offering the shortest route and consequently the lowest costs. It does not provide absolute assurance of reliability, as a disruption along a main branch can result in a crisis for all downstream users.

The plant has been properly oversized to guarantee the service required by the users in every situation. The overall efficiency of the system is not affected by this oversizing, as each component is modulating and controlled by a PLC. The primary objective of the control system is to facilitate the implementation of all necessary adjustments for the optimal and efficient operation of the plant, in response to fluctuations in load over time.

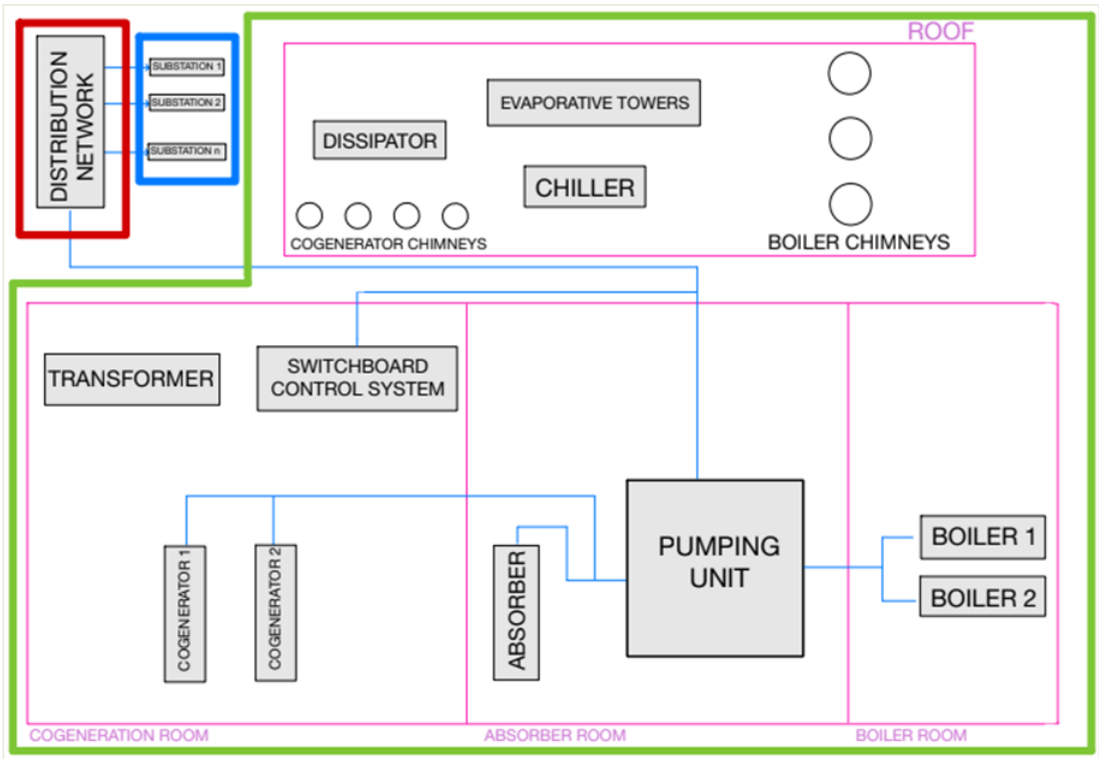

Figure 2.4 – Block scheme of the district heating plant of Padova

The central station that serves the district heating and cooling network in TELEZIP is constituted of the following elements:

- 2 methane gas-fuelled cogenerators
- 2 methane gas-fuelled boilers
- 1 compressor refrigeration unit
- 1 absorption refrigeration unit
- 1 plate heat exchanger for recovering thermal energy from the cogenerator
- 1 evaporative tower serving the absorber
- 1 dry-cooler for possible dissipation of recovered thermal energy from the cogenerator

The following section provides a concise overview of these components.

2.2.1 Pumping and expansion system

The mean supply pressure to the district heating network is 2 bar, while the mean return pressure is 0,8 bar.

The expansion system of the network is connected to the supply manifold and comprises two closed expansion vessels of membrane type and a safety valve, which exhibit the following characteristics:

- Useful volume of each expansion vessel: 1000 l
- Pre-charge pressure of expansion vessel: 3,5 bar
- Maximum operating pressure of expansion vessel: 10 bar
- Safety valve calibration pressure: 6,5 bar

This system is designed to maintain a constant supply pressure within the network.

The network's water flow is assured by a pump unit regulated by an inverter, which draws from the return manifold and adjusts its speed to maintain the pressure differential between the supply and return manifolds.

Figure 2.5 – Pumping system

To guarantee the optimal flexibility and efficiency of the system, modular units comprising several inverter-controlled pumps were installed in the specific locations. This configuration ensures that the potential failure of a single pump does not affect the overall functionality of the system. Indeed, under typical operational conditions, only one pump is active within each individual group. In the event of pump failure, the remaining units are automatically activated.

2.2.2 Cogeneration system

The plant is equipped with two cogenerators, and the installation of two additional units is a viable option in the event of an increase in demand.

The general configuration of the cogenerator is as follows:

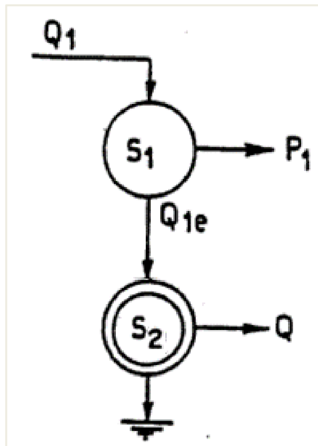

Figure 2.6 – General scheme of the flow of energy for a cogeneration plant [6]

This system consists in the simultaneous conversion of primary energy into mechanical and/or electrical energy and useful thermal energy within a single thermodynamic process. Figure 2.6 illustrates the configuration of the system, wherein section S_1 represents the engine that produces electric energy P_1, while section S_2 schematised the bottoming section with the production of thermal energy Q. It is evident that the waste heat from the above section (Q_{1e}) is transformed into useful thermal energy.

The generation of electricity is achieved using a generator that is splined to the drive shaft. This generator is then connected in parallel to the medium-voltage distribution network.

On the other hand, the thermal energy is recovered from the cooling water of the motor itself.

The main characteristics of the motor are presented in the following table:

POWER AND EMISSION DATA	
Engine	MTU/MDE 3042 LH3
Fuel	Natural gas
LHV	$10\ \frac{kWh}{Nm^3}$
Cycle	Otto
Hot water temperature	100/80 °C
Electric power	323 kW
Thermal power	485 kW
Electric efficiency	35,6%
Thermal efficiency	53%
Overall efficiency	88,6%
Inlet total power	907 kW
NO_2 at the discharge	$< 500\ \frac{mg}{Nm^3}$
CO at the discharge	$< 300\ \frac{mg}{Nm^3}$

Table 2.4 – Power and emission data of cogenerators

In general, the cogeneration system can operate in two principal modes:

- *Electric priority*: the unit follows an electric power value set by the operator. The resulting thermal power produced is dissipated by the heat sink on the roof unless it is not used entirely by the users. It is not typically feasible to set the setpoint at a value that is less than 50% of the rated power of the machine

- *Thermal priority*: the unit follows a set thermal power value (or tracks the heat load of the consumers). Consequently, the system operates at an electric power value at which there is no excess of thermal power to dissipate

In the case of TELEZIP, both cogenerators operate in thermal priority mode. The minimum thermal tracking has a minimum of 60% of the nominal thermal output. If the user requirement is low, only one cogenerator operates, with the two boilers activated in cascade to cover the peak demand.

The heat recovery system allows to obtain hot water by exploiting the heat produced by the engine and conveyed by the water cooling circuit inside the engine itself.

During the winter operational period, the water is heated within the engine water circuit and subsequently transferred to a heat exchanger, where it is used to heat the fluid within the district heating distribution network.

In the summer operating cycle, the hot water discharged from the engine is employed to feed the absorber, thereby producing cold water for distribution within the network.

In both cases, the water must be returned to the motors at a temperature of 80°C. In the event of a temperature exceeding 80°C, it is imperative to direct the water through the heat sink in order to restore the preset temperature.

In order to evaluate the performance of a cogenerator, it is necessary to consider a number of key parameters, which are outlined below:

i. *Fuel utilization index*

$$\eta_l = \frac{P_e + Q_u}{P_f} = \frac{323+485}{907} = 0.89 \quad (2)$$

where:
- P_e is the electric power produced
- Q_u is the thermal power produced
- P_f is the power of the fuel

This indicates that the cogeneration system exploits 89% of the available fuel power at design conditions.

ii. *Energy saving index (IRE) also called Primary Energy Saving (PES)*

$$PES = \frac{\left(\frac{P_e}{\eta_e^*} + \frac{Q_u}{\eta_t^*}\right) - P_f}{\frac{P_e}{\eta_e^*} + \frac{Q_u}{\eta_t^*}} = 1 - \frac{P_f}{\frac{P_e}{\eta_e^*} + \frac{Q_u}{\eta_t^*}} = 1 - \frac{1}{\frac{\eta_e}{\eta_e^*} + \frac{\eta_t}{\eta_t^*}} = 1 - \frac{1}{\frac{0.356}{0.374} + \frac{0.53}{0.81}} = 0.38 \quad (3)$$

where: - η_e^* is the electrical efficiency of the separate electricity production

- η_t^* is the thermal efficiency of the separate thermal production
- η_e is the electrical efficiency of the combined plant
- η_t is the thermal efficiency of the combined plant

The production of the separate plant is contingent upon the country in question being taken as a reference point.

The index was calculated in the design phase of the plant, and it illustrates that the production of the good entailed a 38% reduction in energy consumption when it was manufactured together, that is to say, with cogeneration, in comparison to the separate production of the good.

A plant is deemed to be a high-efficiency cogenerative plant if it meets the following criteria:

- PES > 0 if $P_{el} < 1$ MW
- PES > 10% if $P_{el} > 1$ MW

In order to be eligible for white certificates, it is necessary to fulfil this requirement. These are negotiable securities that certify the achievement of savings in the final uses of energy. One certificate is equivalent to the saving of one ton of oil equivalent (TOE). Moreover, the designation of a cogeneration plant entails the following benefits:

- Dispatching priority
- Release of guarantee origin (GO)
- Reduction of the tax burden on natural gas consumption

iii. *Electrical index*

$$I_e = \frac{P_e}{P_e+Q_u} = \frac{323}{323+485} = 0.4 = 40\% \tag{4}$$

This index indicates the proportion of electric energy produced in comparison to thermal energy.

If $I_e = 0$ only heat is produced, if $I_e = 1$ only electricity is produced.

2.2.3 Boilers

There are two boilers installed with the possibility to add the third one.

The boilers are condensing heat generators equipped with a natural gas modulating burner. The degree of oversizing is such that the failure of one boiler does not preclude the possibility of supplying sufficient heat to the users.

The heat generator is of the horizontal semi-fixed type with three effective smoke passes, passing flame and wet bottom and low NO_X emissions. Furthermore, the generator is furnished with a condensation coil and a condensate water drainage system.

The following table presents the principal characteristics of the boiler:

BOILER	
Model	ICI TNOX 2500 cond
Power	2500 kW
Max operating temperature	110°C
Electric power	7,3 kVA
Efficiency (referred to PCI)	107%

Table 2.5 – Main characteristics of the single boiler

In 2014, the boiler burner was replaced with a smaller burner with the objective of improving power modulation.

If the temperature in the grid return manifold declines to a level below the minimum threshold (i.e. when the heat output produced by the cogenerators is insufficient to meet the demands of the grid), a control loop will be initiated. This will entail the modulation of the opening of the valve of the first boiler, with the objective of achieving a temperature value in the grid return manifold that is equal to the set-point value. Once the load percentage of the burner of the first boiler reaches a predetermined threshold, typically between 85% and 90%, the control valve of the second boiler initiates a regulation loop analogous to the aforementioned process.

2.2.4 Absorber

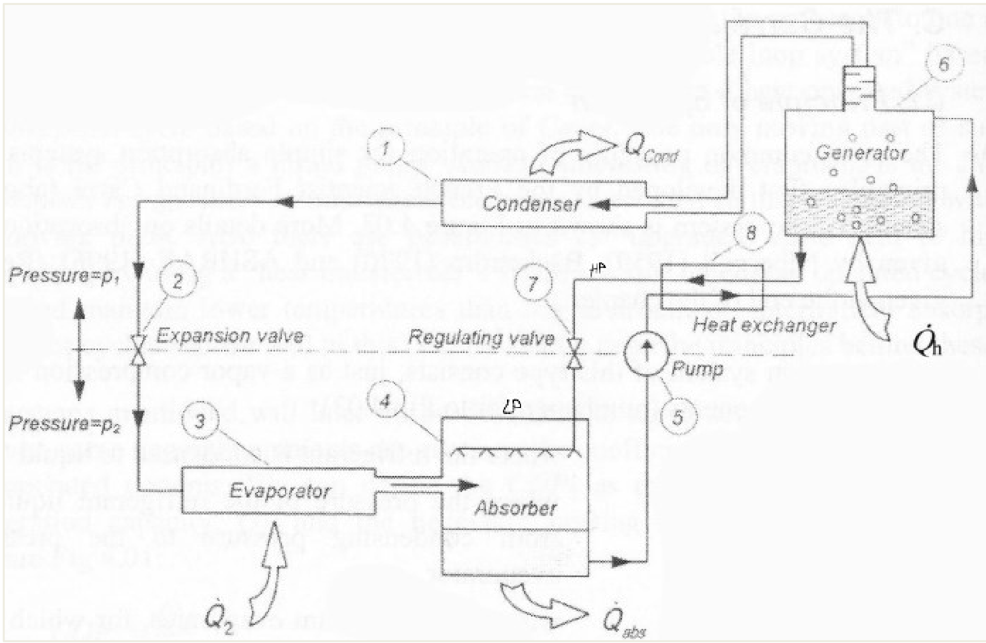

Figure 2.7 – General scheme of an absorption chiller [7]

where: - Q_h is the heat provided by the cogenerator

- Q_2 is the useful effect

- Q_{cond} and Q_{abs} are the heat rejected in order to close the cycle

The absorber is of the Lithium Bromide type and exploits the heat generated by the cogenerators. The absorber, which is supplied with hot water at 98°C from the cogenerators, produces chilled water at 7°C, which is used to feed the district cooling network. In order to produce cold water and complete the absorber circuit, it is necessary to reject some heat to the external environment. The rejected heat is dissipated into the external environment via the evaporative tower. In this type of plant, the latter is typically employed as it enables the exploitation of the wet bulb temperature for the rejection of heat at low temperatures. This is achieved through the spraying of hot water, which originates from the condenser, above the heat exchange pack where it meets the fresh air. This process both vaporises and rejects the latent heat of vaporisation. The hot saturated air is then expelled from the top of the tower at a constant temperature by means of the modulation of the rotation speed of fans.

The machine is modulating and due to the substantial lack of moving parts, it has a high degree of reliability. If the requirements exceed the maximum power that can be provided by the absorber,

the electric chiller is activated to meet the overall demand. The same principle applies in the event of an absorber shutdown.
The operational logic of the ‘cold’ production system is based on the principle of priority for the absorber, which is continuously operational during the summer season.

ABSORBER CHARACTERISTICS	
Model	TSA-16LJ42
Refrigerating capacity	1319 kW
Nominal operating temperature	13°C / 7°C
Cooling water temperature	29°C / 34°C
Feed water temperature	98 °C / 79 °C
Electric capacity	7,3 kW

Table 2.6 – Main characteristics of absorber

The cold water production circuit comprises of two inertial storage tanks, one contains the water returned from the users at 13°C, and the other contains the cold water to be sent to the users at 7°C.
The cold water generators, comprising absorption and compression chillers, are connected in parallel to the aforementioned tanks/collectors.

In reference to figure 2.4 up to this point, the section of the plant situated at ground level has been outlined.
The remaining parts, which will be described in the following part, are located on the roof of the building.

2.2.5 Heat sink

The apparatus comprises a forced-ventilated finned coil, which serves to enhance the efficiency of the heat exchange process.

Figure 2.8 – Heat sink of Telezip plant

HEAT SINK	
Capacity	1500 + 10% kW
Number of fans	14
Finned coil	Copper / Aluminium
Internal circuit volume	282 l

Table 2.7 – Main characteristics of heat sink

In the event that the cogenerator is in operation and there is a considerable reduction in demand for thermal power, it may be necessary to dissipate the excess of heat. Indeed, it is imperative that the temperature of the water entering the cogenerators does not exceed 80°C to ensure optimal operation. If the water returning from the heat exchanger has not transferred an adequate quantity of heat to the fluid directed to the user, or if the water leaving the absorber generator is at a temperature slightly higher than 80°C, the water is directed to the heat sink without activating the motor fans. This is because the exchange area of the heat sink itself is sufficiently capable of reducing the water temperature to 80°C. In the event that the temperature of the water to be returned to the cogenerators is significantly higher than 80°C, then the heat sink fans will also be activated to enhance the dissipation with the surrounding environment.

2.2.6 Chiller

The chiller was replaced in 2022 and the characteristics are resumed below:

CHILLER	
Model	RHOSS – FPE2100056
Absorbed power	267 kW
Nominal refrigerating power	892 kW
Nominal air temperature	35°C
Nominal user temperature	7/12°C
Refrigerant type	R134A (GWP 1430)
Refrigerant temperature min/max	-22/+130 °C
Refrigerant charge circuit 1	55 kg
Refrigerant charge circuit 2	57 kg

Table 2.8 – Main characteristics of chiller

Any cooling energy demands that are not met by the absorber are fulfilled by the electric chiller, which consists of a compression refrigeration unit.

As mentioned in section 2.2.4 the absorber is always in operation during the summer period. In the event that the cooling capacity required by the consumers is not fully met by the operation of the absorption system, the compression chiller is then activated. The system draws water from the storage tank, which is maintained at a temperature of 13°C. The activation of the compression chiller is controlled by a sensor that acts at a trip temperature threshold, which is detected in the storage tank at 13°C. It is possible for the temperature to differ from that of the control temperature of the absorber, with the objective of achieving the overall setpoint temperature level between the absorber and the compression refrigerator. The water leaving the refrigeration unit is collected in the tank at 7°C, after which it is then conveyed through the pumps to the substations.

2.2.7 Current operating mode

The district heating and cooling plant is currently underutilised for several reasons, which are outlined below.

It is important to note that despite the existence of a contract between the owners of the buildings and the customer, not all customers within the buildings that are served by district heating actually utilise the service. The tenants have opted to install their own systems, which are currently not authorised. Another issue pertains to the alteration of rooms configurations, which has resulted in suboptimal customer comfort due to the erection of plasterboard walls or placement of items in front of the terminals, thereby impairing the operational efficacy of the system. Furthermore, the routine maintenance, which is the responsibility of the end customer is frequently not carried out, resulting in continued deficiencies in the equipment.

The aforementioned factors collectively resulted in a reduction in the estimated demand during the project feasibility study. This has significant implications for the plant, as low demand makes the use of cogenerators impractical from both an economic and a technical standpoint. The number of annual operating hours and, in particular, the load capacity at which they would operate would be insufficient to justify their use. Another factor that has contributed to the non-use of cogeneration is the cessation of the minimum guaranteed price on electricity by ARERA from 1st January 2023 for energy fed into the grid by cogeneration plants.

In light of the aforementioned considerations, it can be concluded that:

- In winter, boiler (usually only one) is used to cover the heat demand
- In summer, only the electric chiller is used at reduced load to cover the load demand. The absence of cogenerators in the summer months also precludes the utilisation of the absorption cycle, given the unavailability of heat

These factors have had a deleterious impact on TELEZIP, preventing the utilisation of cogenerators and the benefit from advantageous incentives (white certificates). Additionally, the efficiency of the plant has declined.

Despite these factors affecting the power plant’s performance, a reduction in atmospheric emissions persists due to the scaling factor.

2.3 Advantages and disadvantages of district heating and cooling systems

District heating is particularly well-suited to the heating of dense urban areas, offering several advantages to the citizen-user that contribute to the high adoption of the service in areas where it is available. Compared to conventional heating, the district heating provides significant benefits for both the citizen-user and the community.

The advantages for the user in general are multiple and can be enumerated as follows [8]:

- Absolute safety: the replacement of gas with hot water feeding the heat exchanger eliminates the risk of fire due to the absence of open flames
- All-inclusive tariffs: the companies providing the service assume responsibility for the ordinary and extraordinary maintenance of the central plant
- Continuity and reliability of the service: the reliability of a substation is considerably higher than that of a traditional boiler, which reduces the risk of interruption to the heating service or breakdown for users connected to the network. Substations are typically equipped with remote control and remote management systems, which facilitate the real-time detection of malfunctions and the prompt implementation of corrective measures. In the case of Telezip, this instrument will be implemented through this thesis
- Simplicity and cost-effectiveness: the plant necessary for a user of district heating is considerably more straightforward than that required for a gas boiler, consisting essentially only of a heat exchanger
- Lower cost for the heat than traditional fuel: the cost of the district heating service in Italy is linked to the cost of natural gas established by the GME. Nevertheless, it is notable that the price of district heating service is, in any case, lower than the price of gas
- Maximum usability of the service: the utilisation of district heating is feasible for all buildings. The rooms housing the substations are not subject to any specific requirements.
- Exemption from the use of "*renewable sources*": in the case of new buildings, D. Lgs. n.199 of 8 November 2021 established that at least 60% of the energy needs of the house must be met by renewable sources [9]
- Air pollution can be significantly reduced: the reduction in fuel consumption, resulting from the utilisation of energy recovery and the enhanced efficiency of large centralised plants in comparison to the multitude of smaller condominium plants, has led to a notable decline in the overall energy expenditure

- Reduction of pollutant substances released into the atmosphere: the fuel burned in the central plant may be selected on the basis of market cost considerations. Furthermore, waste heat from a multitude of industrial processes or incineration furnaces can be utilised for this purpose. In a well-designed district heating system, the chimney has a lower environmental impact than that of the chimneys of the individual houses in the city
- Possibility of implementing a rational policy in the use of energy sources

Despite the considerable advantages of district heating, this technology also presents certain disadvantages [10]:

x The high installation and maintenance costs associated with this system render it a viable option only in areas of high population density, where the payback period is not excessive.

x In the case of natural gas power plants, there is a potential for a concentration of pollutants, particularly nitrogen oxides in the vicinity of the plant

2.4 Environmental and social sustainability aspects of using a district heating and cooling network

The necessity for air conditioning in the world is increasing in line with global population growth, urbanisation and rising prosperity. The IPCC (Intergovernmental Panel on Climate Change) anticipates a rise in cooling demand from 300TWh in 2000 to 4000TWh in 2050, which will have an impact on CO_2 emissions and climate change [11]. Moreover, the climate change observed in recent years has resulted in longer periods with more severe temperatures throughout the year. This therefore requires the implementation of robust and sustainable solutions to ensure the maintenance of an adequate indoor thermal environment, particularly during the summer months. The utilisation of the district heating network for cooling purposes during the summer months represents a further advantage of the system, as it allows for the comprehensive coverage of the building's annual thermal requirements without the necessity for the installation of two distinct types of terminals for the two thermal seasons.

