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HVAC design of a public library in Italy.

Tesi di Laurea Magistrale in
Energy Engineering - Ingegneria Energetica

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Aknowledgments

A tutti coloro i quali hanno reso possibile il raggiungere questo grande traguardo: ai miei genitori, a cui devo la maggior parte dei meriti, ai familiari, agli amici d'università per il loro supporto costante e alla squadra di B.R.E.. Un merito particolare a Diego e Gioele.

Abstract

Although a large share of energy is used for buildings, the actual presence of indoor environmental quality, and underlying air quality, can be questioned. The objective of this thesis is to develop the design of a building for municipal library and auditorium use. It is, therefore, to understand the requirements necessary to achieve thermal comfort and acceptable air quality. In addition to this, it is necessary to think about the distribution of air in the rooms and the type of system that best suits the context and needs.

Keywords: HVAC system, library, auditorium, desing, IAQ, thermal comfort

Abstract in lingua italiana

Nonostante una grande quotaparte dell'energia sia usata per gli edifici, l'effettiva presenza della qualità dell'ambiente interno e, ad essa sottesa, la qualità dell'aria, può essere messa in discussione. L'obiettivo di questa tesi è di sviluppare la progettazione di un edificio ad uso biblioteca comunale e ad uso auditorium. Si tratta, perciò, di comprendere le richieste necessarie al raggiungimento del comfort termico e di una qualità dell'aria accettabile. Oltre a ciò è necessario pensare alla distribuzione dell'aria negli ambienti e al tipo di impianto che più si adatta al contesto e alle necessità.

Parole chiave: impianto HVAC, biblioteca, auditorium, progettazione, IAQ, comfort termico.

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Introduction

Although a large portion of energy is used for HVAC (Heating Ventilation and Air Conditioning) systems in libraries and public assembly spaces, the actual presence of Indoor Environmental Quality (IEQ) can be questionable. IEQ depends mainly on factors connected to human nature and on factors connected to the environments. Those related to environment are mainly three: noise, daylight levels, and Indoor Air Quality (IAQ). IAQ depends on various air parameters such as velocity, humidity, temperature, and the degree of purification from pollutants. In a study of Yang et al. [1] about educational classrooms, artifical lighting does not affect student's performance in their school carrer but affects their satisfaction. In a study of Zhang et al. [2], instead, poor acoustic performance affects both student's performance and satisfaction. Overall, a good level of indoor environment is an essential piece when it comes to student's health and student's performance. Another aspect of IAQ is air renewal. In a study of Santos et al. [3] air renewal and filtration is a method to exploit HVAC equipment to reduce the risk of infection by bacteria and viruses. The spread of Coronavirus in 2019 and the following years has been a drive for public and non public entities to promote IAQ. Given all these aspects, it is well known that improvements on IEQ carries over a greater energy consumption and costs for equipment supply, maintenance and operation. Thus, a look on the energetic aspect is mandatory to deliver a more sustainable building's operation scheme.

The aim of this project work is to develop a HVAC system for a public library comprehending an auditorium and some multipurpose spaces. The main challenge is that building's spaces have got a variety of needs in term of ventilation and heat loads as over time the occupancy is not constant and the frequency of use is not constant as well. In this work the procedural iter is to pass through considering the best practices, the laws and the guidelines of HVAC field as first step; then, the energy analysis of thermal loads and ventilation requirements as second step; then the evaluation of the equipment size and rooms' air distribution layout as third and last step.

Thesis' objectives and geo-historical framework

Figure 1: State of fact of the old cinema located in Lainate to be requalified.

The area covered by this intervention is located in the centre of the Lainate municipal area, in Largo Vittorio Veneto, a few metres from the main entrance to Villa Litta and the municipal headquarters. The former cinema building was built after World War II and has been unused. The aim of the project is to redevelop the existing building into a multipurpose centre with technological features and architectural configurations that meet contemporary functional requirements and plant configurations. The project is part of a programme called "Re-industrialisation Plan" of the former Alfa Romeo plant in Arese (MI), which also covers the municipal territories of Lainate