It is recommended that the use of district heating networks be encouraged wherever feasible for several reasons. It is of great importance to disseminate information regarding the benefits and advantages that the use of district heating and cooling to the community.

It is beyond doubt that the implementation of a district heating solution will result in a reduction of atmospheric emissions associated with the use of domestic air conditioning. At the Italian level, the PNIEC (Piano Nazionale Integrato per Energia e Clima) has assigned an important role to district heating, with the objective of extending the network.

It is also important to consider the enhanced energy efficiency that can be achieved through alternative production methods, which often surpass the efficiency of individual domestic systems. One must consider of the absorption cycle, which can produce cold through the utilisation of the “free” heat released by the cogenerator. Furthermore, during the summer season, this source is classified as green and sustainable, as the cycle utilises pure water as a refrigerant, thereby resulting in a Global Warming Potential (GWP) of zero.

The role of district heating in the energy transition is of significant importance. Its compatibility with renewable sources renders it an advantageous solution for both the user and the producer. As the level of demand from users increases, the losses between the central plant and the users decrease. This is a tangible reality, as evidenced by the current system that has been in use for over 15 years.

Moreover, the deployment of district heating services facilitates compliance with legal obligations concerning the minimum percentage of renewable energy sources to be utilised in domestic applications. In particular, new buildings or buildings undergoing substantial renovation must comply with the requirement that 60% of their energy consumption for air conditioning and hot water production must be covered by renewable energy sources.

From the perspective of the producer, a crucial element influencing the operation of district heating is the capacity to measure individual consumption on an hourly basis. This allows for the monitoring of consumption trends and peak demand, which in turn enables the optimisation of production planning. The utilisation of such data can enhance the efficiency of production planning.

Furthermore, the government has introduced economic incentives to encourage the adoption of district heating, thereby facilitating a more profitable investment from an economic standpoint. These incentives are designed to encourage the connection of utilities to the network.

The price of the service may be determined in three ways throughout its utilisation:

- Avoided cost
- Gas-dependent cost
- Operating costs

The pricing structure of the district heating system under examination in this thesis is based on the cost of gas. However, in contrast to the electricity and natural gas sectors, there is no unified market due to the lack of physical interconnection between networks. The costs associated with each network are contingent upon the energy source used for thermal energy production and the overall efficiency of the system [12].
The centralisation of heat production has the additional benefits of enhancing energy security and reducing the risk of domestic accidents caused by boiler failure. Furthermore, the district heating and cooling network design and operation facilitate the creation of local jobs by fostering the development of technical skills in the energy sector and promoting the green economy.

2.5 Regulations and standards overview

It is imperative that legal considerations concerning the district heating service do not deviate from the normative definition set forth in Legislative Decree n.28 of 3 March 2011 (D.Lgs. 28/2011). In particular, according to article 2, paragraph 1, letter g, district heating or cooling is defined as follows:

> *"The distribution of thermal energy in the form of steam, hot water or cooling liquids from one or more production sources to a plurality of buildings or sites through a network is a process whereby heat is transferred from the source to the buildings or sites in order to heat or cool spaces, facilitate processing and supply domestic hot water".*

In accordance with legislative mandate, all district heating networks are required to undergo inscription in the register of the *"Autorità di Regolazione per Energia Reti e Ambiente" (ARERA)*
The objective of implementing thermal energy transport networks on public land is to facilitate the connection of interested parties to the network for the supply of thermal energy for space heating or cooling, for processing and for covering domestic hot water needs, in accordance with the extent of the network and the relevant regulations.
The concept of "*efficient district heating or cooling system*" was first defined by EU Directive 2012/27/EC, which was subsequently transposed into Italian law by Legislative Decree 102/14. This is a system that uses, alternatively, at least:

- 50% energy from renewable sources
- 50% waste heat
- 75% of cogenerated heat
- 50% of a combination of the preceding

In the European context the efficient district heating has gained further relevance in the "*Clean Energy for all Europeans Package*", which defines European energy and climate policies until 2030. The package stipulates, for instance, that:

- Efficient district heating can be used to verify the attainment of minimum RES share obligations in buildings (EU Directive 2018/2001 – so called RED II, Art.15, par. 4)
- The right of disconnection of inefficient district heating users is guaranteed (EU Directive 2018/2001 – so called RED II, Art.24, paragraph 2)
- In the compulsory annual energy savings imposed by the EU, energy savings generated by efficient district heating and cooling can also be taken into account for metering (EU DIRECTIVE 2018/2002, modifying Directive 2012/27/EU – called EED recast, Art.7, paragraph 4)

With regard to Italian legislation, the "*Piano Nazionale di Ripresa e Resilienza*" (PNRR) has earmarked an allocation of 200 million euros for the construction, transformation and extension of efficient district heating and cooling networks within Mission 2, Component 3, Investment 3.1. The conservation of incentives subsequent to the completion of these interventions, financed by the PNRR, is contingent upon the fulfilment of efficient district heating conditions for a minimum of one year within the two-year period following the conclusion of construction activities.

2.6 Structure of the TELEZIP district heating distribution network

The hot and chilled water produced in the central plant is distributed to the customers via an insulated underground pipe network. The exchange of heat between the heating network and the air conditioning systems of the individual building units occurs in dedicated substations.

In this context, the term "distribution line" refers to the piping system that facilitates the flow of heat transfer fluid, connecting the central heating system to the substations. The substation is comprised of a plate heat exchanger. In substations, heat exchange occurs between the heat transfer fluid from the thermal power source (primary side) and that of the customer's air conditioning system (secondary side). It is thus imperative that the pipes be thermally insulated to minimise the exchange of heat between the heat carrier fluid and the external environment as much as possible.

The distribution substations are located within the building units in technical rooms or in dedicated external containers. The compartmentalisation of the system renders the occurrence of breakages or faults due to accidental impacts or actions by unqualified personnel highly improbable.
The distribution line is fully insulated and buried between 1,7 m and 0,5 m deep, with a length of approximately 3,5 km. The pipes are made of steel for the main sections and plastic for the final branches. Given the considerable length of the distribution network, the loss of thermal energy is non-negligible.

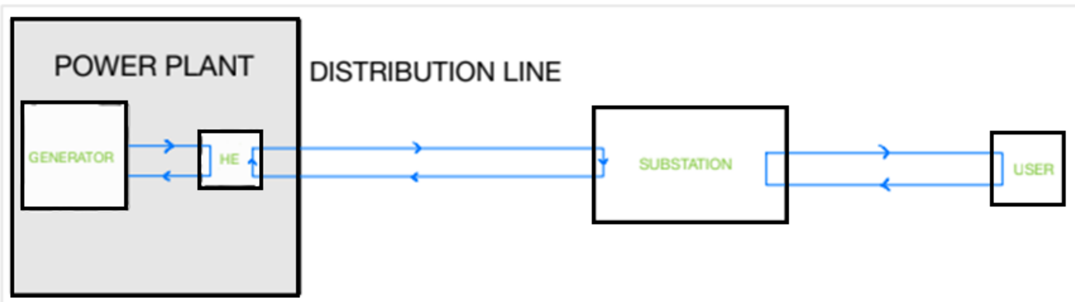

Figure 2.9 – Scheme of the plant from the power plant to the users

The substations are multiple and have different capacities depending on the type of utility. The fluid temperatures in both the primary and secondary circuits are almost identical for each substation, although the size of the heat exchanger differs. Indeed, a different flow rate of heat transfer fluid will be circulated depending on the user.

The characteristics common to all substations are shown in the following table.

SUBSTATION DATA	
Max and min temperature (primary side)	110°C / 5°C
Max pressure (primary side)	16 bar
Max pressure (secondary side)	10 bar
Winter inlet/outlet temperature (primary side)	80°C / 55°C
Summer inlet/outlet temperature (primary side)	7°C / 13°C
Winter inlet/outlet temperature (secondary side)	50°C / 65°C
Summer inlet/outlet temperature (secondary side)	15°C / 9°C

Table 2.9 – Common data for the substations

Figure 2.10 – Scheme of typical substation of TELEZIP

It is evident that the inlet temperature from the primary side of the heat exchanger cannot exceed 110°C, as this is the limit permitted for substations. However, for the pipe network, the maximum permissible temperature is 85°C.

2.7 Current metering system configuration

The meters that have been installed for the purpose of recording thermal consumption, regardless of whether the customer is currently utilising the district heating service or not, are as follows:

- N.16 Bmeters – Hydrosplit M3
- N.11 Coster - IET 7383
- N.13 Siemens – UH50-C52CIT06-E
- N.1 Sontex DE-07-MI004-PTB012

At the present time, the meters are read one by one monthly by an operator whose role is to visit the site and read and record the data in a dedicated register.
Since the Coster, Siemens and Sontex meters were installed over 7 years ago, it is necessary that they be replaced.

Conversely, data pertaining to the power centre, including electricity and gas consumption, thermal energy production, and other relevant parameters, are collected by a remotely manageable PLC that can be managed remotely through an hourly reading.

3. COMMUNICATION NETWORKS FOR REMOTE METER READING

This section will set out the requirements for the implementation of remote meter reading, as well as the requisite devices that are necessary to accomplish this task. It will also present the entire communication infrastructure, including an examination of the communication protocols and standards employed.

3.1 Requirements for the implementation of remote meter reading

The implementation of a remote metering system requires the establishment of an infrastructure capable of managing the software that enables the signal to be conveyed from the meter to the control centre and subsequently transmitted to the pertinent database. In order to achieve this, it is necessary to install a repeater and signal receiver in addition to the software.
The selection of devices to be utilised is designated to the “Bmeters”, as a number of these have recently been installed and are compatible with remote reading.

To utilise the “Bmeters” software, the following system requirements must be met:

- Windows 7/8/10/11
- CPU: dual core x86/x64 bit 1,5 GHz
- RAM 4 GB
- HDD: 150 MB
- 1 GB for database and readings
- 2 USB ports

For the repeater to be configured correctly, the following requirements must be met:

- From windows XP SP3 to Windows 10
- CPU: x86\x64 bit – 1,5 GHz
- RAM: 1 GB
- HDD: 3 MB
- Microsoft .NET Framework 4.0

The receiver’s technical specifications are as follows:

- From windows XP SP3 to Windows 10
- CPU: x86\x64 – 1 GHz
- HDD: 10 MB
- RAM: 512 MB

The computer characteristics present in the central plant that manages the PLC are in accordance with the requirements for remote reading devices.

3.2 Communication protocols and standards used

The communication protocol employed is that of the *LoRaWan system*. LoRaWAN is a Low Power Wide Area Network (LPWAN) radio technology that enables long-range transmission for objects with modest power consumption. LoRaWAN operates within the free sub-Gigahertz band, which exhibits favourable propagation characteristics, thereby enabling the coverage of longer distances than those achievable at higher frequencies, such as those used by 2.4 GHz of Wi-Fi or Bluetooth. The network structure needs the presence of a minimum of one LoRaWAN gateway, capable of spanning a wide geographical area, to oversee the management of a considerable number of end points. This involves the collection of data from these end points and its subsequent upload to the cloud.

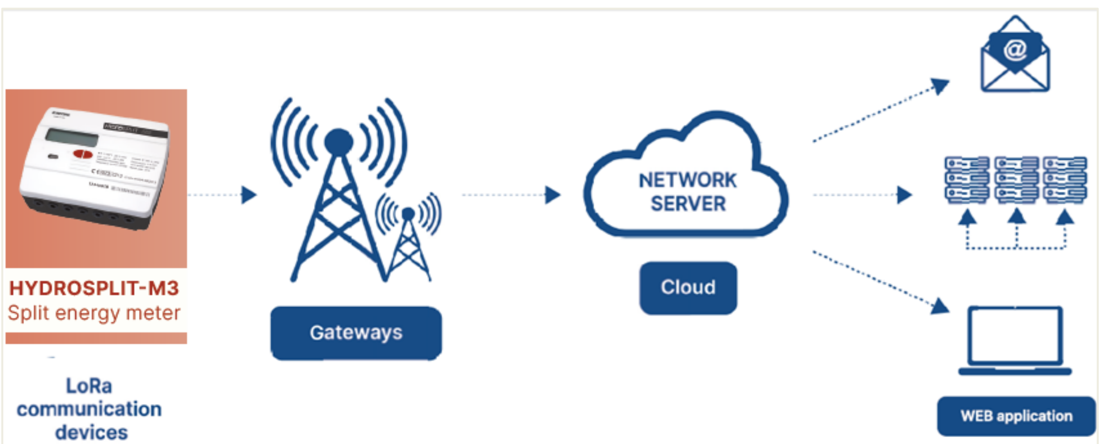

Figure 3.1 – example of LoRaWan system

The standard employed for meter reading is the M-BUS standard.
Meter-Bus or M-Bus is a European Standard (EN 13757) dedicated to the remote reading of water, gas and electricity meters. The system developed with the specific objective of creating a system for remote meter reading. In fact, it was designed to meet the specific requirements of battery-powered or remote systems. Upon interrogation, the meters are required to provide data to a central "Master", such as a laptop, which requests the data at periodic intervals.
The data transfer speeds are relatively low, allowing for long-distance communication. The typical speeds range is between 300 bps and 2400 bps.

The architectural scheme employed is the Master-Slave configuration. M-Bus belongs to the fieldbus family, consisting of information transmission systems based on a common transmission medium to which devices capable of receiving and transmitting information are connected.

As M-Bus is an open system, with the technical specifications and communication protocol freely available, it is possible to connect devices from different manufacturers in order to exchange information. The communication is overseen by a single device (Master) which periodically queries other devices (Slaves) to collect information. A Slave is not able to assume the role of initiator in the communication process; rather, it must wait for the query issued by the Master. In addition to its function as a data concentrator, the Master generally offers additional functions such as data storage (data logger) and remote control via a communication interface. [13]

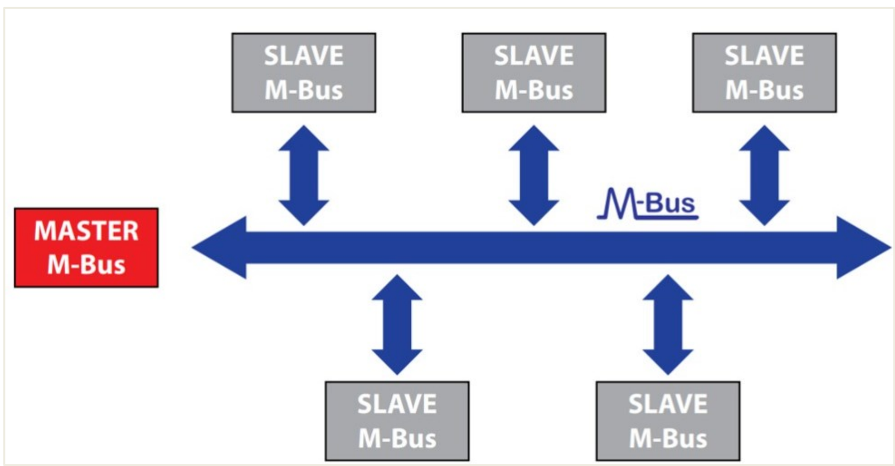

Figure 3.2 – Master/Slave architecture

The M-BUS system was developed with the specific purpose of centralising consumption data and meter operating parameters. It offers a number of important technical advantages:

- High security in data transmission
- Low cabling costs
- It supports long distances
- High number of centralise devices
- Automatic recognition of connected devices

In addition to the technical standards, remote meter reading is also a matter of European legislation, as set forth in the EU Directive 2018/2002 [14].
Article 9c of this Directive states the obligation of remote reading, as follows:

"1. For the purposes of Articles 9a and 9b, meters and heat cost allocators installed subsequent to 25th October 2020 shall be deemed to be remotely readable devices. The conditions of technical feasibility and cost-effectiveness set out in Article 9b(1) shall continue to apply.

2. Meters and heat cost allocators which are not remotely readable but that have already been installed shall be rendered remotely readable or replaced with remotely readable devices by 1 January 2027, unless the Member State in question demonstrates that this is not cost-efficient."

3.3 Bmeters devices: features and functions

3.3.1 Hydrosplit-M3 - Bmeters

This thermal energy meter is designated for use with external flowmeters that are equipped with pulse output. It can be installed on sites where the pipe size is larger than ¾" or where access is limited due to presence of obstacles. In such cases, the compact heat meter is not a viable option. Main characteristics:

- Hot/cold combined
- 2 pulse inputs + 2 pulse outputs integrated
- M-BUS EN13757-2/3 output integrated
- Wireless M-BUS EN13757-4 output version.

Parameter	*Values*
Temperature measuring range (heating) [°C]	5 ÷ 180
Temperature difference range (heating) [K]	3 ÷ 150
Temperature measuring range (cooling) [°C]	2 ÷ 24
Calibration temperature (cooling) [K]	3÷ 20 K
Max measurable flow rate [m^3/h]	2.000
Input pulse rate [L]	0,1-0,25-1,0-2,5-10-25-100-250
Battery life [y]	10

Table 3.1 – main characteristics of thermal energy meter

This meter must therefore be furnished with cables for the purpose of connecting the measuring probes. Subsequently, the device must then be connected to a flow meter and synchronised to the same number of pulses. An external controller then responds to the thermal energy demand from the secondary side.

3.3.2 Electronic pulse output device

The meter described in section 3.3.1 is equipped with a pulse emitter device.
The mechanical core of the meter is the multi-jet turbine, the number of revolutions of which is directly proportional to the circulating flow rate. The pulse emitter is furnished with a connecting cable, which enables the transmission of the turbine's number of revolutions. The pulse launcher, through a rotating magnet, converts the mechanical movement into an electromagnetic contact (reed contact), which emits electrical impulses with a frequency proportional to the number of turbine revolutions and thus to the circulating water flow rate. The dedicated electronics then acquire these signals together with those originating from two temperature probes located on the system's supply and return pipes. Subsequently, the data is processed, and the thermal energy consumed is calculated.

3.3.3 Repeater

The wireless signal repeater M-Bus-RFM RPT3, is a device that enables the replication and expansion of the radio signal transmitted by the radio modules installed on heat meters in accordance with the WMBUS standard. The device is programmable with a daily activities window, within which it performs the repetition of WMBUS telegrams in accordance with the EN13757 standard. Subsequently, the device repeats the signals transmitted by the measuring devices excluding those that have already been repeated and sent from other repeating devices. The same device can be configured to enable the establishment of up to 3 levels repeating chains. In conditions of optimal signal propagation, the range of each individual device is 300 meters.

3.3.4 Receiver

The RFM-RX2 wireless M-BUS receiver is a USB device that is used for the configuration and collection of the reading data that is transmitted by the radio modules that have been installed on heat meters in accordance with the WMBUS standard. The receiver is furnished with a SMA connector, facilitating the expedient removal and replacement of the 360° orientable antenna. This device can cover distances up to 300 meters.

3.3.5 Concentrator

The RFM-C3 is a gateway/data concentrator that collects data sent from Wireless M-Bus devices and transmits them via GPRS signal or via an Ethernet/LAN/Wi-Fi network. In the case of TELEZIP a mobile phone SIM card is used. The data and information pertaining to the consumption of energy, as transmitted by the Bmeters devices equipped with a wireless M-Bus, are collected by the RFM-C3 concentrator and in general subsequently transmitted via e-mail and/or FTP by utilising the GPRS network or, alternatively, an internet connection via an Ethernet/LAN/Wi-Fi network. The e-mail contains the file of the acquired telegrams in either ".txt" and/or ".csv" format, with the option of hourly, daily, weekly or monthly frequency. The configuration of the device is performed via a PC connected to the concentrator via a LAN cable, utilising the ethernet port.

3.4 Technology infrastructure for remote reading

The remote reading systems facilitate the collection of the data recorded by the meter, thereby eliminating the necessity for manual operation.
The aforementioned features collectively facilitate the following [15]:

- Time saving in reading procedure
- Attempted fraud detection
- Water leak detection
- Error free data reading
- More privacy for the users
- Data are directly stored into a PC thus eliminating any transcription operation.

The floor plan of the buildings that TELEZIP reaches is presented in Appendix A.
The total extension of the area to be served is approximately 12 hectares, which equates to approximately 120.000 square meters.

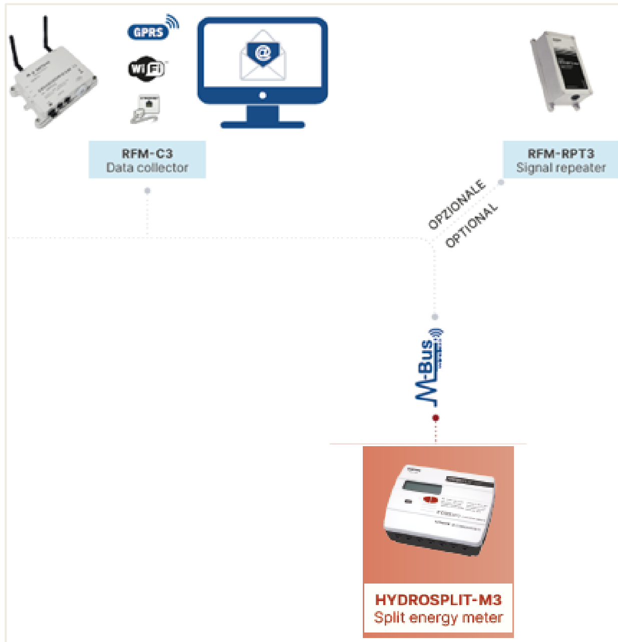

Figure 3.3 – scheme of the infrastructure from the meter to the concentrator [15]

Given that the power centre is not in a barycentric position, it is imperative that the infrastructure be arranged with the highest degree of precision and care.

It is of the greatest importance that the repeaters be situated in the most optimal locations, to ensure the minimum number of installations while simultaneously preventing any potential communication issues. In consideration of the data sheet of the repeater, which indicates a coverage range of 300 meters, the selection of the location will incorporate a safety margin of 100 meters, given that not all meters are within the line of sight of the receiver. It is therefore proposed that a maximum distance of approximately 200 meters from the meter to the repeater be considered.

The location of the repeaters is determined with the objective of providing a connection for all the units that are eligible for service via district heating network. In fact, as stated in chapter 2.2.7, it should be noted that at present not all the units are connected to the district heating network.

The following section of the thesis will analyse two time horizons:

- In the short term, the replacement of meters for users who are already served and who have an old meter is the proposal under consideration
- In considering the long-term scenario, it is necessary to take into account prospective projections and to establish a hypothetical connection with all units that may potentially be served.

In order to satisfy the aforementioned requirements, the network was configured in the following manner based on the results of a series of measurements:

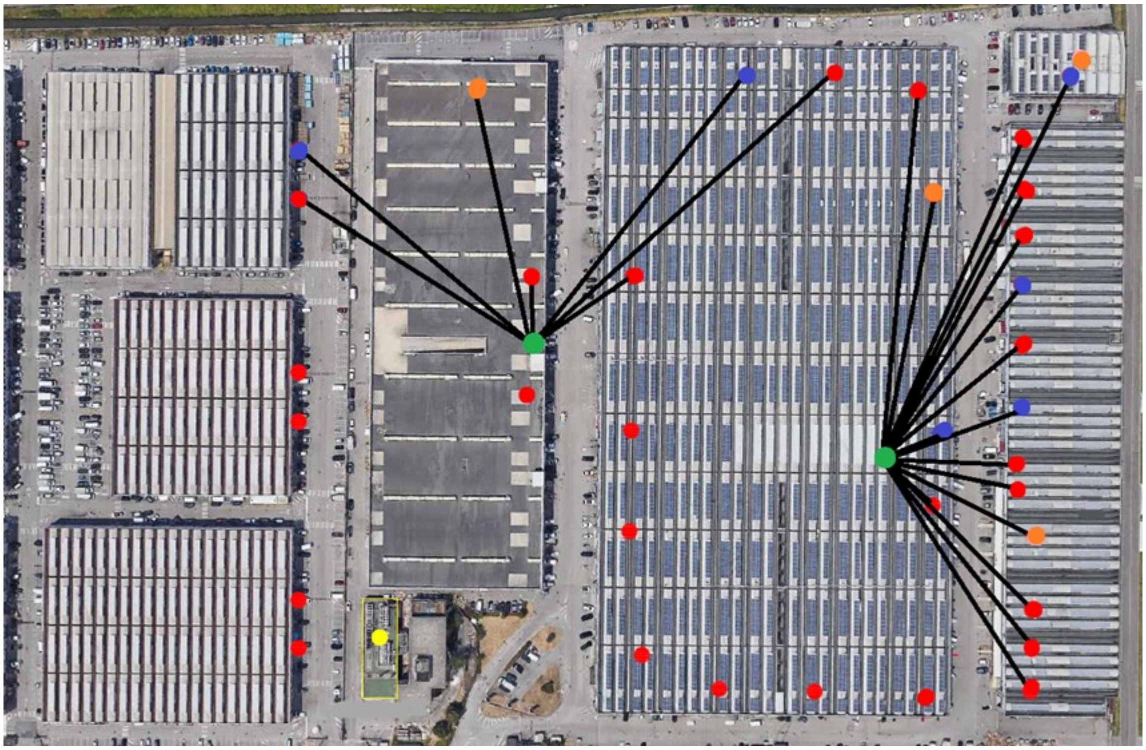

Figure 3.4 – Arrangement of Bmeters devices

Where:

- The red dots indicate the position of the meters that are not currently provided but may be operational in the future
- The yellow dot represents the concentrator, which is placed in the central unit
- The green dots represent the repeaters. They are in direct communication with the concentrator, which is situated in the power centre
- The orange dots indicate the location of meters that have already been replaced
- The blue dots represent the location of the meters to substitute
- The black line represents the link between the meters and the repeater.

For the sake of simplicity, the representation does not include the lines connecting the meters directly to the power centre and the lines connecting the repeaters to the receiver. Moreover, in some units, the sub-counters have been omitted from the representation in order to prevent unnecessarily complex visualisation. As illustrated in figure 3.4, only the sub-counters in the upper right section are displayed, as they have already been replaced (orange dots), whereas the corresponding general meter still requires replacement (blue dots).

3.5 Integration of Bmeters devices into the system

The short-term intervention involves the replacement of users whose apparatus has reached the end of its operational lifespan:

- N.6 Hydrosplit M3 with probes
- N.6 flow meters
- N.2 repeaters
- N.1 concentrator
- N.1 receiver
- N.1 software license.

The long-term intervention concerns the replacement of all potential customers, with the exception of those who have already been replaced in the short-term scenario:

- N. 28 Hydrosplit M3 with probes
- N. 28 flow meters

The software interface provides comprehensive information for parameter configuration, thereby facilitating the generation of valuable insights into user and network behaviour.
Once the configuration of all the devices had been completed in accordance with the instructions provided by the manufacturer, the data could be accessed remotely.

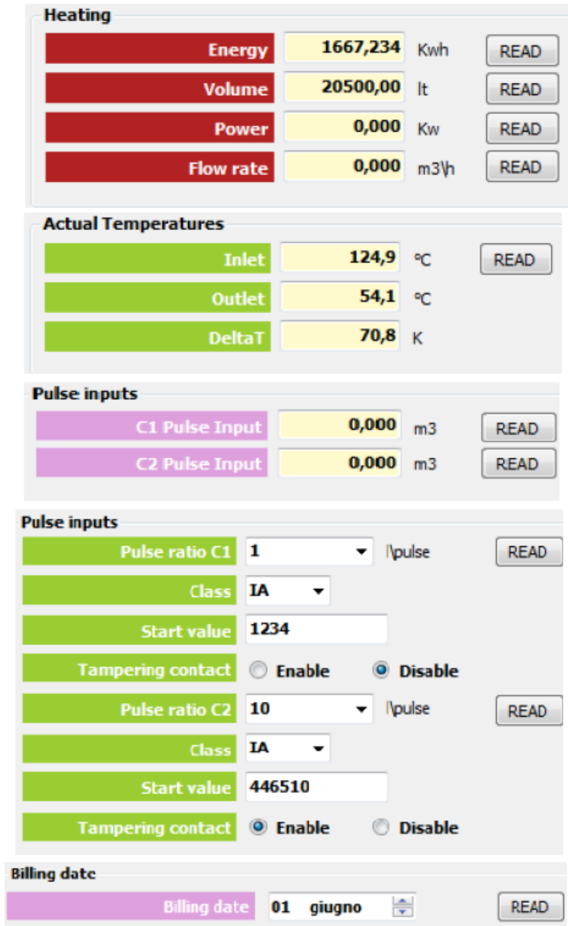

Figure 3.5 – Main output data readable

3.6 Cost-benefit analysis of the implementation of remote meter reading

The implementation of remote meter reading offers many significant advantages. At the present time, the meter reading is conducted at the end of each month by an operator responsible for reading and reporting the consumption of the meter users. The possibility of human error in the reading or transcription of values is considerable. Furthermore, the company was required to pay a fixed monthly fee for the technician's services. The cost of manually reading customers' meters is approximately €150 per month, equating to an annual cost of €1.800.

Two distinct economic analyses will be conducted: the first focusing on the short term and the second on the long term.

In the context of a short-term analysis, the cost estimate is presented in comprehensive detail, given that the replacement is imminent due to the fact that customers are actively utilising the district heating service.

Unit	*Diameter*	*Hydrosplit M3 + probes*	*Pulse launcher*	*Installation*	*Overall costs*
C27	DN100	€ 261,00	€ 498,30	€ 650,00	€ 1.409,30
A34	DN100	€ 261,00	€ 498,30	€ 650,00	€ 1.409,30
C17	DN40	€ 261,00	€ 307,40	€ 400,00	€ 968,40
C15	DN40	€ 261,00	€ 307,40	€ 400,00	€ 968,40
O69	DN65	€ 261,00	€ 365,50	€ 500,00	€ 1.126,50
A38	DN100	€ 261,00	€ 498,30	€ 650,00	€ 1.409,30
Total					€ 7.291,20

Table 3.2 – costs for the implementation of the new measurement system

The term "installation" encompasses the costs associated with the utilisation of consumables, fittings, flanges, and working hours.

The financial outlay required for the implementation of a remote reading infrastructure can be itemised as follows:

Item	*Price*	*Quantity*	*Cost*
Replacement of old meters	-	6	€ 7.291,20
Receiver RFM-RX2	€ 182	1	€ 182,00
Repeater RFM-RPT3	€ 210	2	€ 420,00
Concentrator	€ 420	1	€ 420,00
Software license	€ 210	1	€ 210,00
Total			*€ 8.523,20*

Table 3.3 – initial costs for the infrastructure

The implementation of a new remote reading system has resulted in a notable decrease in the occurrence of human error. The introduction of an automated reading step has enabled for more precise measurement of consumption. This, in turn, results in a more accurate representation of the demand trend and peaks, which allows for more effective scheduling and more accurate billing.

In consideration of the long-term analysis, an estimate was derived by taking the average cost for potential future connections. It is not possible to make an accurate estimate in this case, due to the inherent variability of material prices and the variable unit occupancy. This estimate is limited to substations, as the possibility of further subdivision cannot be foreseen at this time. Furthermore, a considerable number of customers are not under a supply contract, rendering the replacement of meters on those units an inefficient use of resources.

Diameter	*Hydrosplit M3 + probes*	*Pulse launcher*	*Installation*	*Quantity*	*Overall costs*
DN40	261.00 €	307.40 €	400.00 €	9	€ 8.715,60
DN50	261.00 €	337.40 €	450.00 €	4	€ 4.193,60
DN65	261.00 €	365.50 €	500.00 €	7	€ 7.885,50
DN100	261.00 €	498.30 €	650.00 €	8	€ 11.274,40
Total					€ 32.069,10

Table 3.4 – overall costs for the infrastructure in long term

The cost-benefit analysis shows that remote reading offers advantages in the medium to long term due to following factors:

- Reduction of operational costs
- Improving the data management
- Better monitoring
- Optimization of consumption
- More privacy for the users
- Attempted fraud detection
- Water leak detection

Nevertheless, the full extent of the benefits that this measurement system can provide can only be evaluated after one thermal year, as a comparison with the previous thermal year can be made.

4. TECHNICAL-ECONOMIC SOLUTIONS FOR END USERS

This chapter presents an analysis of the unconnected users of building C (see figure A.2 on the right-hand side) with the objective of calculating the winter and summer thermal energy demand. Subsequently, a technical-economic evaluation is conducted to analyse the potential for connecting terminals to the district heating and cooling network, with the objective of defining the criteria for selecting the most suitable solution.

4.1 Analysis of non-connected users: types and characteristics

The users served by the district heating and cooling network are predominantly wholesale stores, the size of which can vary considerably: from 470 to 5.000 square meters.

The unserved buildings in hall C are almost all users divided into three zones:

- Offices
- Warehouse
- Shop (display area)

Figure 4.1 - Arrangement of the non-connected users in hall C. All the quotas in meters. Orange arrow indicates North direction

The quotas represented in the figure 4.1 correspond to the internal dimensions.

The floor plan is divided into the following sections:

- 152 m^2 for the display area
- 133 m^2 for the offices
- 300 m^2 for the warehouse

This equates to a total area of 585 m^2.

The structure is 6 meters in height.

The plant is located within the "Centro Ingrosso Cina" district in Padova. It can be observed that the majority of users are of Chinese nationality. In traditional Chinese culture, the practice of leaving the doors of a store open is not merely a practical indication that the store is open; it may also be perceived as a gesture associated with the ancient Chinese philosophical and spatial discipline of "feng shui"[1]. This results in a greater loss of heat to the external environment than would occur in a store with closed doors.

In this paper, the plants analysed will be based on this type of user, and thus the aforementioned factors should be properly taken into account during the evaluation of energy needs.
The energy performance of the building will be evaluated in accordance with the standard EN ISO 13790:

> *"Method: quasi-steady-state methods, calculating the heat balance over a sufficiently long time (typically one month or a whole season), which enables one to take dynamic effects into account by an empirically determined gain and/or loss utilization factor."*

In general air conditioning systems are defined as systems capable of achieving, maintaining and controlling, within the rooms they serve, predefined conditions of temperature, humidity, air quality and movement. It is of the highest importance that these parameters be taken into account when studying and implementing the plants.

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[1] Ancient Taoist geomantic art of China, auxiliary to architecture, similar to Western geomancy. It takes into consideration aspects of the psyche and astrology. It is a pseudoscience, as there is no evidence to support the assumptions on which it is based [27].

4.1.1 Heating demand

The evaluation of the heating demand is conducted in accordance with the methodology delineated in the standard UNI EN 12831-2017.

A steady-state calculation is employed for the evaluation of winter heat loads.

The calculation method presented below is based on the following fundamental assumptions:

- Uniform temperature distribution
- Heat losses calculated under permanent conditions assuming constant properties, such as temperature values and characteristics of the building elements
- Heated rooms
- Identical air temperature and operative temperature values

The application of the calculation method is based on the determination of the heat losses to the outside, which are determined by the following terms:

- Thermal conduction through the surrounding surfaces
- Design heat losses due to ventilation or infiltration through the building envelope.

The following section presents the procedure [16] for defining the winter heat load in a step-by-step manner:

Step 1: Definition of the external design conditions ($\theta_{des,win}$)

The buildings are located in Padova. To assess the winter energy demand, it is necessary to employ the design outdoor temperature and the degree day of the city in question, in accordance with the standard UNI 10349.

Location	*DD*	*Zone*	$\theta_{des,win}$
Padova	2383	E	-5 °C

Table 4.1 – Characteristics of the location: UNI 10349

Zone E encompasses climates with degrees day range of 2101 to 3000.

The concept of climatic zones is defined in DPR n.412 of 26 August 1993. In particular, the national territory has been subdivided into six climatic zones according to Degree Day, which is to say, according to the average climate of the municipality regardless of geographical location. The Degree Day is an index of the climate, whereby a higher Degree Day value correlates with a colder temperature. Zone E is associated with the permitted system switch-on period, which runs from 15 October to 15 April, with a maximum of 14 hours per day.

Step 2: Identification of heated zones

In consideration of the illustration depicted in figure 4.1, it can be ascertained that all three rooms are heated. It is assumed that the adjacent rooms on the east and west sides of the buildings are unheated for reasons pertaining to safety margins.

Step 3: definition of the space and the building envelope

The buildings under investigation are industrial structures erected in the early 2000s. In the absence of precise stratigraphic data regarding the walls and roof, several hypotheses have been formulated based on databases from that period. It seems likely that the materials employed were those described in the document of 2011 produced by “Comitato Termotecnico Italiano” [17]. The document presents a series of potential options, applicable to both residential and industrial contexts. In consideration of the prefabricated sheds, one of the options exhibiting a low degree of insulation was determined to be the most suitable, given the low energy class of the structure.
The table below provides a detailed overview of the wall and roof stratigraphy, accompanied by the respective surface resistance values:

WALL							
Layer	s [m]	λ [W/(m K)]	ρ [kg/m^3]	R [(m^2 K)/W]	c_p [J/(kg K)]	c_v [MJ/(m^3 K)]	m_{sp} [kg/m^2]
R_{se}				0.040			
Precast concrete	0.06	0.58	1400	0.103	1000	1.40	84.0
Polystyrene	0.08	0.16	30	0.500	670	0.02	2.4
Precast concrete	0.06	0.58	1400	0.103	1000	1.40	84.0
R_{si}				0.125			

Table 4.2 – Characteristics of the wall

where:
$$R = \frac{s}{\lambda} \tag{5}$$

$$m_{sp} = \sum_q \rho_q * s_q \tag{6}$$

The same considerations that were made for the walls also apply to the roof.

The stratigraphy of the roof is presented in detail in the table below:

ROOF							
Layer	s [m]	λ [W/(m K)]	ρ [kg/m^3]	R [(m^2 K)/W]	c_p [J/(kg K)]	c_v [MJ/(m^3 K)]	m_{sp} [kg/m^2]
R_{se}				0.040			
Bituminous WPR membrane	0.01	0.17	1200	0.059	1000	1.2	12
Concrete screed	0.12	1.06	2000	0.113	1000	2.0	240
Mortar	0.02	1.40	2000	0.014	1000	2.0	40
Reinforced concrete	0.04	1.60	2400	0.025	1000	2.4	96
Slab	0.18	0.55	900	0.327	1000	0.9	162
R_{si}				0.100			

Table 4.3 – Characteristics of the roof

The boundary condition considered for the internal and external surfaces was the one of the 3rd type: fixed heat transfer coefficient and temperature of the boundary fluid (Neumann condition). This allowed the evaluation of the thermal resistance of the inner and outer surfaces, which contribute to the heat flux:

$$R_{si} = \frac{1}{\alpha_{s,j}} \tag{7}$$

The term α is used to denote the global heat transfer coefficient, which encompasses both the convective and radiative components. The UNI EN ISO 6946 standard defines different values depending on whether the surface in question is horizontal or vertical, and furthermore, distinguishes between upward and downward flow.
In the case under consideration, the coefficients employed are summarised in the following table:

Surface	α_{int} $[W/(m^2\ K)]$	α_{ext} $[W/(m^2\ K)]$
Wall (vertical surface)	8	25
Roof (horizontal surface descendent flux)	10	25

Table 4.4 – global heat transfer coefficient for inner and outer surfaces

In addition to the lack of detailed information on the stratigraphy of the walls and roof, there is also a paucity of data regarding the materials used to fabricate the windows and doors. A visual analysis suggests that the windows are likely to be 4-6-4 double glazing, without any treatment and without the use of any gas in the cavity. In order to calculate the transmittance of the window (comprising the glass and frame), the procedure set out in EN ISO 10077 is followed.
The dimensions of the windows in question are identical for the entire structure. For safety margin, doors are also to be considered as windows.
The dimensions of the doors and windows are provided in the following table:

Element	*Length [m]*	*Height [m]*
Glass	*1.4*	*1.7*
Shop door	*2.3*	*2.5*
Other doors	*2.3*	*0.9*

Table 4.5 – Dimensions of the glazed elements

The characteristics of the glazed elements are evaluated in the following section.

GLAZED ELEMENTS						
s_f [m]	A_f [m^2]	U_g[2] [W/(m^2 K)]	A_g [m^2]	ψ_g [-]	p_g [m]	U_w [W/(m^2 K)]
0.06	0.30	3.30	2.08	0.06	5.8	3.04

Table 4.6 – Characteristics of the glazed elements

where:

$$U_w = \frac{U_g * A_g + s_f * A_f + p_g * \psi_g}{A_g + A_f} \tag{8}$$

The overall transmittance of the glazed elements, comprising both the frame and glass, is $3.04 \frac{W}{m^2 * K}$.

[2] https://biblus.acca.it/guida-al-vetro-in-edilizia-i-vetri-bassoemissivi/

The overall transmittance of the walls, roof and windows is summarised in the following table:

Element	*Resistance* $[(m^2\ K)/W]$	*Transmittance* $[W/(m^2\ K)]$
Walls	0.872	1.147
Roof	0.679	1.474
Glazed	0.329	3.038

Table 4.7 – Resume of the transmittances of the building envelope

As illustrated in figure 4.1 all unconnected buildings are representative of the aforementioned layout. The glazed elements within the building envelope are arranged as follows:

Room	*Windows*	*Doors*
Shop	6	1
Office	3	1
Warehouse	5	1

Table 4.8 – Distribution of glazed elements in the building

In accordance with the standard UNI 7357, if a room to be heated borders a non-heated room, specific coefficients must be employed, taking into account the particular circumstances of each case. In this report, however multiple buildings are considered. The first and the last buildings (figure A.2) are situated in proximity to the external environment, whereas the remaining structures are positioned in proximity to other buildings. In order to ensure consistency in the calculations, it will be assumed that all the walls facing east and west are adjacent to non-heated spaces.

Room	A_{FLOOR} $[m^2]$	*h* $[m]$	*V* $[m^3]$	A_W $[m^2]$	L_{wall} $[m]$		S_{wall} [m^2]	
					outdoor	*unheated space*	*outdoor*	*unheated space*
SHOP	152	6	912	20.0	8	19	28.0	114.0
OFFICE	133	6	798	9.2	7	19	32.8	114.0
WAREHOUSE	300	6	1800	14.0	15	40	76.0	240.0

Table 4.9 – Geometric parameters of the typical non-connected users

As previously stated, some walls are situated in proximity to the external environment, while others are adjacent to unheated rooms. Accordingly, the two surfaces are distinguished in table 4.9, as they will be treated separately in the dispersion calculation.

Step 4: evaluation of the required peak power for heating and the corresponding energy demand.

Prior to start the calculation, it is of the utmost importance to stipulate that all the procedures for the heating demand have been conducted with the assumption of neglecting heat gain due to human activity, electrical equipment, and lighting. This is done in order to ensure the inclusion of a further safety margin.

The overall design heat loss is calculated as the sum of the transmission and ventilation losses, as specified in the standard UNI EN ISO 12831:

$$q_{tot} = q_T + q_V \tag{9}$$

In order to evaluate the two terms, it is necessary to find out the transmission (H_T) and the ventilation (H_V) coefficients respectively.

About transmission losses, the following procedure is recommended:

$$q_T = H_T * \left(\theta_{int,des} - \theta_{win,des}\right) \tag{10}$$

and:

$$H_T = (U_{wl} * A_{wl} + U_w * A_w + U_{fl} * A_{fl}) * b_u \tag{11}$$

To calculate the transmission coefficient, it is also necessary to consider the impact of thermal bridges, which are represented by the coefficient b_u. In the field of building science, a thermal bridge is defined as a part of a building envelope where the otherwise uniform thermal resistance is significantly altered. There are two main scenarios in which thermal bridges may occur:

- Discontinuities in the composition of the building envelope
- Variations in the thickness of the building fabric.

In accordance with UNI 7357, a preliminary approximation to consider the effect of thermal bridges in "*prefabricated concrete industrial buildings*" requires an increase of 30% in the coefficient of transmission losses for opaque surfaces. In order to ensure a sufficient safety margin, the overall coefficient of transmission of the wall is considered.

As previously stated, the transmission coefficient (H_T) must then be further corrected for surfaces bordering unheated rooms. In accordance with UNI 7357, the transmission coefficient for those walls must be weighted with the value b_u. The case considered in this thesis is that of a "*room with at least 2 external walls and windows*", in which b_u = 0.6.

The second term for the evaluation of the winter peak power calculation concerns ventilation heat losses.

Given that the height of the buildings in question exceeds 4 meters, ventilation heat losses are affected. This is accounted for by the coefficients derived from the standard UNI EN 12831:

Heat emission system	*Air temperature gradient* θ_{grad} *[K/m]*	*Difference between air and operative temperature* $\Delta\theta_{air-oper}$ *[K]*
Warm air heating without additional destratification	1,00	0
Warm air heating with additional destratification	0,35	0

Table 4.10 – Coefficients for buildings higher than 4 meters for the ventilation heat losses

In the context of the winter evaluation, the internal design temperature employed is set at 20°C. Nevertheless, in view of the fact that the height in question exceeds 4 meters, it is imperative to consider the parameters set out in table 4.10 when evaluating the ventilation losses:

$$\theta^{\#}_{int,win} = \theta_{int,des} + \theta_{grad} * \left(\frac{h_i}{2} - h_{occup}\right) - \Delta\theta_{rad} = 20 + 1 * \left(\frac{9}{2} - 2\right) = 22°C \quad (12)$$

The corrected internal design temperature is set at 22 °C.

For what concern the transmission losses (H_T) the temperature is considered 20°C.

Considering the requisite safety margin, the possibility of additional destratification devices was excluded from the hypothesis. However, in the actual case study, some of the plants will be presented in the following sections with the inclusion of destratifiers.

To calculate the ventilation coefficient:

$$q_V = H_V * \left(\theta^{\#}_{int,win} - \theta_{win,des}\right) \quad (13)$$

where:

$$H_V = \frac{\rho * c_p * G_{air,peak}}{3600} \quad (14)$$

and: $$\dot{G_J} = n_j * V_{room} \qquad (15)$$

For the evaluation of peak power in winter the air change rate to consider is 0.5 h^{-1} as stated in the standard UNI 10339.

Room		H_T *[W/K]*	q_T *[W]*	H_V *[W/K]*	q_V *[W]*	$G_{air,peak}$ *[m³/h]*	q_{peak} *[kW]*	P_V *[W/m³]*	P_S *[W/m²]*
SHOP	*ext*	412.0	10299.9	153.1	4132.7	456.0	19.3	21.2	126.9
	unheated	324.2	4862.4						
OFFICE	*ext*	340.1	8501.4	133.9	3616.1	399.0	16.3	20.4	122.3
	unheated	276.7	4150.3						
WAREHOUSE	*ext*	743.3	17094.8	302.1	7552.5	900.0	32.3	18.0	107.8
	unheated	592.6	7704.3						
TOTAL							67.9		

Table 4.11 – Evaluation of peak power for heating demand

Regarding the calculation of energy requirements, the standard UNI 10339 is again the primary reference point for the determination of air change rate coefficients. In consideration of the intended use of the local context, the coefficient 0.3 h^{-1} is employed for the evaluation of $G_{air,energy}$ in accordance with formula (15).

The following formula is used to calculate the energy demand:

$$E_{des} = \frac{DD*\left(H_T+\left(\frac{\dot{G}_{air,energy}}{3600}*\rho*c_p\right)\right)*24}{1000} \qquad (16)$$

Given that the two walls in the same room have disparate heat loss transmission coefficients (H_T), it is deemed prudent to consider the higher value, that of the wall bordering the external environment, in the calculation to ensure an adequate margin of safety.

Room	$G_{air,energy}$ *[m³/h]*	*Energy demand [kWh]*	*Specific energy demand [kWh/(m² y)]*
SHOP	273.6	28815.4	189.6
OFFICE	239.4	24044.3	180.8
WAREHOUSE	540.0	52874.7	176.2
TOTAL		105734.4	180.7

Table 4.12 – Heating demand calculations

4.1.2 Cooling demand

In contrast with the winter case, the evaluation of cooling demand needs the consideration of extreme temporal variability in heat flows (heat gains), predominantly due to the substantial and rapid fluctuations in solar radiation throughout the day. The instantaneous heat flow that penetrates a room is not transformed immediately into a cooling load due to the differing level of thermal inertia exhibited by structures. It is of the utmost importance to consider this factor in order to prevent errors in the estimation of the potential of the system.

The internal design temperature for the summer season is set at 26 °C in the office and shop, and 28°C for the warehouse, in accordance with the standard UNI EN 16798.

There are several methods for the evaluation of the cooling loads. In this discussion the Carrier method is employed. This method allows for the obtaining of an hourly profile of the loads, whereby the cooling load is defined as the sum of six terms:

$$q_{sum,tot} = q_V + q_{inf} + q_w + q_{op} + q_{sol} + q_{ig} \tag{17}$$

The following section provides a detailed explanation of the terms included in the equation (17), accompanied by the relevant calculations.
For the sake of simplicity, the results of only one of the three rooms are presented; the procedure for the other rooms is identical, with only the numbers differing.
The main results are presented in Appendix B.

In contrast to the winter evaluation, which is static, the Carrier method requires the definition of a time profile for the outside temperature. The external air design temperature in summer is calculated according to the following formula:

$$\theta_{air,sum} = \theta_{air,ext,max} - p(\tau) * \Delta\theta_{air,ext} \tag{18}$$

The standard UNI 10349 is employed, which in the case of Padova, yielded the following parameters:

$\theta_{air\ ext,max}$	$\Delta\theta_{air,\ ext}$	$p(t)$	$\theta_{air,\ sum}$ [°C]	*Hour*
34	13	0.98	21.26	6
		0.93	21.91	7
		0.84	23.08	8
		0.71	24.77	9
		0.56	26.72	10
		0.39	28.93	11
		0.23	31.01	12
		0.11	32.57	13
		0.03	33.61	14
		0.00	34.00	15
		0.03	33.61	16
		0.1	32.7	17
		0.21	31.27	18
		0.34	29.58	19
		0.47	27.89	20
		0.58	26.46	21
		0.68	25.16	22
		0.76	24.12	23
		0.82	23.34	24
		0.87	22.69	1
		0.92	22.04	2
		0.96	21.52	3
		0.99	21.13	4
		1.00	21.00	5

Table 4.13 – Summer design temperature for Padova

The characteristics of the building can be found out in the chapter 4.1.1.

The six terms that appear in the equation 17 are analysed and explained in detail below.

1st term – q_V: convective heat flow due to the incoming and outgoing air rates in the room

$$q_V = G_{air} * c_p * (\theta_{imm} - \theta_{int}) \tag{19}$$

This contribution is only considered in the presence of mechanical ventilation. Indeed, the mass flow rate, specific heat, and the inlet temperature are all referenced to the inlet air. In this initial approximation this term is disregarded since it does not take into account any mechanical ventilation system.

Term	q_v *[W]*	*Reason*
Ventilation	0	Because there is not consider any ventilation system

Table 4.14 – Ventilation gains for shop

2nd term – q_{inf}: infiltration gains

$$q_{inf} = G_{air} * c_p * (\theta_{ext} - \theta_{int}) \tag{20}$$

The parameters that contribute to the infiltration and ventilation gains are, for the most part, similar. The sole distinction lies in the Δθ. As previously stated, due to the dynamic nature of the evaluation, the value must be calculated on an hourly basis.

The air flow rate is determined by the calculation presented in equation 15, which assesses the peak power required for the heating. The Δθ is calculated between the external design temperature (table 4.13) and the internal setpoint chosen: 26°C during the summer season for shop and offices, and 28°C for the warehouse.

Term	*Hour*	q_{inf} *[W]*
Infiltration	6	-725.52
	7	-626.03
	8	-446.95
	9	-188.27
	10	110.21
	11	448.48
	12	766.85
	13	1005.63
	14	1164.82
	15	1224.51
	16	1164.82
	17	1025.53
	18	806.65
	19	547.97
	20	289.29
	21	70.41
	22	-128.57
	23	-287.76
	24	-407.15
	1	-506.64
	2	-606.13
	3	-685.73
	4	-745.42
	5	-765.32

Table 4.15 – Infiltration gains for shop

In the table 4.15 there are some negative values. It means that for some hours during the day the heat flow is reversed because the indoor temperature overcome the outdoor one. The highest recorded value is observed at 3 p.m. when the difference between the internal and external temperatures is at its maximum value.

3rd term – q_W: conduction heat flow through the glazing element

$$q_W = \sum_k U_k * S_k * (\theta_{ext} - \theta_{int}) \quad (21)$$

In the evaluation of summer heat loads, as in winter, doors are considered as windows to have a safety margin. In the equation 21 U_k and S_k are referred to the windows and doors of the room considered.

Term	*Glazed area [m²]*	*Hour*	*q_w [W]*
Windows	20.03	6	-288.42
		7	-248.87
		8	-177.67
		9	-74.84
		10	43.81
		11	178.28
		12	304.84
		13	399.77
		14	463.05
		15	486.78
		16	463.05
		17	407.68
		18	320.66
		19	217.83
		20	115.00
		21	27.99
		22	-51.11
		23	-114.39
		24	-161.85
		1	-201.40
		2	-240.95
		3	-272.60
		4	-296.33
		5	-304.24

Table 4.16 – Conduction through windows for shop

Windows and doors are considered to have no thermal capacity: no thermal inertia.
As observed in table 4.16, which is analogous to table 4.15, some values are negative. Additionally, the highest value is still recorded at 3 p.m. which coincides with the maximum difference between indoor and outdoor temperatures.

4th term – q_{op}: conduction heat flow through the opaque wall

$$q_{op} = \sum_i U_i * S_i * \Delta T_{eq,i} \tag{22}$$

The opaque elements have a capacity that, as a consequence of the effect of inertia, results in a delay in the peak load over time. Moreover, the cooling load becomes smoother throughout the day.

In order to guarantee a safety margin when evaluating the summer peak power, the building is treated as if it were isolated. Consequently, all surfaces (north, south, west and east) are included into the calculation. In this analysis the heat flux through the floor is not considered, given that it is lower, and furthermore, the calculation of this flux is more challenging.

The transfer of heat through opaque external walls and the roof is attributable to two primary factors: solar radiation absorbed by external surfaces and to the temperature differential between external and internal environments. Consequently, the amount of heat transferred is contingent upon a number of factors, including latitude, the specific period of the year, the time of the day, the degree of solar exposure, the mass and composition of the wall and roof materials.

In order to facilitate the calculations, the concept of sol-air temperature ($\theta_{sol\text{-}air}$) is introduced. The term is defined as the temperature of the outside air that, in the absence of any solar radiation, would result in the same heat flux as observed in reality. This is due to incident solar radiation, exchanges by radiation with the sky and surrounding surfaces, and convective exchanges with the outside air. To obtain this value, complex formulae may be employed, but they can be simplified by utilising tabulated values from manuals. In the literature, tabulated values are provided for specific orientations, masses and the wall types (UNI EN ISO 52016). The corresponding values of the specific mass are derived from table 4.2 and table 4.3 respectively for the walls and the roof.

In equation 22, the transmittance (U_i) and the surface (S_i) are referred to the specific element considered. It is necessary to analyse the wall on a case-by-case basis, as the value of $\Delta T_{eq,i}$ is dependent on the specific orientation.

Term	Wall name	Orient.	S $[m^2]$	Hr.	ΔT_{eq} (β)	q_{op} (β) [W]	ΔT_{eq} (ξ)	q_{op} (ξ) [W]	ΔT_{eq} (ψ)	q_{op} (ψ) [W]	q_{op} [W]
Opaque surface	β	S	28	6	-0.8	-25.7	0.8	104.6	6.9	1545.6	1624.5
	ξ	W	114	7	-1.9	-60.9	0.2	26.1	6.4	1433.6	1398.8
	ψ	OR	152	8	-2.5	-80.2	-0.3	-39.2	5.8	1299.2	1179.7
				9	-1.9	-60.9	-0.3	-39.2	5.8	1299.2	1199.0
				10	-1.4	-44.9	-0.3	-39.2	6.4	1433.6	1349.4
				11	3.6	115.5	0.8	104.6	6.9	1545.6	1765.6
				12	6.4	205.3	1.9	248.4	8.5	1903.9	2357.7
				13	10.8	346.5	3.6	470.7	11.9	2665.5	3482.7
				14	13.1	420.2	5.3	693.0	14.1	3158.3	4271.5
				15	13.6	436.3	10.2	1333.6	15.2	3404.7	5174.6
				16	14.1	452.3	14.1	1843.6	17.5	3919.9	6215.8
				17	12.5	401.0	18.6	2431.9	19.2	4300.7	7133.6
				18	10.8	346.5	21.9	2863.4	20.3	4547.1	7757.0
				19	8.1	259.8	22.5	2941.9	20.3	4547.1	7748.8
				20	6.4	205.3	19.7	2575.8	19.2	4300.7	7081.8
				21	5.3	170.0	15.2	1987.4	18.6	4166.3	6323.7
				22	4.2	134.7	8.5	1111.4	18.6	4166.3	5412.4
				23	3	96.2	5.3	693.0	17.5	3919.9	4709.1
				24	1.9	60.9	3	392.2	16.4	3673.5	4126.7
				1	0.8	25.7	2.5	326.9	14.7	3292.7	3645.2
				2	0.2	6.4	1.9	248.4	12.5	2799.9	3054.8
				3	0.2	6.4	1.3	170.0	10.8	2419.1	2595.5
				4	-0.3	-9.6	1.3	170.0	9.7	2172.7	2333.1
				5	-0.8	-25.7	0.8	104.6	7.4	1657.6	1736.5

Table 4.17 – Conduction heat flow through the wall for the shop

The values of ΔT_{eq} taken from the tables are the one that correspond at specific mass of 300 kg/m^2 for the walls (at the specific orientation) and 400 kg/m^2 for the roof.

It is evident that the proportionality of the contributions of the three opaque surfaces to the exchange surface area results in greater losses for the roof than for the side walls. In contrast to the preceding terms, the highest value is observed to shift in time at 6p.m.

5th term – q_{sol}: solar radiation

$$q_s = \sum_k S_k * I_{x,k} * f_{a,k} * C_{s,k} \tag{23}$$

In contrast to the winter case, the solar radiation assumes a central role in the summer evaluation. The solar radiation transmitted through the glazing components is dependent on the orientation:

- $I_{x,k}$: maximum incident solar radiation. It is taken from literature and, for the design calculation, the value of July is considered.
- $f_{a,k}$: attenuation factor. It depends on the orientation, hour of the day and also on the thermal capacity of the room (M_R). The latter can be evaluated through:

$$M_R = \frac{\sum_j m_{f,j} * s_j + 0.5 * \sum_r m_{f,r} * s_r}{s_{floor}} \tag{24}$$

  The thermal inertia of the j-th generic wall facing outside has to be considered as whole, while the r-th internal wall is counted for half of the thermal capacity because the other half is considered in the adjacent room. Also in this case the value of the attenuation factor can be found tabulated in literature. In this case study the attenuation factor taken from tabulated values is that which corresponds to 490 kg/m^2 for each orientation.
- $C_{s,k}$: shading coefficient. It is calculated as the ratio between the solar factor of the analysed glass and the solar factor of a reference one (clear glass, 3 mm):

$$C_s = \frac{I_{gl}}{I_{ref,gl}} \tag{25}$$

  It depends on the shading elements and on the type of glass UNI EN ISO 52016. In the case study considered the building is equipped with a "*double clear glazing*" that correspond to $C_s = 0.77$.

So, with all this information the solar radiation for the single room can be calculated:

Term	*Window name*	*Orientation*	S_w $[m^2]$	C_s [-]	I_x	*Hour*	f_a	q_s [W]
Solar radiation	β	S	20.03	0.77	276	6	0.07	297.97
						7	0.06	255.41
						8	0.12	510.81
						9	0.2	851.36
						10	0.3	1277.03
						11	0.39	1660.14
						12	0.48	2043.25
						13	0.54	2298.66
						14	0.58	2468.93
						15	0.57	2426.36
						16	0.53	2256.09
						17	0.45	1915.55
						18	0.37	1575.01
						19	0.31	1319.60
						20	0.27	1149.33
						21	0.23	979.06
						22	0.2	851.36
						23	0.18	766.22
						24	0.16	681.08
						1	0.14	595.95
						2	0.12	510.81
						3	0.11	468.25
						4	0.1	425.68
						5	0.08	340.54

Table 4.18 – Solar heat gain for the shop

It is evident that as the quality of the glass improves, the dispersion to the exterior decreases. In this case the maximum contribution is observed at 2 p.m. The maximum value for the other rooms is contingent upon the attenuation factor, which is in turn dependant upon the orientation.

6^{th} term – q_{ig}: internal gains due to electric appliances, people and lighting

$$q_{ig} = q_{ig,light} + q_{ig,ppl} + q_{ig,el.app.} \tag{26}$$

In calculating the summer peak demand, it is imperative not to neglect the internal gains from lighting, people and electrical appliances. Such heat loads must be subtracted.

The first contribution to be calculated is that pertaining to the lighting system.

$$q_{ig,light} = q_{max,light} * f_{utilization} \tag{27}$$

The data pertaining to the evaluation of the internal gains resulting from the illumination are presented in table 4.20 [18]. In this case, the assumption is that fluorescent lamps are present, and the installed power is calculated in accordance with the standard UNI EN 12464, taking into account the different levels of illumination required in each room.

The second term refers to the presence of persons on the site. In this case, the occupancy rate is determined in accordance with the intended use of the room, with reference to the standard UNI EN 16798.

The internal contribution provided by individuals is determined by the specific activity they are engaged in within the designated area, as illustrated in the subsequent table.

Room	*Occupancy rate [ppl/m^2]*	*Floor area [m^2]*	*N. of people [ppl]*	*Gain [W/ppl]*
Shop	0.10	152	19	65
Office	0.12	133	16	65
Warehouse	0.05	300	15	70

Table 4.19 – Presence of people in the rooms

The third term concerns the electrical appliances that are employed.

In the context of this analysis, the focus is on computers and printers in the offices.

In the context of the warehouse and shop, the lighting system represents the sole aspect that is subject to consideration.

Room	*Electric power installed for lighting [W/m²]*	*Other electric appliances [W/m²]*
Shop	20	-
Office	20	5
Warehouse	10	-

Table 4.20 – Internal gain due to electric appliances

It can be observed that the contribution of the lighting system varies depending on the specific room in question, as the lighting requirements are contingent upon the intended use, in accordance with the UNI EN 12646 standard.
The accumulation factor is derived from the standard UNI EN 15193 with consideration given to the utilisation of fluorescent lamps.

In all the three rooms, the presence of people is considered between the 7a.m. and 8p.m.

Table 4.21 presents the hourly contribution provided by the internal gains.

Term	*S_{floor} [m^2]*	*Hour*	*$f_{acc,light}$*	*Electric power [W/m^2]*	$q_{ig,light}$ [W]	$q_{ig,ppl}$ [W]	q_{ig} [W]
Internal gains	152	6	0.00	20	0.0	0.0	0.0
		7	0.58	20	11.6	1235	2998.2
		8	0.85	20	17.0	1235	3819.0
		9	0.88	20	17.6	1235	3910.2
		10	0.88	20	17.6	1235	3910.2
		11	0.9	20	18.0	1235	3971.0
		12	0.92	20	18.4	1235	4031.8
		13	0.93	20	18.6	1235	4062.2
		14	0.94	20	18.8	1235	4092.6
		15	0.94	20	18.8	1235	4092.6
		16	0.94	20	18.8	1235	4092.6
		17	0.95	20	19.0	1235	4123.0
		18	0.95	20	19.0	1235	4123.0
		19	0.95	20	19.0	1235	4123.0
		20	0.48	20	9.6	1235	2694.2
		21	0.00	20	0.0	0.0	0.0
		22	0	20	0	0	0
		23	0	20	0	0	0
		24	0	20	0	0	0
		1	0	20	0	0	0
		2	0	20	0	0	0
		3	0	20	0	0	0
		4	0	20	0	0	0
		5	0	20	0	0	0

Table 4.21 – Internal gain for the shop

At this point, it is possible to resume all the contributions in a single table, thus affording a more wide-ranging overview of the room. In order to facilitate a more comprehensive understanding, the hourly plot is presented.

Hour	q_v [W]	q_{inf} [W]	q_w [W]	q_{op} [W]	q_{ig} [W]	q_s [W]	q_{tot} [W]	Peak power for cooling [W]
6	0.0	-725.5	-288.4	1624.5	0.0	298.0	908.5	
7	0.0	-626.0	-248.9	1398.8	2998.2	255.4	3777.5	
8	0.0	-446.9	-177.7	1179.7	3819.0	510.8	4884.9	
9	0.0	-188.3	-74.8	1199.0	3910.2	851.4	5697.4	
10	0.0	110.2	43.8	1349.4	3910.2	1277.0	6690.7	
11	0.0	448.5	178.3	1765.6	3971.0	1660.1	8023.5	
12	0.0	766.9	304.8	2357.7	4031.8	2043.3	9504.4	
13	0.0	1005.6	399.8	3482.7	4062.2	2298.7	11248.9	
14	0.0	1164.8	463.0	4271.5	4092.6	2468.9	12460.9	
15	0.0	1224.5	486.8	5174.6	4092.6	2426.4	13404.9	
16	0.0	1164.8	463.0	6215.8	4092.6	2256.1	14192.3	
17	0.0	1025.5	407.7	7133.6	4123.0	1915.5	14605.4	14605.4
18	0.0	806.6	320.7	7757.0	4123.0	1575.0	14582.3	
19	0.0	548.0	217.8	7748.8	4123.0	1319.6	13957.2	
20	0.0	289.3	115.0	7081.8	2694.2	1149.3	11329.6	
21	0.0	70.4	28.0	6323.7	0.0	979.1	7401.2	
22	0.0	-128.6	-51.1	5412.4	0.0	851.4	6084.1	
23	0.0	-287.8	-114.4	4709.1	0.0	766.2	5073.2	
24	0.0	-407.2	-161.9	4126.7	0.0	681.1	4238.8	
1	0.0	-506.6	-201.4	3645.2	0.0	595.9	3533.2	
2	0.0	-606.1	-241.0	3054.8	0.0	510.8	2718.5	
3	0.0	-685.7	-272.6	2595.5	0.0	468.2	2105.4	
4	0.0	-745.4	-296.3	2333.1	0.0	425.7	1717.0	
5	0.0	-765.3	-304.2	1736.5	0.0	340.5	1007.5	

Table 4.22 – Resume of the main results for the shop. In green the hour at which the maximum occurs

The results for each room are presented in appendix B, with a detailed account of the contribution of each term.

The peak power for the cooling is determined as the maximum value calculated over the course of the entire day for each room. With regard to the shop, the maximum value is recorded at 5 p.m.

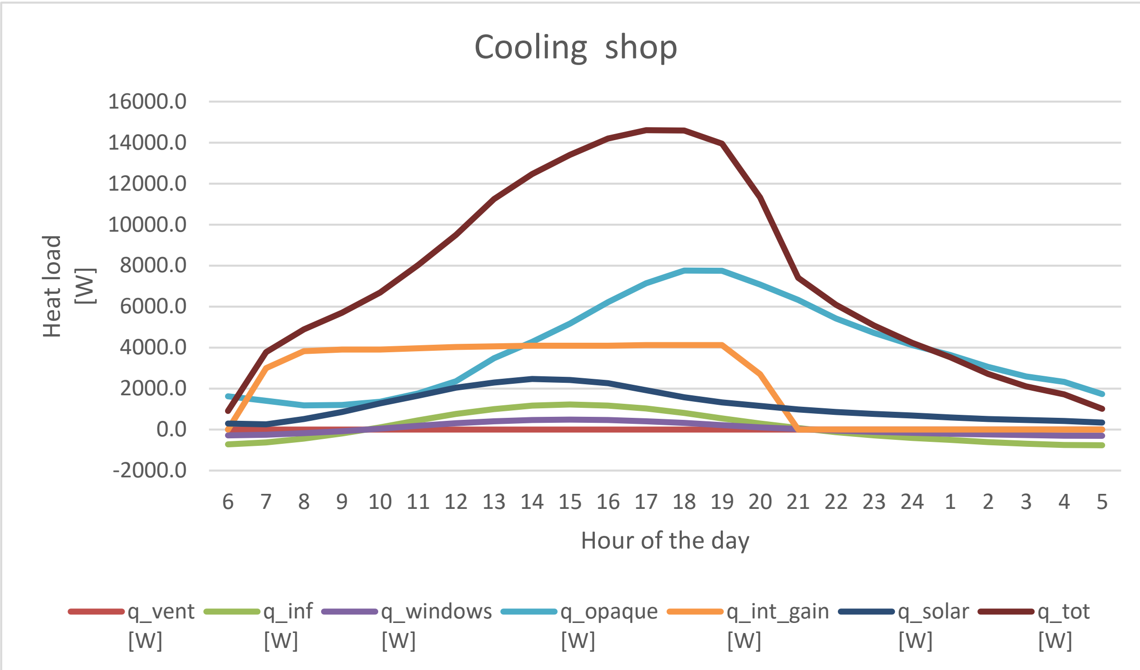

Figure 4.2 – Plot of the results of the shop

The detailed chart for each room is in the appendix C.

As illustrated in the chart above, the cooling load calculation exhibits an hourly profile that is not constant and also varies from room to room.

For what concern the shop, the most significant contributions are attributable to internal gains and through opaque walls. The influence of solar radiation and glazed elements is relatively insignificant, given that the glazed surface is modest in comparison with the total wall and roof surface area of the buildings. It is evident that the impact of each term within the room is contingent upon the utilisation of the room itself, the orientation and the presence of glazed components.

Figure 4.3 presents the contribution of each room during the day.

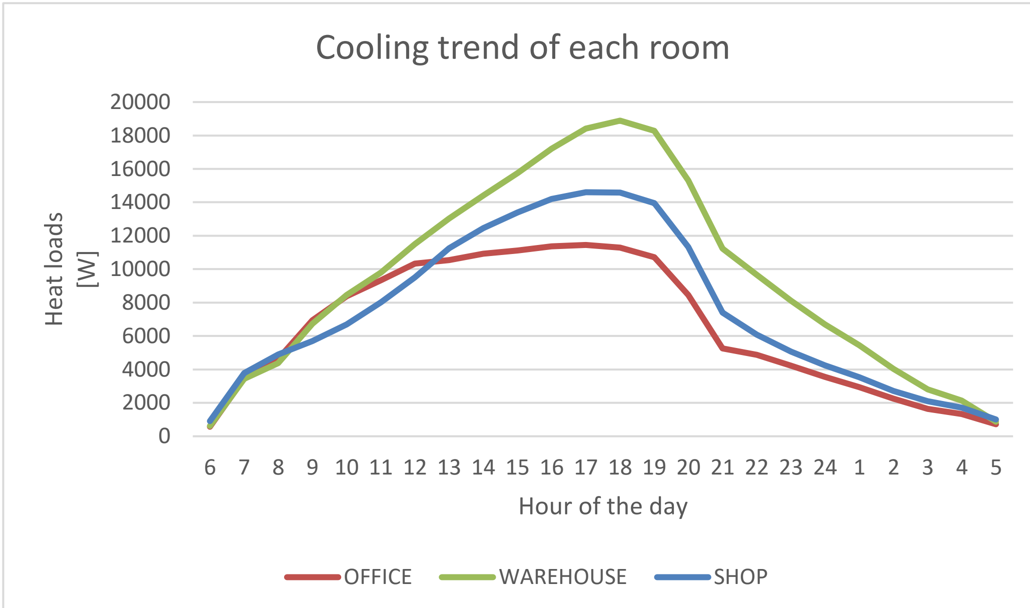

Figure 4.3 – Contribution of each room during the day

As demonstrated in the graph, the warehouse is the primary contributor to the cooling load. Despite having a setpoint temperature 2°C higher than that of the shop and offices, the large opaque surface area of the warehouse results in a significantly higher dispersion term than that of the other rooms. The contribution of the shop and the offices is not significantly disparate, given that they are of comparable dimensions and similarly exposed. The distinction between the two lies in the greater solar radiation in the shop, which can be attributable to the larger glazed area. The influence of electrical appliances in the office is relatively limited, given that a low occupancy rate was assumed: 6.5 workstations per 100 m^2.

For each room, the cooling load is reported for each hour as the sum of all contributing components. Subsequently, the sum of all the rooms is calculated for each hour. To determine the peak load power for cooling, the maximum value must be taken.

To gain a more comprehensive understanding of the contributions made, the main results can be summarised in a single table, which will facilitate a more detailed overview.

q_{tot} [W]					
Hour	*SHOP*	*OFFICES*	*WAREHOUSE*	$q_{tot,sum}$ [W]	$q_{peak,sum}$ [W]
6	908.5	569.8	617.0	2095.3	44760.8
7	3777.5	3484.6	3436.2	10698.3	
8	4884.9	4625.3	4386.2	13896.4	
9	5697.4	6947.3	6741.9	19386.7	
10	6690.7	8373.2	8454.2	23518.0	
11	8023.5	9335.9	9794.0	27153.4	
12	9504.4	10334.3	11492.7	31331.4	
13	11248.9	10547.4	13034.5	34830.8	
14	12460.9	10917.9	14412.1	37790.8	
15	13404.9	11119.6	15752.8	40277.3	
16	14192.3	11371.4	17214.0	42777.8	
17	14605.4	11450.1	18418.7	44474.2	
18	14582.3	11289.9	18888.7	44760.8	
19	13957.2	10721.0	18275.7	42953.9	
20	11329.6	8450.5	15308.3	35088.4	
21	7401.2	5254.2	11227.8	23883.2	
22	6084.1	4867.4	9668.3	20619.8	
23	5073.2	4231.3	8112.4	17416.9	
24	4238.8	3549.6	6701.0	14489.4	
1	3533.2	2939.1	5458.4	11930.7	
2	2718.5	2249.5	4034.5	9002.5	
3	2105.4	1644.7	2811.4	6561.5	
4	1717.0	1327.6	2137.2	5181.8	
5	1007.5	721.6	870.8	2599.8	

Table 4.23 – Resume of the building room by room. Green line represents the hour at which the maximum occurs.

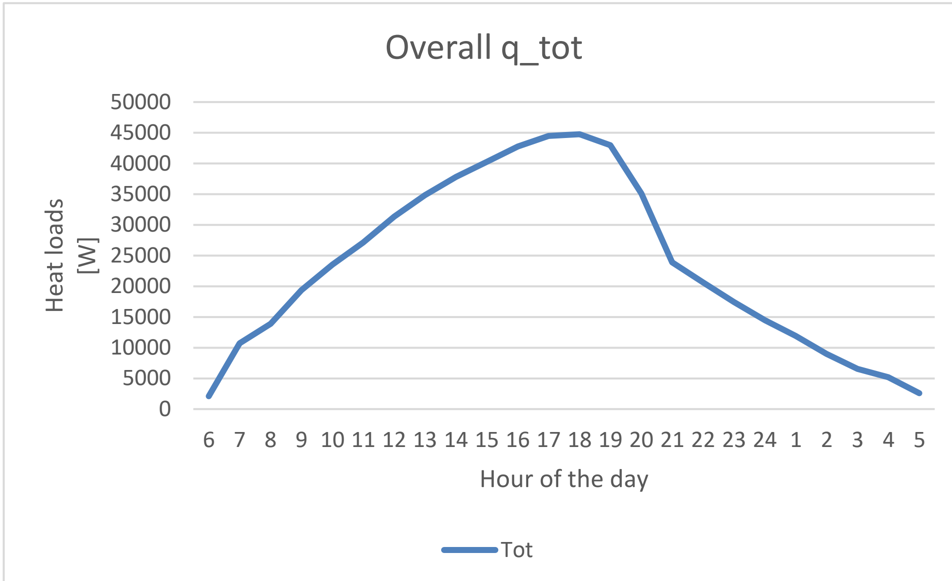

Figure 4.4 – Sum of the three rooms of the case study

The case study considered presents the maximum at 6 p.m. This indicates that at this hour, the sum of the contributions of all the rooms is the maximum of the day. Therefore, despite the maximum external temperature occurring at 3 p.m. as indicated in table 4.13, the maximum cooling load is shifted forward by 3 hours.

4.2 Other possible types of user

As previously stated, the most probable type of user that can be expected in building C is the one described in chapter 4.1. Nevertheless, in order to conduct a more comprehensive and expansive examination of the issue, a preliminary investigation into the building's characteristics is undertaken in this section. This is done on the assumption that the building is solely utilised for office or warehouse purposes. This approach will provide a spectrum of potential energy consumption patterns within the hall, which can then be compared to the aforementioned probable user profile.

4.2.1 Office user

If all 585 m^2 area were to be utilised as offices, without consideration of potential issues arising from an inadequate lighting due to the limited window area, the heat demand would not differ significantly from the standard case previously outlined. The distinction lies in the fact that in this instance, the entire building is maintained at a setpoint temperature of 20°C, which results in a 12% increase in peak power.

The calculation procedure is identical to that described in chapter 4.1.

This yields a peak power output of 76,2 kW.

Room	A_{floor} $[m^2]$	h $[m]$	V $[m^3]$	A_W $[m^2]$	S_{wall} $[m^2]$ out	S_{wall} $[m^2]$ unheat	H_T $[W/K]$ out	H_T $[W/K]$ unheat	H_V $[W/K]$	q_{peak} $[kW]$	E_{demand} $[kWh]$
Office	585	6	3510.0	39.5	140.5	468.0	1486.3	1184.8	589.1	76.2	105217.0

Table 4.24 – heat demand office user only

The most significant distinction is the summer load. Given that the entire room is utilised as an office, the presence of electrical equipment exerts a considerable impact on the overall load, which must be taken into account when assessing the room's capacity. It is recommended (UNI EN 16798) that an average room occupancy rate of 10.8 W/m² be assumed, which corresponds to 8.5 workstations per 100 m². It is evident that the summer heat load is considerably higher in this instance.

The second factor contributing to the observed increase in peak power for cooling load is the maintenance of a constant temperature of 26°C. The section of the warehouse previously designated with a higher setpoint is no longer subject to that designation.

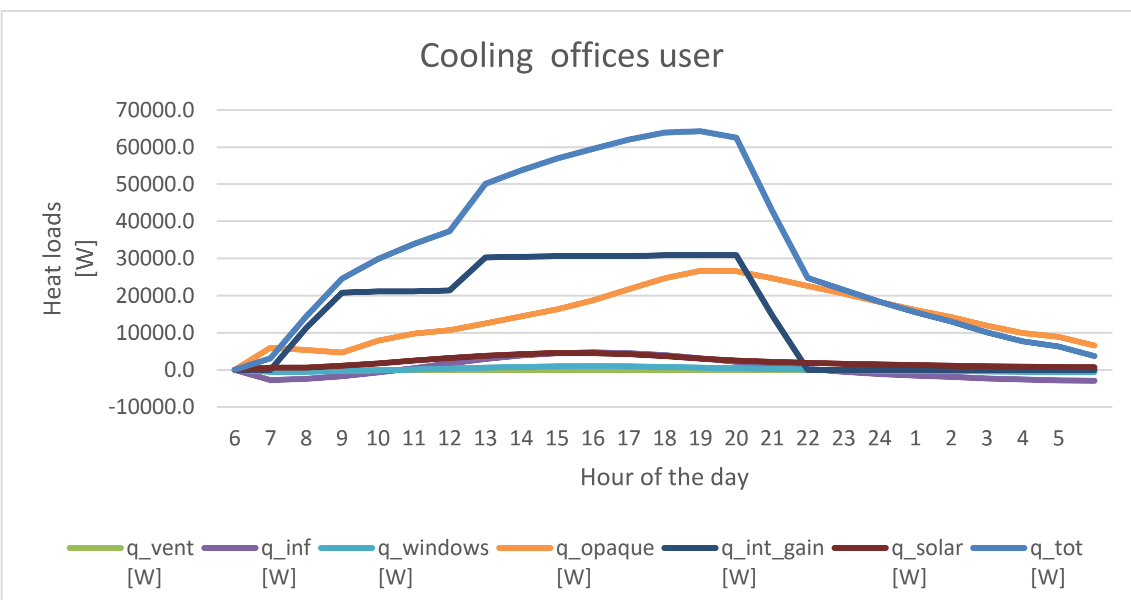

Figure 4.5 – cooling profile of offices user

It is evident that the influence of internal heat gains is the most significant in this instance. To facilitate an understanding of the relative magnitude of each contribution, the following table is provided.

Hour	q_v [W]	q_{inf} [W]	q_w [W]	q_{op} [W]	q_{ig} [W]	q_s [W]	q_{tot} [W]	Peak power for cooling [W]
6	0.0	-2792.3	-569.2	5921.2	0.0	563.5	3123.2	
7	0.0	-2409.4	-491.1	5329.1	11336.0	634.8	14399.4	
8	0.0	-1720.2	-350.6	4673.6	20813.0	1127.9	24543.7	
9	0.0	-724.6	-147.7	7841.2	21164.0	1736.0	29868.9	
10	0.0	424.1	86.5	9772.5	21164.0	2478.5	33925.6	
11	0.0	1726.0	351.8	10725.4	21398.0	3144.1	37345.4	
12	0.0	2951.4	601.6	12491.0	30243.2	3803.3	50090.5	
13	0.0	3870.4	789.0	14394.4	30453.8	4251.3	53758.8	
14	0.0	4483.0	913.8	16323.6	30664.4	4558.6	56943.5	
15	0.0	4712.8	960.7	18691.2	30664.4	4501.1	59530.1	
16	0.0	4483.0	913.8	21716.6	30664.4	4238.9	62016.7	
17	0.0	3946.9	804.6	24660.8	30875.0	3682.3	63969.6	
18	0.0	3104.5	632.8	26683.3	30875.0	2990.5	64286.1	64286.1
19	0.0	2109.0	429.9	26588.1	30875.0	2510.2	62512.2	
20	0.0	1113.4	227.0	24700.8	14658.8	2183.6	42883.5	
21	0.0	271.0	55.2	22593.1	0.0	1857.0	24776.4	
22	0.0	-494.8	-100.9	20523.0	0.0	1613.7	21541.0	
23	0.0	-1107.5	-225.8	18266.2	0.0	1453.6	18386.6	
24	0.0	-1567.0	-319.4	16106.7	0.0	1300.0	15520.2	
1	0.0	-1949.9	-397.5	14246.3	0.0	1133.4	13032.3	
2	0.0	-2332.8	-475.5	11927.5	0.0	979.8	10099.0	
3	0.0	-2639.1	-538.0	9914.7	0.0	890.1	7627.7	
4	0.0	-2868.9	-584.8	8921.1	0.0	806.8	6274.3	
5	0.0	-2945.5	-600.4	6577.5	0.0	646.8	3678.3	

Table 4.25 – cooling demand office user only

It is evident that despite alterations in utilisation, the peak remains at the same time, as the structure remains unaltered.

The cooling peak load is thus found to be around 40% higher than in the standard case.

4.2.2 Warehouse user

In this case, the entire 585 m^2 area is designated for warehouse. Once more, the discrepancy in winter peak power remains relatively low. The 8% divergency can be attributed to the temperature setpoint, which is established at 18°C.

Room	A_{floor} $[m^2]$	h $[m]$	V $[m^3]$	A_W $[m^2]$	S_{wall} $[m^2]$		H_T $[W/K]$		H_V $[W/K]$	q_{peak} $[kW]$	E_{demand} $[kWh]$
					out	*unheat*	*out*	*unheat*			
Wareh.	585	6	3510.0	39.5	140.5	468.0	1486.3	1184.8	589.1	63.1	105217.0

Table 4.26 – heat demand office user only

Similarly, the summer load is not significantly divergent from the standard case. The primary factor contributing to this discrepancy is the higher temperature setpoint, which is established at 28°C.

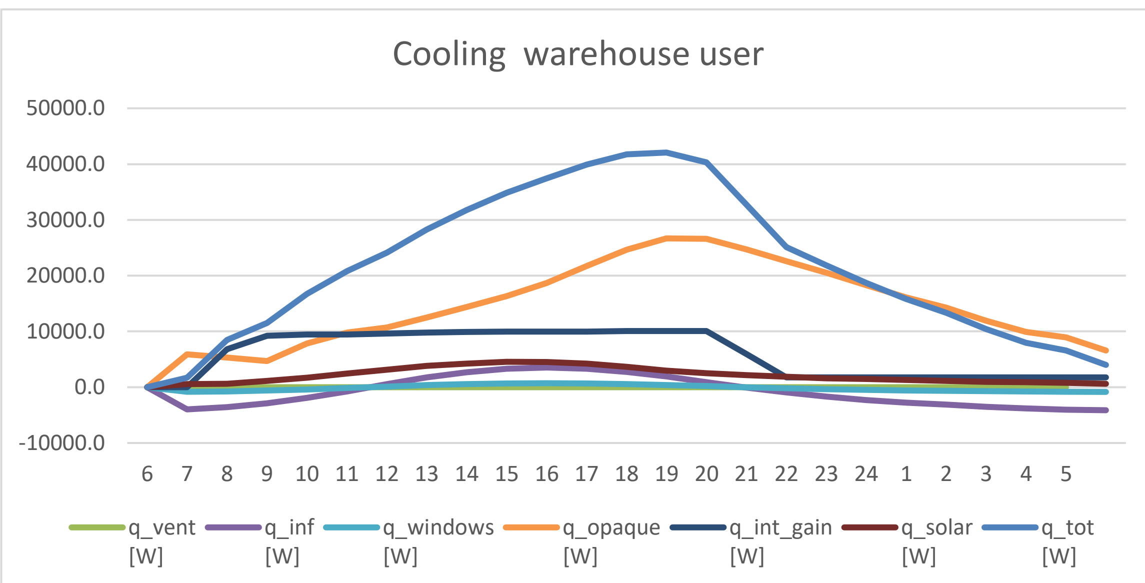

Figure 4.6 – cooling profile for warehouse user

The primary contribution is made by the opaque wall. In this instance, the space is to be regarded as a storage room, and thus only internal gains associated with people and light are considered.

The following table illustrates the numerical contribution of each term, expressed in hourly terms.

Hour	q_v *[W]*	q_{inf} *[W]*	q_w *[W]*	q_{op} *[W]*	q_{ig} *[W]*	q_s *[W]*	q_{tot} *[W]*	*Peak power for cooling [W]*
6	0.0	-3970.5	-809.4	5921.2	0.0	563.5	1704.8	42069.0
7	0.0	-3587.6	-731.3	5329.1	6829.5	634.8	8474.5	
8	0.0	-2898.3	-590.8	4673.6	9198.8	1127.9	11511.1	
9	0.0	-1902.8	-387.9	7841.2	9462.0	1736.0	16748.6	
10	0.0	-754.0	-153.7	9772.5	9462.0	2478.5	20805.2	
11	0.0	547.9	111.7	10725.4	9637.5	3144.1	24166.5	
12	0.0	1773.2	361.5	12491.0	9813.0	3803.3	28241.9	
13	0.0	2692.2	548.8	14394.4	9900.8	4251.3	31787.4	
14	0.0	3304.8	673.7	16323.6	9988.5	4558.6	34849.2	
15	0.0	3534.6	720.5	18691.2	9988.5	4501.1	37435.8	
16	0.0	3304.8	673.7	21716.6	9988.5	4238.9	39922.5	
17	0.0	2768.7	564.4	24660.8	10076.3	3682.3	41752.5	
18	0.0	1926.3	392.7	26683.3	10076.3	2990.5	42069.0	
19	0.0	930.8	189.7	26588.1	10076.3	2510.2	40295.1	
20	0.0	-64.8	-13.2	24700.8	5952.0	2183.6	32758.4	
21	0.0	-907.2	-184.9	22593.1	1740.0	1857.0	25098.0	
22	0.0	-1673.0	-341.0	20523.0	1740.0	1613.7	21862.6	
23	0.0	-2285.7	-465.9	18266.2	1740.0	1453.6	18708.2	
24	0.0	-2745.2	-559.6	16106.7	1740.0	1300.0	15841.8	
1	0.0	-3128.1	-637.7	14246.3	1740.0	1133.4	13354.0	
2	0.0	-3511.0	-715.7	11927.5	1740.0	979.8	10420.6	
3	0.0	-3817.3	-778.1	9914.7	1740.0	890.1	7949.3	
4	0.0	-4047.1	-825.0	8921.1	1740.0	806.8	6595.9	
5	0.0	-4123.7	-840.6	6577.5	1740.0	646.8	4000.0	

Table 4.27 – cooling demand warehouse user only

As observed in previous instances, the peak occurs at 6 p.m.

4.2.3 Synthesis of main results for thermal energy demand

The aforementioned considerations regarding the requisite peak power in both summer and winter are presented in tabular form below:

Case	*Heat peak power [kW]*	*Cool peak power [kW]*
Standard	67.9	44.8
Only offices	76.2	64.3
Only warehouse	63.1	42.1

Table 4.28 – resume of the peak power

In any case, the most probable user profile is that described in the layout presented in chapter 4.1. Accordingly, all calculations pertaining to the sizing and choice of plant terminals are based on this configuration.

4.3 Terminal options for connection to the district heating system and criteria for selecting the most suitable solutions

In order to identify potential solutions for connecting terminals to the district heating and cooling network, it is necessary to focus the search on water-based devices. It is therefore evident that solutions utilising refrigerants, such as heat pumps, are not viable options within the context of this discussion.

A review of the solutions employed by other users of district heating and cooling in the neighbourhood, coupled with an examination of potential avenues in the existing literature, has led to the identification of the following systems as viable options:

- Air heaters and coolers, and hydronic fan coils
- Air Handling Unit (AHU)

The implementation of the aforementioned solutions is contingent upon the prior transfer of heat to (or from) the technologies in question, with the heat transfer fluid originating from the central plant. Accordingly, the initial step is to determine the dimensions of the plate heat exchanger that will receive the fluid from the power plant on one side and the fluid from the user on the other.

The substation will remain identical regardless of the selected terminal, given that the thermal power to be delivered is the same in all cases.

The company “*Thermowave*” was selected for the plate heat exchanger on the basis of two key factors: firstly, the high quality of its products; and secondly, the transparency of the data. Furthermore, the company has a documented history of supplying Telezip with heat exchangers in other buildings that have already been served.
In particular, the model designated “*thermolineECO*”[3] was selected in order to guarantee a compact technology with minimal requirements for maintenance. As indicated in the product data sheet, this heat exchanger is suitable for district heating and district cooling applications. The capacity of the plate heat exchanger in question ranges between 10 and 1.000 kW. This is due to the modular solution, which allows for a wide range of operational flexibility.
As a rough estimate, the cost of a substation of this size is approximately €10.500, based on previous quotations for the materials and €1.000 for the installation costs.

Furthermore, the installation of an air curtain over the entrance door of the shop is essential to divide the internal space from the external environment as much as possible. To illustrate, one might consider the product of “Frico: Cordilla commercial air curtains”[4]. From the economic point of view the estimated cost is approximately €1.000.

It is thus evident that, in the case of both the systems presented subsequently, consideration must be given to the fact that €12.500 is to be allocated towards the installation of the air curtain and the substation.

In order to select the optimal system, it is essential to consider multiple criteria, including those pertaining to occupants’ comfort, process heating, space heating, cooling and ventilation. These can be evaluated through the following parameters [19]:

- Temperature
- Humidity
- Air motion
- Air/water velocity
- Indoor air quality

[3] https://www.thermowave.eu/wp-content/uploads/2024/04/thermowave_thermolineEco_Info_EN.pdf
[4] https://www.frico.net/en/products/air-curtains/commercial-air-curtains/customised/cordilla

- ACH
- Acoustics and vibration
- Mold and mildew prevention
- Capacities (existing, proposed, and future expansion)
- Spatial requirements (present and future)
- Environmental health and safety design
- Initial cost
- Energy consumption costs
- Operator labour costs
- Serviceability
- Reliability
- Controllability
- LCA

The choice of terminal typology will be determined by a comprehensive assessment of all relevant technical and economic factors, employing a systematic approach to ensure the optimal decision-making process.

The energy demand calculation and all other assessments in the technical area were based on the assumption that the entire building would be heated. Accordingly, even in the event of future alterations to the building's intended use, for instance, should it become wholly a commercial establishment, the system must be capable of meeting the new energy requirements. It is thus imperative to subdivide each category of plant into a minimum of three distinct zones: shop, office and warehouse.

4.4 Technical evaluation of different terminal solutions

This section presents a detailed examination of the two potential solutions for the terminals.

4.4.1 Air heaters and fan-coil

They are heating bodies that emit heat through forced convection. They essentially consist of:

- Finned heat exchange coil
- Helical fan
- Containment casing

Based on the direction of their air jets, unit heaters can be divided into two categories:

- Horizontal projection

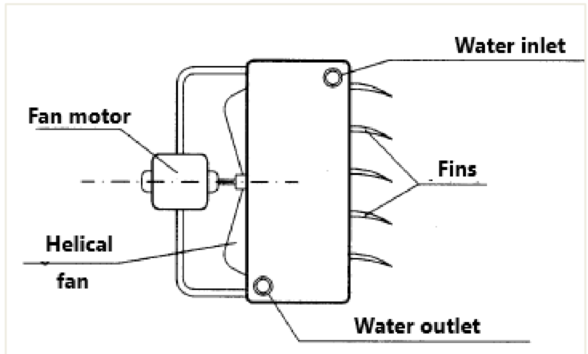

Figure 4.7 – horizontal projection type [20]

- Vertical projection

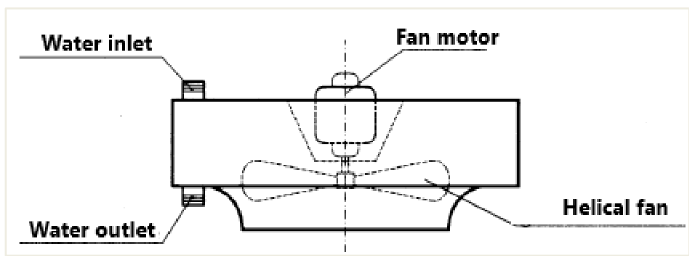

Figure 4.8 – vertical projection type [20]

The fan coils are similar, with a slightly different configuration, which can be schematised as follows:

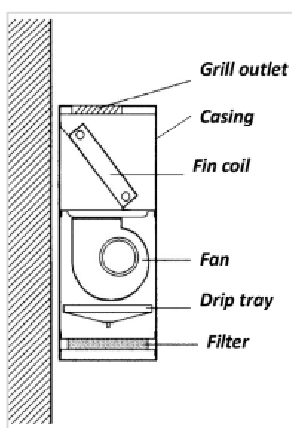

Figure 4.9 – general scheme of fan coil

The appropriate selection of these devices necessitates the consideration of the following factors:

- Type and location of the terminals
- Air outlet temperature
- Sound level.

It is essential to select the type and location of these terminals in such a way as to avoid the formation of areas with either excessively low or high temperatures.

To achieve this objective, it is recommended that the following general guidelines be followed:

- Install at least two unit heaters in each room
- Check that the sum of the fans' hourly capacities is between 3 and 4 times the volume of the room to be heated
- Avoid interference of the air jets with columns, machines or other obstacles

The temperature of the air leaving the devices should be between:

- 40°C and 45°C for horizontal terminals
- 30°C and 45°C for vertical type

These values represent a good compromise between two different requirements:

- Preventing air currents, generated by the terminals themselves, from causing local discomfort
- Preventing the formation of strong air stratification

The temperature of the air leaving the unit heaters is usually specified in the manufacturer's technical specifications.

The noise emitted by terminals must not exceed the permissible sound level in the room, as indicated in the technical specifications. The value in question is contingent upon the intended use of the room and can be selected on the basis of values recommended in technical literature (UNI 8199).

For shops and offices, the limit value is around 40 ÷ 45 dB (A).

It is recommended that the following operations and checks be carried out to maintain this type of terminal in optimal condition:

- Clean the heat exchanger coil with a brush or compressed air jets. In some cases, particularly where there are grease deposits, it may be necessary to wash the fins with soap and water, taking care not to wet the electrical equipment. The frequency of these operations depends on the degree of cleanliness of the environment and the type of battery
- Check the tightness of the nuts securing the motor and housing at least once a year, or if irregular noises and vibrations are felt during operation
- Check the power consumption if the motor protector trips frequently

In consideration of the functionality and aesthetic appeal of the rooms, the selected heating system comprises unit heaters in the warehouse, while fan coils are employed in the shop and office. In order to guarantee uniformity in the installation of the terminals, the same manufacturer will be selected for both locations.

The Riello catalogues will be utilised for the purpose of selecting the terminals.
In particular: the "ACUF[5]" unit heaters, as specified in the technical specifications, are designed for use in industrial, commercial and sports environments for the purposes of heating and cooling. The terminals consist of 2 or 3 rows of copper coils, depending on the specific application.
With regard to fan coils in office and shop, the technical sheet indicates that the "Design inverter"[6], allows for continuous operation within the range between 0% and 100%. This feature enables a corresponding adjustment of the heating and cooling capacity, which is a notable advantage. This product facilitates the rapid heating and cooling of any environment, thereby providing a solution for a variety of applications. Two distinct mapping configurations are available. The "performance" mapping approach is designed for implementation in commercial settings where high efficiency and effectiveness are of paramount importance. This particular option will be taken into consideration when formulating the project specifications that follow.

It is of the highest importance to take both winter and summer demand into account when undertaking sizing calculations. The selection of the terminal will then be based on the least favourable configuration. The following calculation procedure will first determine the appropriate size for the fan coils in both summer and winter conditions, and then apply the same process to

[5] https://www.riello.it/catalogo/condizionamento?action=download&id=88BBBXXERF-c9b9d0b64bf739d462ff0cb1fb16b3af
[6] https://www.riello.it/catalogo/condizionamento?action=download&id=82AABXIWRF-2762319f5f41c547117fee7a4a849d0a

the fan coils. The final stage will entail making the terminal and numbering choices in each room based on the most unfavourable configuration, i.e. the one that requires the highest number of units.

In regard to the selection of unit heaters, the manufacturer presents 3 potential water temperature ranges for heating efficiency: 90°C-70°C, 85°C-70°C and 50°C-40°C. Given that the water temperature leaves the central plant at approximately 70°C during the winter season, the temperature range of 50°C-40°C was selected for the design calculations.
In consideration of the summer season, the manufacturer presents two distinct temperature ranges: 7°C-12°C and 11°C-15°C. Given that the water temperature at the central plant is approximately at 7°C, the temperature range of 11°C-15°C was employed as the design value. Furthermore, the manufacturer provides specifications for disparate air temperatures. Among the various options, the intake temperature of 30°C was selected for the summer season and 20°C for the winter season.

Model	*Riello ACUF 33*	
	Heating	*Cooling*
Inlet air temperature	20°C	30°C
Outlet air temperature	34°C	19.6°C
Inlet water temperature	50°C	11°C
Outlet water temperature	40°C	15°C
Potentiality	9.1 kW	8.5 kW
Air flow rate	1900 m^3/h	1900 m^3/h
Water flow rate	794 l/h	1823 l/h
Cost	486 €/terminal	

Table 4.29 – characteristic of air heaters [21]

Once chosen the model, the number of units can be evaluated:

$$n = \frac{Q_{req}}{Q_{terminal}} \tag{28}$$

So, the overall power available is:

$$Q_{TOT,win} = n * Q_{unit} \tag{29}$$

From the catalogue the air penetration values in the room can be taken:

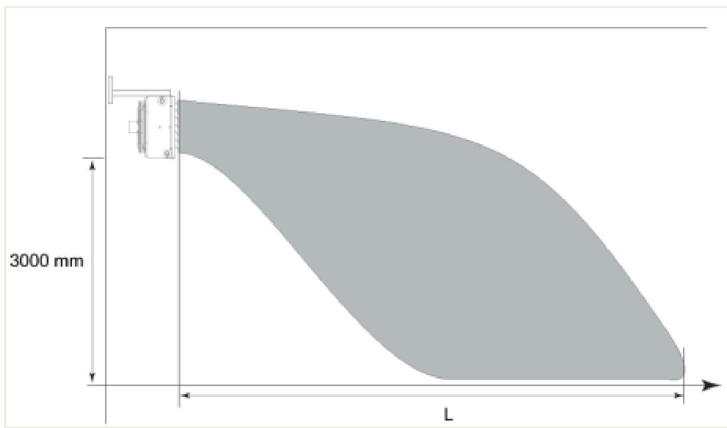

Figure 4.10 – air penetration of air heaters [21]

For model ACUF 33, L = 17 m with fins inclined at 70°.

In consideration of the dimensions of the warehouse, specifically 19 x 15 meters, it can be concluded that the aforementioned model can be installed on the longer side (figure 4.11).

With regard to the shop and office, the selected fan coil units are resumed in the following table:

Model	*Riello Design inverter 46B*	
	Heating	*Cooling*
Inlet air temperature	20°C	30°C
Inlet water temperature	50°C	7°C
Potentiality	5.9 kW	4.56 kW
Air flow rate	787 m^3/h	787 m^3/h
Water flow rate	784 l/h	784 l/h
Cost	664 €/terminal	

Table 4.30 – characteristic of fan coils [22]

As with unit heaters, the manufacturer provides a range of water temperatures for fan coils, which are designated to accommodate varying requirements. In the process of dimensioning the terminals, the temperature range 50°C-40°C is taken into consideration. With regard to the cooling performance, only the data for a water temperature of 7°C-12°C is available. In this case, the temperature of the intake air is assumed to be 27°C during the summer months and 20°C during the winter.

Based on an initial calculations, the most unfavourable season is winter. Accordingly, the number of terminals will be selected in accordance with the specific requirements of the latter.

The subdivision of the units in the rooms are as follow:

Room	*Load*		N_{units}	*Real power output*		*Air treated* $[m^3/h]$	*Room volume* $[m^3]$	*ACR [1/h]*
	Winter [kW]	*Summer [kW]*		*Winter [kW]*	*Summer [kW]*			
Shop	19.3	14.6	4	23.6	18.2	3148	912	3.4
Office	16.3	11.5	3	17.7	13.7	2361	798	3.0
Warehouse	32.4	18.4	4	36.4	34.0	7600	1800	4.2
TOTAL	67.9	44.5	77.7	65.9	6592	3510	-	-

Table 4.31 – the most disadvantage configuration is the winter one

It is evident that, given the number of terminals required during the winter season (11), the potential available in the summer case is that of the 11 units.

In table 4.31, the term “real power output” is understood to represent a scenario in which all terminals are operating under nominal conditions. It is evident that, particularly during the summer months, the peak power demand is significantly higher than the required level, resulting in the terminals functioning under partial load conditions.

In consideration of the ACR in question, it can be observed that the value in each room falls between 3 and 4, thus satisfying the constraints initially outlined in the paragraph.

The scheme of the possible configuration of the plant is reported in the figure below:

Figure 4.11 – possible location of the units in the building

The current discussion does not provide a detailed description of the proposed methodology for connecting the substation to the terminals. In general terms, the following section presents some of the potential system solutions.

From the substation, the heat transfer fluid could be conveyed to the supply manifold, which then feeds one-by -one the single terminals. It is possible that each terminal be supplied and returned via a dedicated pipework and manifold.
An alternatively approach would be to implement the Tichelmann loop scheme.
In any case the 3 zones of the building must be divided in order to facilitate independent control of each zone. The user is thus empowered to determine the specific areas of the building to which water is to be distributed.
In either case, since the water flow rate of the individual terminal is provided by the manufacturer, once the diameter has been selected, the water velocity must be calculated. The optimal velocity is that which falls between 0.5 m/s and 1.5 m/s, representing a compromise between efficient heat transfer and the avoidance of distribution problems or the generation of noise. A higher velocity would result in an increased the pressure drop and a decreased the life of the system due to increased stress in the components.

As previously stated, in order to derive the greatest benefit from this solution, it is essential to integrate the system with destratifiers. Such devices facilitate the recovery of ambient heat during the winter months, when this heat tends to move upwards. Furthermore, this enhances the efficiency of the system. The fan directs the heat to the floor, thereby reducing heat loss and

preventing the heat transfer of heat from the ceiling to the outside. This results in a more efficient heating of the building, which in turn leads to a reduction in energy consumption.
In this case the destratification units are sized according to the specification provided by ROBUR's online software [23]. The software requires the dimensions of the room and the height at which the destratifier would be installed as inputs. The "Air Tech 520" model was selected from the available solutions. The number of units proposed is one per room in barycentric position. The manufacturer guarantees savings of approximately 20% for the user.

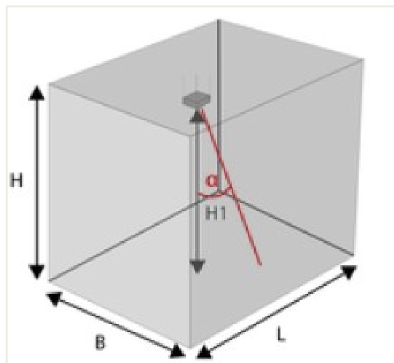

Figure 4.12 – Black parameters represent the input data required by the software. Red parameter is an output parameter to set the fins.

Model	*ROBUR – Air tech 520*
Fan	3 velocities
Electric power [W]	280
Air flow rate [m^3/h]	5200 | 4200 | 2800
α [°]	45
Sound level [dB (A)]	65.7 | 80.0
T_{MAX} [°C]	60
Weight [kg]	13.9

Table 4.32 – characteristics of destratifier

In order to gain insight into the economic implications of this technology, it is essential to have an estimation of the order of magnitude of the cost. This will enable a comprehensive understanding of the economic implications of this type of installation.

Model	*Acuf 33 Riello (4 units)*	*Design inverter 46B (7 units)*
Material costs	486,00 €/unit [24]	664,00 €/unit
Thermohydraulic materials	773,80 €/unit	654,01 €/unit
Total each unit	1.259,80 €/unit	1.318,01 €/unit
Labour cost	€ 1.800	
Destratifiers	€ 1.500 (€ 500 each one)	
Substation	€ 11.500	
Air curtain	€ 1.000	
TOTAL	€ 30.065,27	

Table 4.33 – overall costs of air heaters and fan-coils

The costs of materials were derived directly from the Riello catalogue for the plant terminals and from the ROBUR catalogue for the destratifier.
The costs associated to thermohydraulic materials and labour costs were taken from the Veneto price list [24].

4.4.2 Air Handling Unit (AHU)

A full-air system consists of an air handling unit, which is responsible for supplying controlled air to the rooms via a network of ducts. Such systems can be classified into two principal categories: those that utilise full outdoor air and those that employ recirculation.

The heat load of the room can be balanced by supplying air at a temperature that is higher than that of the room during the winter season and lower during the summer. As the same system controls the temperature in both winter and summer, the design conditions are determined by the most severe conditions, that is to say, those with the highest load. In full air systems, all parameters (temperature, relative humidity and air quality) are typically regulated by the AHU.

In order to guarantee the most effective treatment of the total air flow, it is essential to select the most appropriate parameters and to ensure that all the components, including fans, filters, preheating and reheating coils, humidification section, cooling and dehumidification coils, are accurately sized.

The main sections that permit the intake of air at a specific temperature and humidity are as follows:

- *Preheating coil*: during the winter months, its primary function is to preheat the external air. These are manufactured with copper tubes and aluminium finned packs
- *Cooling and dehumidification coil*: during the summer months, the coil is responsible for the removal of both sensible heat and latent heat from the treated air. However, it should be noted that not all the air that passes through the battery is treated. Therefore, the bypass efficiency of the coil is introduced, with a potential range of 3% to 30%. The material used in the construction of these batteries is usually copper for both the tubes and the finning (which must be tinned in a wet bath), to prevent corrosion phenomena due to battery's wet nature, resulting from the condensation of the water vapor present in the air.
- *Humidification section:* it is responsible for supplying the requisite amount of water vapor to maintain the desired relative humidity within the rooms. Depending on the intended application, this section can be constructed in a variety of ways
- *Reheat coil:* it is employed for the purpose of adjusting the design temperature and ensuring that the input temperature is maintained at the desired level both in heating and cooling mode

The main advantages of this type of system are as follows:

- ✓ The air handling unit containing the equipment is located in the area reserved for it, and therefore there is the possibility of being able to carry out control and maintenance operations with ease
- ✓ Control the propagation of vibrations and noise very well

- ✓ With these systems, it is possible to carry out accurate filtration of all the air fed into the room
- ✓ Simultaneous control of temperature and humidity in the room
- ✓ Possibility of heat recovery from the exhaust air
- ✓ Intake of air with special care to avoid possible local discomfort (e.g. draft risk)
- ✓ Do not create problems of furnishing nor use of all available surface area

On the other hand, there are also some disadvantages:

- x High costs of installations
- x Need of regular maintenance
- x High footprint of distribution circuits can give rise to structural problems
- x Criticality in balancing and calibration of circuits

The sizing procedure for an AHU starts with the analysis of the loads required in summer and winter and the flow rate to maintain a certain degree of air quality. The heating and cooling requirements are already calculated in table 4.11 and table 4.23. For what concern the indoor environmental quality, the choice for this building was based on the specifications outlined in standard UNI 10339 and standard UNI/TS 11300, with the assumption of a number of occupants per room equivalent to that utilised in the assessment of the internal gains in the carrier method in chapter 4.1.2.

Room	G_{ppl} $[(m^3/h)/ppl]$	f_{ppl} $[ppl/m^2]$	A $[m^2]$	*n. people* *[ppl]*
SHOP	42	0.10	152	19
OFFICE	42	0.12	133	16
WAREHOUSE	33	0.05	300	15

Table 4.34 – room occupancy rate

So, the air flow rate to fulfil the IAQ requirements is:

$$G_{IAQ} = Q_{ppl} * f_{ppl} * A \tag{30}$$

The following relationship can be used to facilitate the conversion of the heating and cooling load into air flow rate:

$$q_i = \rho_{air} * G_{load} * c_{p,air}\left(\left|\theta_{amb} - \theta_{sup}\right|\right) \tag{31}$$

In light of the established values for peak power associated with heating and cooling, the remaining variable for determination is that of the air flow rate (G_{load}). This can be calculated by inverting the equation 31. In this case, the air flow rate (G_{load}) does not necessarily refer to fresh air, but it could be partially recirculated. This process results in a reduction in energy consumption, as the external air is combined with the air within the room.

Season	θ_{room} *[°C]*	θ_{supply} *[°C]*	$\Delta\theta$ *[°C]*
WINTER (shop and office)	20	35	15
WINTER (warehouse)	18	35	17
SUMMER (shop and office)	26	16	10
SUMMER (warehouse)	28	16	12

Table 4.35 – boundary conditions

In order to maintain the desired temperature within the design room, which is set at 20°C during the winter months and 26°C during the summer months (with the exception of the warehouse, where the temperature is set at 18°C during the winter and 28°C during the summer), it is assumed that the air supply to the room will be 35°C during the winter and 16°C during the summer. This is done in order to avoid any potential issues of local discomfort.

The air flow associated with the heat load required can be resumed in the following table:

Room	$G_{heating}$ *[m³/h]*	$G_{cooling}$ *[m³/h]*	G_{IAQ} *[m³/h]*
SHOP	3780	4292	638.4
OFFICE	3187	3365	670.32
WAREHOUSE	5593	4511	495

Table 4.36 – air flow rate for different purposes

Subsequently, the maximum of the three calculated air flow rates must be considered in the design phase:

$$G_{des} = \max\left(G_{heating}; G_{cooling}; G_{IAQ}\right) \tag{32}$$

The minimum air requirement for the air quality is calculated using the equation (33). To simplify the sizing process, the maximum B_j value was assumed for all the environments. This approach entailed utilising the room with the highest percentage of fresh air as the reference point for all other rooms.

In regard to equations, the following can be stated:

$$B_i = \frac{G_{IAQ,i}}{G_{des,i}} \tag{33}$$

$$B_{tot} = max(B_i) \tag{34}$$

Resuming in a table all the calculations described above:

Room	G_{des} *[m³/h]*	G_{des_tot} *[m³/h]*	B_j *[-]*	B_{tot} *[-]*
SHOP	4292		0.15	
OFFICE	3365	13250	0.20	0.20
WAREHOUSE	5593		0.09	

Table 4.37 – evaluation of the minimum share of fresh air

This indicates that the proportion of fresh air in the total supply must be a minimum of 20%.
In light of these constraints, the calculation of the fresh and recirculation air used in the design phase can be evaluated as follows:

Room	*Fresh air [m³/h]*	*Recirc. air [m³/h]*	*Supply air [m³/h]*
SHOP	855	3437	4292
OFFICE	670	2695	3365
WAREHOUSE	1114	4479	5593
TOTAL	2639	10610	13250

Table 4.38 – main results for each room with 20% of fresh air

Given the design flow rates, it is now possible to assume a potential system structure for distribution within the building.

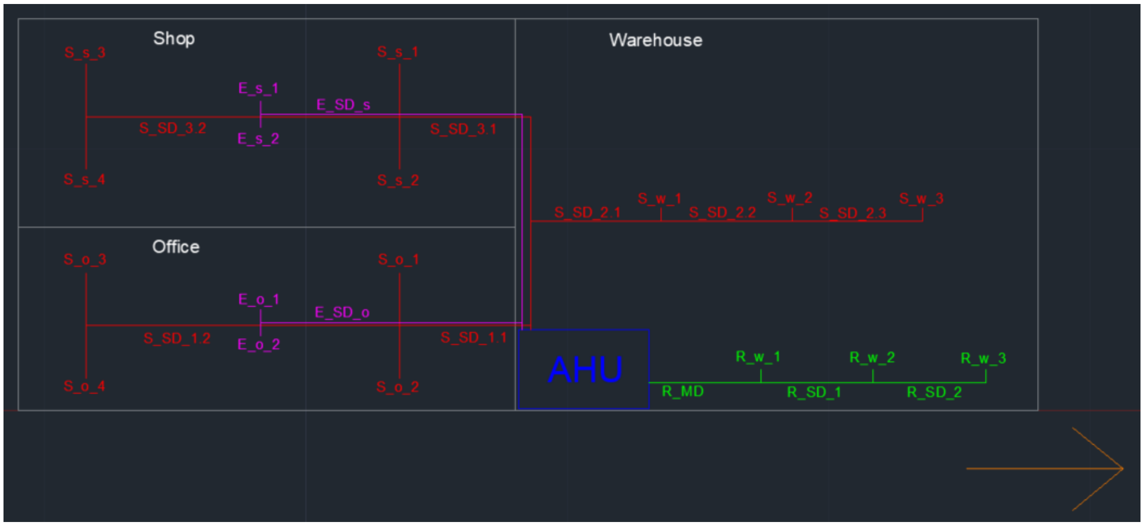

Figure 4.13 – sketch of the air distribution system

Caption:

- *Red line*: supply (S)
- *Green line*: recirculation (R)
- *Purple line*: exhaust (E)
- *MD*: main duct
- *SD*: secondary duct
- *s*: shop
- *w*: warehouse
- *o*: office

For the sake of simplicity, the curves are represented in the figure as 90° curves. However, in practice, it is preferable to have a path that is as straight as possible in order to avoid a high pressure drop. Additionally, the phenomenon of obstruction that arises when pipes cross was not considered.

The most crucial aspect of determining the optimal location for the AHU is the fact that the pipe connections originating from the central plant and passing through the network are situated in this precise position. Therefore, no further work is required to connect the substation to the network. Moreover, figure 4.13 illustrates the strategic positioning of the air handling unit, which is situated in a central location within the building. This location is approximately aligned with the building's barycentric area, which serves to minimise the operational constraints and ensure minimal noise disruption.

To size the entire system, it is necessary to consider the entire volume of the building as single entity. It is evident that the exhaust and recirculation lines do not reach all the rooms. This is due to the fact that the exhaust air is extracted from the contaminated room (shop and office), while the recirculation air is taken from the room in which the air is “neutral” (warehouse).

Given that the supply and exhaust are located in the same rooms, it was deemed prudent to establish a suitable distance between them in order to mitigate the risk of short-circuit flow.

The dimensions of the branches were selected in accordance with the velocity constraints [16] in table 4.39. The choice of an appropriate air speed for the ducts represents a compromise between two main considerations. On one hand, there is the necessity to prevent the entrainment of condensed water droplets on the surface of the coil. On the other hand, there is the need to ensure that the pressure drops do not exceed the levels that are responsible for the power absorbed by the fans.

The following table presents the optimal velocities that must be maintained for each component of the plant:

Tube	v_{max} *[m/s]*	v_{limit} *[m/s]*
Main ducts	8	6
Secondary	6	5
Air inlet	4	3
Air outlet	3	2

Table 4.39 – velocity constraints

The maximum air velocities prescribed by the standard are indicated on the left. The velocities illustrated on the right have been selected to guarantee a safety margin. Conversely, the velocity must be sufficiently high to enable the air to flow out from the terminal units.

The size of the tubes is selected on the basis of the closest commercially available diameter currently on the market.
In order to evaluate the diameters, the following procedure was employed.

The preliminary stage of the process entails the calculation of the flow area and the corresponding diameter:

$$A = \frac{\dot{G}_{air}}{3600 * v} \tag{35}$$

$$D_{calc} = \sqrt{\frac{4 * A}{\pi}} \tag{36}$$

Subsequently, the nearest commercial diameter is chosen, and the actual area and velocity can be calculated.
This procedure was then applied to all branches.

At this point, the tool[7] was employed for the assessment of the continuous and localised pressure drop for each segment. The tool is only capable of sizing the secondary ducts and final branches; for the main ducts, only the continuous pressure losses were considered to maintain simplicity. In essence, the tool is able to assess the continuous and localised pressure losses through the following relations:

- *Continuous pressure drops* due to fluid-wall interactions:

$$\Delta p_{f,c} = f * \frac{\rho * v^2}{2} * \frac{L}{D} \tag{37}$$

Where the friction factor (f) depends on the Reynolds number, duct diameter and roughness.

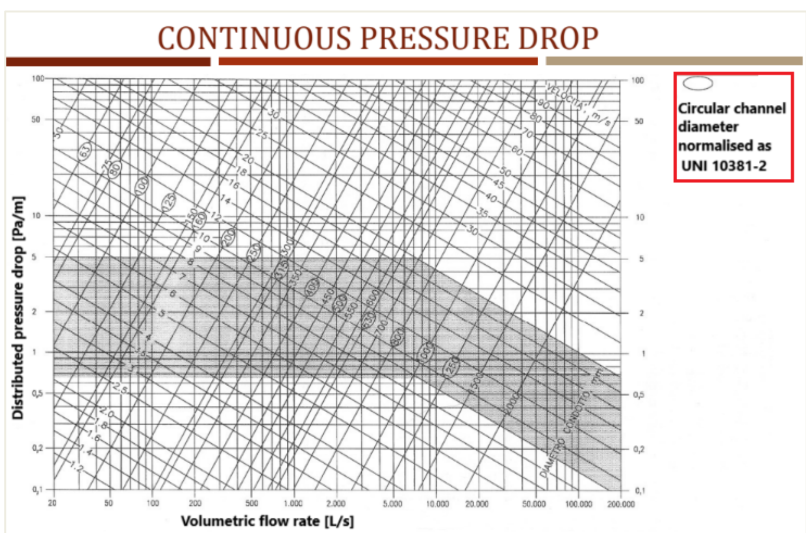

Figure 4.14 –pressure drop as a function of the flow rate and diameter

[7] Excel tool provided by prof M. De Carli during the course of HVAC a.y. 2023-2024.

The material assumed for the tubes is aluminium sheet.

- *Localised pressure drops* whenever flow meets discontinuity in its path:

$$\Delta p_{f,l} = \xi * \frac{\rho * v^2}{2} \tag{38}$$

The localised pressure drop coefficient (ξ) is dependent upon the specific obstacle present within the circuit. The pipe elements that induce localised pressure losses have been taken into account in accordance with the representation depicted in figure 4.13. The values are presented in a tabular format in accordance with the standard for the most common circuit elements.

The data necessary for utilisation of the aforementioned tool are listed here:

- Number of secondary branches
- Number of final branches for each secondary branch
- Material of the tubes
- Commercial diameter of each section
- Length of each section
- Flow rate of the final branches
- Components that introduce pressure drop in each section

In addition to the calculated continuous and localised pressure drops, the tool considers by default 420 Pa of pressure losses due to the components present in AHU that interact with the air path including filters, dampers, heat exchangers, and so forth.

A summary of the principal outcomes obtained from the three circuits is presented in the tables below. The sections under consideration are identified in accordance with the illustration in figure 4.13.

SUPPLY CIRCUIT											
Section	*G_{air} [m³/h]*	*A [m²]*	*D_{calc} [m]*	*D_{comm} [mm]*	*A_{real} [m²]*	*v_{real} [m/s]*	*Δp_{loc} [Pa]*	*Δp_{cont} [Pa/m]*	*L [m]*	*$\Delta p_{cont,tot}$ [Pa]*	*Δp_{tot} [Pa]*
S_MD	13250	0.61	0.88	800	0.50	7.32	0.00	0.40	0.20	0.08	0.08
S_SD_1.1	5600	0.31	0.63	630	0.31	4.99	19.44	0.29	5.00	1.45	20.89
S_o_1	1400	0.13	0.41	400	0.13	3.09	13.23	0.40	2.00	0.80	14.03
S_o_2	1400	0.13	0.41	400	0.13	3.09	13.23	0.40	2.00	0.80	14.03
S_SD_1.2	2800	0.16	0.45	630	0.31	2.50	0.00	0.08	12.00	0.96	0.96
S_o_3	1400	0.13	0.41	400	0.13	3.09	13.80	0.40	2.00	0.80	14.60
S_o_4	1400	0.13	0.41	400	0.13	3.09	8.05	0.40	2.00	0.80	8.85
S_SD_2.1	3850	0.21	0.52	500	0.20	5.45	20.63	0.40	5.00	2.00	22.63
S_w_1	1250	0.12	0.38	400	0.13	2.76	9.72	0.08	0.50	0.04	9.76
S_SD_2.2	2600	0.14	0.43	400	0.13	5.75	3.47	0.57	5.00	2.85	6.32
S_w_2	1300	0.12	0.39	400	0.13	2.87	9.72	0.08	0.50	0.04	9.76
S_SD_2.3	1300	0.07	0.30	400	0.13	2.87	0.00	0.16	5.00	0.80	0.80
S_w_3	1300	0.12	0.39	400	0.13	2.87	10.26	0.08	0.50	0.04	10.30
S_SD_3.1	3800	0.21	0.52	500	0.20	5.38	22.57	0.44	9.00	3.96	26.53
S_s_1	950	0.09	0.33	400	0.13	2.10	6.09	0.19	2.00	0.38	6.47
S_s_2	950	0.09	0.33	400	0.13	2.10	6.09	0.19	2.00	0.38	6.47
S_SD_3.2	1900	0.11	0.37	400	0.13	4.20	2.12	0.36	12.00	4.32	6.44
S_s_3	950	0.09	0.33	400	0.13	2.10	6.36	0.19	2.00	0.38	6.74
S_s_4	950	0.09	0.33	400	0.13	2.10	6.36	0.19	2.00	0.38	6.74

Table 4.40 – main results of supply circuit

RECIRCULATION CIRCUIT											
Section	*G_{air} [m³/h]*	*A [m²]*	*D_{calc} [m]*	*D_{comm} [mm]*	*A_{real} [m²]*	*v_{real} [m/s]*	*Δp_{loc} [Pa]*	*Δp_{cont} [Pa/m]*	*L [m]*	*$\Delta p_{cont,tot}$ [Pa]*	*Δp_{tot} [Pa]*
R_MD	10610	0.49	0.79	800	0.50	5.86	0.00	0.29	2.00	0.58	0.58
R_w_1	3500	0.49	0.79	800	0.50	1.93	5.17	0.02	0.50	0.01	5.18
R_SD_1	7110	0.40	0.71	800	0.50	3.93	0.00	0.13	5.00	0.65	0.65
R_w_2	3510	0.49	0.79	800	0.50	1.94	5.17	0.02	0.50	0.01	5.18
R_SD_2	3600	0.20	0.50	800	0.50	1.99	2.96	0.04	5.00	0.20	3.16
R_w_3	3600	0.50	0.80	800	0.50	1.99	2.28	0.02	0.50	0.01	2.29

Table 4.41 – main results of recirculation circuit

EXHAUSTIVE CIRCUIT											
Section	G_{air} [m³/h]	A [m²]	D_{calc} [m]	D_{comm} [mm]	A_{real} [m²]	v_{real} [m/s]	Δp_{loc} [Pa]	Δp_{cont} [Pa/m]	L [m]	$\Delta p_{cont,tot}$ [Pa]	Δp_{tot} [Pa]
E_MD	2639	0.12	0.39	400	0.13	5.83	0.00	0.50	0.20	0.10	0.10
E_SD_o	1319	0.07	0.31	400	0.13	2.92	5.96	0.16	10.00	1.60	7.56
E_o_1	659	0.09	0.34	400	0.13	1.46	2.75	0.09	2.00	0.18	2.93
E_o_2	660	0.09	0.34	400	0.13	1.46	2.75	0.09	2.00	0.18	2.93
E_SD_s	1320	0.07	0.31	400	0.13	2.92	6.07	0.16	18.00	2.88	8.95
E_s_1	660	0.09	0.34	400	0.13	1.46	2.75	0.09	2.00	0.18	2.93
E_s_2	660	0.09	0.34	400	0.13	1.46	2.85	0.09	2.00	0.18	3.03

Table 4.42 – main results of exhaust circuit

The selection of these diameters thus ensures compliance with the speed constraints. It is notable that the main duct of the supply circuit is the only component that exceeds the safety margin; however, it remains considerably below the limit specified in the technical guide.
It should be noted that this description is merely an outline of the system, and that the actual calculation will have to be carried out on a case-by-case basis.

The data regarding the pressure drop of three circuits, when combined with the pressure drop in the air handling unit, allows for the selection of the optimal fans to provide, extract and recirculate the air within the building.

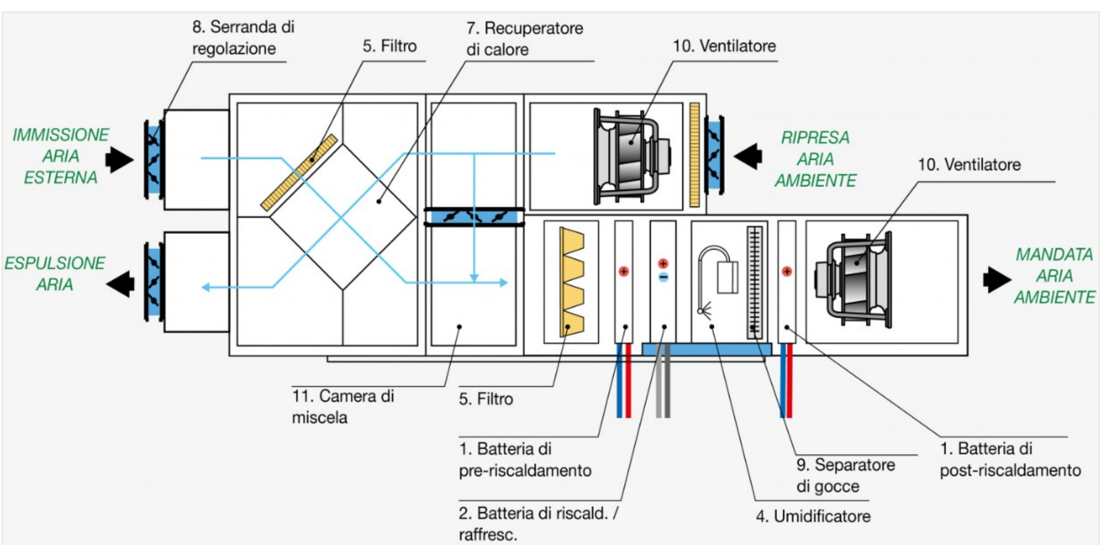

Figure 4.15 – typical scheme of AHU [25]

A detailed examination of the climatic conditions of both winter and summer reveals the transformations that occur in the external air as it passes from the outdoor to the indoor environment.
Subsequently, an evaluation of the capabilities of the heat exchangers can be conducted.

The air handling unit plant is fed by a district heating and cooling plant. This implies that the availability of hot water is confined to the winter months, while the availability of cold water is limited to the summer months. In order to maintain the control over the relative humidity and temperature, it is necessary to implement certain measures during both seasons.

Starting from the summer operation, the first solution is to modify the by-pass factor of the cooling and dehumidification coil in order to align the inlet conditions of the air with the load line of the rooms. The load line is defined as the ratio between the sensible load contribution and the overall load. The range within which the bypass factor can vary is typically between 3% and 30%. This indicates that a certain percentage of the air passing through the coil is not subject to any treatment. Despite the cost-effectiveness of this solution, it necessitates meticulous and challenging design. Furthermore, a continuous adjustment of the operating conditions is necessary to compensate for sensible and latent loads.

The second option for the cooling mode is to provide post-heating with electric resistances after the cooling and dehumidification coil. The objective of the heating process is to raise the temperature of the air to the point of entry. This approach offers superior control of the conditions and fast response within the rooms; however, it results in increased operational costs for the system and non-optimal efficiency from an energetic point of view.

The third option concerns the utilisation of free-heating through the incorporation of external air. In this instance, a heat recovery unit is employed to facilitate the transfer of heat from the external air to the air introduced into the room. The primary constraint of this system is the variability of external conditions. Although this approach is undoubtedly cost-effective, it does not offer the precision in air control that some may desire.

With regard to the winter season, the issue can be attributed to the humidification system. In general, humidification can be classified as isothermal or adiabatic.
The most straightforward and efficacious method is through the atomisation of water supplied from the central plant, which serves to increase the relative humidity of the air.

In this discussion, the precise structure of the air handling unit has not been addressed in detail, as there are multiple solutions, and the optimal choice depends on the specific requirements of the customer. The aim of this thesis is to provide an order of magnitude estimation of the costs and benefits associated with the individual systems.

In terms of costs [24], it should be noted that the installation is not the only expense to be considered.

Description		*Price*
AHU up to 12.500 m^3/h	*Material*	*€ 21.077,21*
	Thermohydraulic works	*€ 30.404,00*
	Labour costs	*€ 1.800*
	Substation	*€ 11.500*
	Air curtain	*€ 1.000*
TOTAL		*€ 65.781,21*

Table 4.43 – costs of AHU and plasterboard

4.4.3 Other possibilities

This thesis considers only fan coil and UTA as potential feasible installations. Other potential solutions were not investigated further due to technical and economic limitations.

One potential solution is the utilisation of radiant panels, which can be employed throughout the year. This solution, however, presents certain critical issues, which are outlined below.
Firstly, it would be prohibitively expensive to demolish and rebuild the entire floor in order to accommodate the installation of radiant panels, given that the buildings are already constructed.
Secondly, the plant provides heat at a high temperature (approximately 65°C), whereas the technology in question functions optimally at around 40°C.
The third critical issue pertains to the technical aspects of the proposed solution. The design of a system that can operate effectively in both summer and winter is a challenging undertaking. The differing potentialities of these two seasons of the system would likely necessitate the integration of fan coils to ensure optimal performance.
A fourth issue is the necessity to address the problem of dehumidification in the context of summer cooling.

Conversely, a system of this nature offers the advantage of continuous operation due to its thermal inertia. Consequently, even during the night, the heat from the central plant can be exploited by this solution.

4.5 Overview of the proposed solutions

This section will provide a summary of the costs associated with the installation, maintenance, average lifetime, air quality and comfort of both the solutions presented.

Marker	*Air heaters and fan-coil*	*Air Handling Unit*
Installation costs	Medium	Very high
Maintenance costs	Low	Medium
Average lifetime	10-15 years	15-20 years
Air quality	Low	Very high
Flexibility	Good	Good
Space	Low	High
Comfort level	Medium	Very high

Table 4.44 – resume of the pros and cons of the plant analysed

5. CONCLUSIONS

The main objectives of this thesis are twofold. The first aim is to implement a remote reading system for customers' meters. The second aim is to propose guidelines for customers wishing to connect to the district heating and cooling network, including an analysis of the costs and benefits associated with each option.

5.1 Summary of thesis results

The first section of the thesis investigated the cost-benefit implications of the implementation of remote meter reading technology, and the substitution of existing obsolete meters currently in operation. At the time of writing, 6 customers who are currently using the district heating and cooling network have been identified as requiring replacement. From an economic point of view, it is evident that the associated initial costs are not negligible, but the operational costs for the company are reduced. Indeed, thus far, the reading of customers' meters has been conducted manually on a month basis through a site visit by a worker. This cost equates to an approximate annual cost of €1.800 per year. The implementation of the network and replacement of the currently active, obsolete meters has a total estimated cost of to €8.523. Then, a long-term scenario was analysed, which consider the replacement of all the meters that could use the network. In this case the total costs would be €32.069. Nevertheless, the real advantage of remote meter reading is not economical, but rather technical and practical. By establishing a reading at a chosen interval, it is possible to gain a comprehensive understanding of consumption trends, which in turn allows for more effective heat production planning. The availability of data at hourly or daily intervals permits the construction of demand curves that can be employed to modulate the production in real time. Moreover, this technology facilitates the prompt and precise detection of any faults or inappropriate use of substations.

In addition to the technical considerations, the implementation of the remote meter reading infrastructure allows the plant to comply with the timeframe specified by European Directive 2018/2002, which mandates the remote reading of thermal meters by 1st January 2027.

The second part of the thesis examined the potential for integrating plants into the district heating and cooling network from the perspective of the end user. As the substation and secondary plant are not present in building C, the analysis was conducted on these units. Firstly, the intended use and zoning of the shed were assumed on the basis of the other users in the served area. Once the material of the building envelope had been defined, an evaluation of the winter and summer heat demand was conducted. Then, the choice of the plant's size was based on the most critical conditions. In order to gain insight into the potential range of requirements for the building, two additional scenarios were considered. This was done to account for the possibility that the substation and system may, with a low probability, also be required to serve other types of user.

It is essential that the solutions under consideration must be capable of being employed for both heating and cooling purposes, thereby ensuring the utilisation of a single terminal throughout the overall thermal year. In considering potential plant solutions, only the water-based were deemed feasible, given the necessity for coupling with district heating and cooling network.

Two types of systems were identified as potential solutions: unit heaters with fan coils and air handling unit. Theoretically, alternative plant solutions could be considered, such as radiant floor system. However, technical and economic constraints make this hypothesis uncompetitive in comparison to the other two.

With regard to the solution that employs the use of unit heaters and fan coils, it should be noted that based on technical and practical evaluation, the unit heaters are intended to be used in the warehouse, while the fan coils are intended for use in the shop and offices. It is of the utmost importance to select the appropriate location for the terminals in accordance with the room's layout, as this will have a significant impact on the functionality and performance of the system. From the economic point of view the overall costs associated with installation and maintenance are relatively low. On the other hand, the quality of the air and, consequently the level of comfort it guarantees are not optimal.

In consideration of the full-air system, the design phase presents a more complex set of considerations than the previous solution. To ensure optimal functionality, it is essential to address a number of challenges, including those related to indoor air quality and velocity constraints in the duct network. From the economic point of view, the initial costs are considerable, as are the subsequent maintenance costs. The main advantages of this solution are the high level of air quality and comfort it provides.

Thus, the first solution is more advantageous from an economic standpoint, although the level of comfort is not optimal. In contrast, the second solution requires a higher economic investment, but it guarantees significantly enhanced comfort and air quality. Ultimately, the optimal choice between the two solutions will be determined by the specific needs of the customer.

The results presented in this thesis are beneficial for both Forgreen and users.
From the company's standpoint, the results offer an estimation of the costs and an overview of the benefits that remote meter reading can provide during normal operation of the plant.
From the customer's perspective, the analysis of the system solutions provides a comprehensive overview of the technical, economical and environmental aspects.

5.2 Limitations of the study

At the moment, it is not possible to provide an accurate estimation of the number of users that will be connected to the district heating and cooling network in the future and, consequently, consumption of thermal energy, cannot be precisely quantified. Therefore, the real benefits of meter reading cannot be precisely estimated. However, a more precise estimation could be achieved after one thermal year, given that a comparison could be made with the previous thermal year.

The calculation of heating and cooling needs was conducted on building C. It should be noted that not all the units in the district zone are identical, and therefore the results may not be representative of the entire district. In fact, it is notable that there is a considerable degree of variation in terms of floor area, height, and usage between the units. It seems reasonable to assume that the types of installations remain the same in the other units, with the difference of installed power.

5.3 Suggestions for future implementations

It would be imperative to install a meter on the supply line at the outset of the distribution network in order to gain a comprehensive understanding of the precise amount of energy that leaves the power station and how much reaches the substations. Such system would facilitate the accurate determination of the precise share of losses along the distribution network.

Another beneficial strategy would be to implement an upgrade to the power centre's remote control system, thereby establishing a reliable infrastructure for monitoring and control.

In order to enhance control over water consumption, it is recommended that a water meter be installed at the entrance to the power station, with remote accessibility to facilitate real-time monitoring of water consumption due to reintegration resulting from potential network leaks.

APPENDIX

APPENDIX A

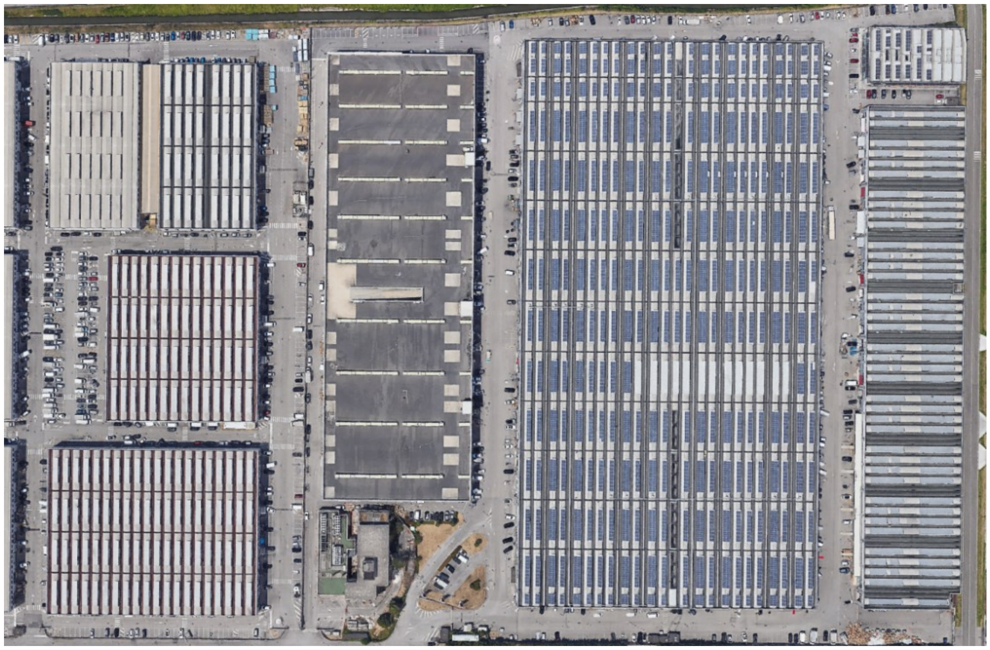

Figure A.1 – plan view of the area served by Telezip

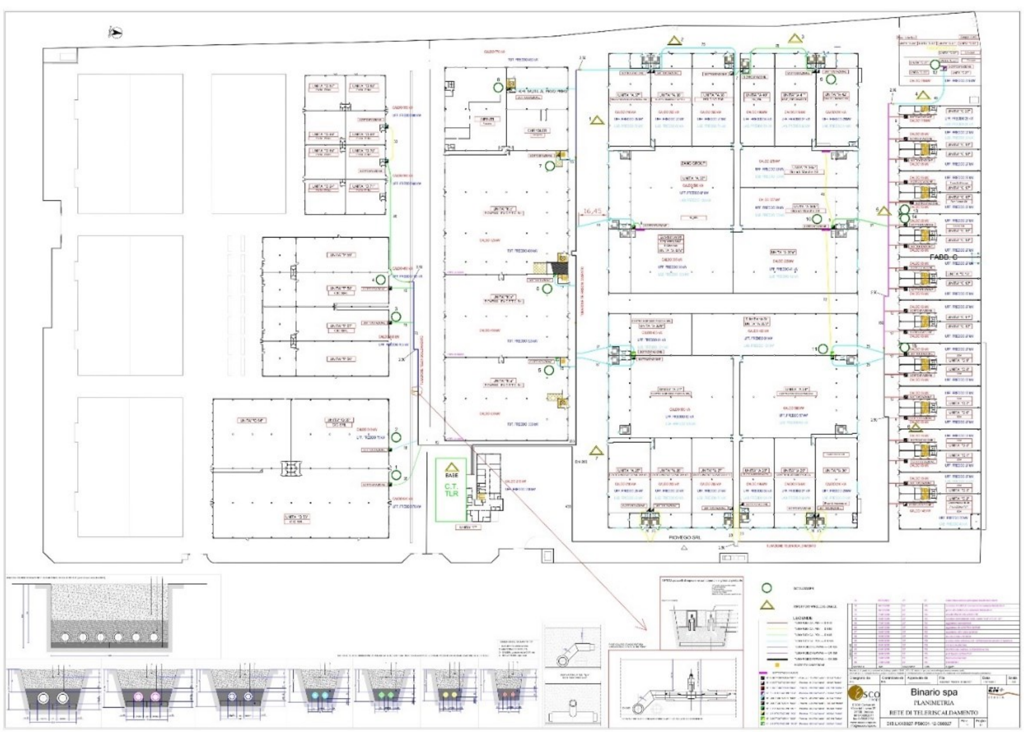

Figure A.2 – identification of the name of the buildings in the area served by Telezip

APPENDIX B

OFFICES							
Hour	q_{vent} *[W]*	$q_{infiltration}$ *[W]*	$q_{windows}$ *[W]*	q_{opaque} *[W]*	q_{int_gain} *[W]*	q_{solar} *[W]*	q_{tot} *[W]*
6	0	-634.83	-132.62	1837.93	0.00	137.01	1207.50
7	0	-547.78	-114.43	1667.80	2968.50	117.44	4091.53
8	0	-391.08	-81.70	1400.84	3866.25	234.88	5029.19
9	0	-164.74	-34.41	1422.37	3966.00	391.46	5580.68
10	0	96.43	20.14	1459.12	3966.00	587.19	6128.89
11	0	392.42	81.98	1854.90	4032.52	763.35	7125.14
12	0	670.99	140.17	2476.53	4099.02	939.51	8326.20
13	0	879.93	183.82	3625.41	4132.25	1056.95	9878.35
14	0	1019.21	212.91	4481.07	4165.54	1135.24	11013.94
15	0	1071.45	223.83	5034.71	4165.54	1115.67	11611.15
16	0	1019.21	212.91	5406.20	4165.53	1037.37	11841.21
17	0	897.34	187.45	5676.06	4198.75	880.79	11840.39
18	0	705.82	147.45	5663.56	4198.75	724.20	11439.78
19	0	479.47	100.16	5342.08	4198.75	606.77	10727.24
20	0	253.13	52.88	4956.75	2636.00	528.47	8427.23
21	0	61.61	12.87	4765.93	0.00	450.18	5290.59
22	0	-112.50	-23.50	4503.53	0.00	391.46	4758.99
23	0	-251.79	-52.60	4161.29	0.00	352.32	4209.22
24	0	-356.26	-74.42	3841.61	0.00	313.17	3724.10
1	0	-443.31	-92.61	3445.78	0.00	274.02	3183.89
2	0	-530.37	-110.79	3009.51	0.00	234.88	2603.22
3	0	-600.01	-125.34	2685.21	0.00	215.30	2175.16
4	0	-652.24	-136.25	2384.29	0.00	195.73	1791.52
5	0	-669.66	-139.89	2158.64	0.00	156.58	1505.68

Figure B.1 – main results of Carrier method for offices

WAREHOUSE							
Hour	q_{vent} *[W]*	$q_{infiltration}$ *[W]*	$q_{windows}$ *[W]*	q_{opaque} *[W]*	$q_{int\ gain}$ *[W]*	q_{solar} *[W]*	q_{tot} *[W]*
6	0.00	-2036.15	-286.03	3686.50	0.00	54.43	1418.74
7	0.00	-1839.79	-258.45	3323.34	2790.00	163.29	4178.39
8	0.00	-1486.33	-208.80	2702.10	3600.00	217.72	4824.69
9	0.00	-975.78	-137.08	2791.87	3690.00	252.36	5621.37
10	0.00	-386.69	-54.32	2909.09	3690.00	282.05	6440.13
11	0.00	280.95	39.47	3656.52	3750.00	306.79	8033.73
12	0.00	909.32	127.74	4884.33	3810.00	326.58	10057.97
13	0.00	1380.60	193.94	6823.35	3840.00	346.37	12584.26
14	0.00	1694.78	238.08	8365.48	3870.00	366.16	14534.51
15	0.00	1812.60	254.63	9410.95	3870.00	376.06	15724.24
16	0.00	1694.78	238.08	10370.22	3870.00	390.91	16563.99
17	0.00	1419.87	199.46	11101.19	3900.00	395.85	17016.38
18	0.00	987.87	138.77	11299.58	3900.00	296.89	16623.11
19	0.00	477.32	67.05	11073.73	3900.00	252.36	15770.46
20	0.00	-33.23	-4.67	10768.71	2490.00	217.72	13438.53
21	0.00	-465.23	-65.35	10702.98	0.00	183.08	10355.48
22	0.00	-857.96	-120.52	10080.63	0.00	158.34	9260.49
23	0.00	-1172.15	-164.66	9398.33	0.00	143.50	8205.02
24	0.00	-1407.79	-197.76	8584.97	0.00	133.60	7113.02
1	0.00	-1604.15	-225.35	7591.88	0.00	113.81	5876.19
2	0.00	-1800.52	-252.93	6507.52	0.00	103.91	4557.99
3	0.00	-1957.61	-275.00	5721.52	0.00	89.07	3577.98
4	0.00	-2075.43	-291.55	4987.82	0.00	79.17	2700.01
5	0.00	-2114.70	-297.07	4304.19	0.00	64.33	1956.75

Figure B.2 – main results of Carrier method for warehouse

APPENDIX C

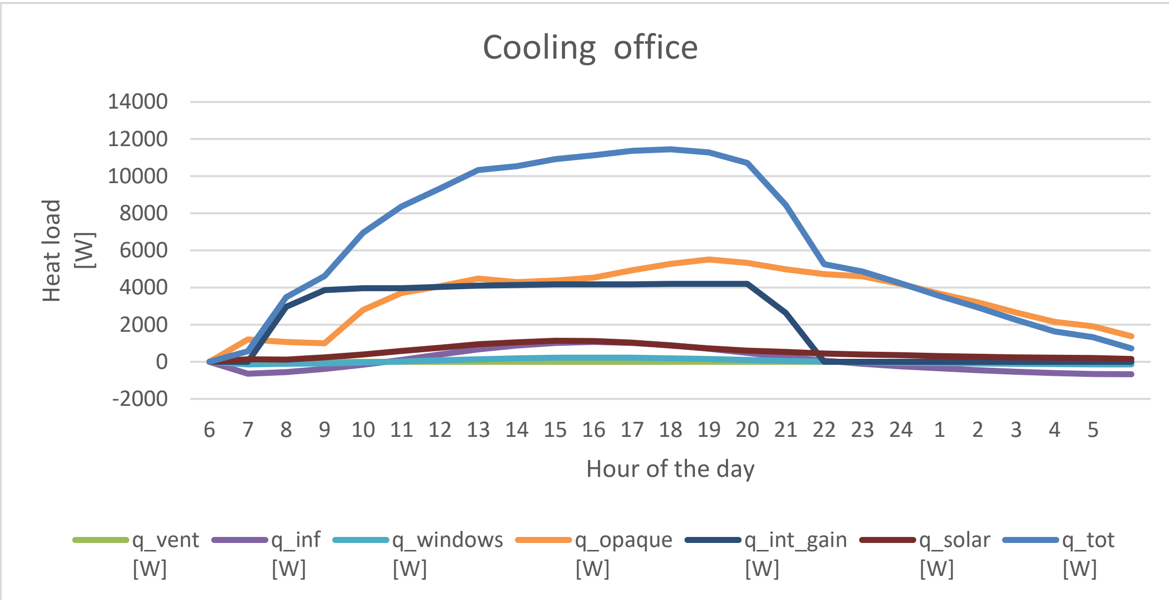

Figure C.1 – cooling profile of the office

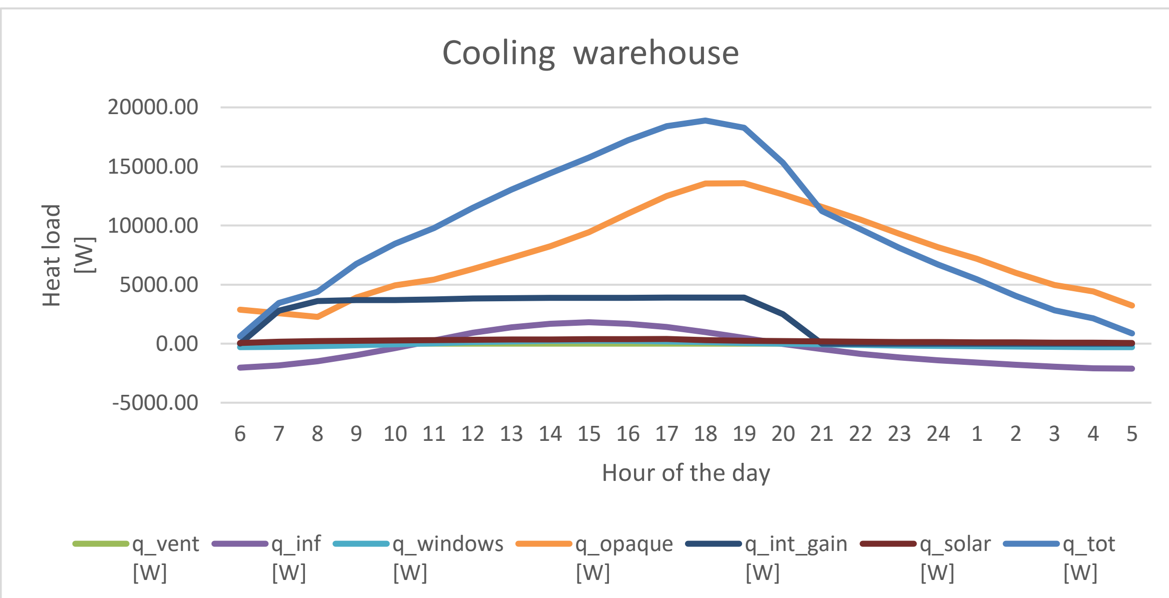

Figure C.2 – cooling profile of the warehouse

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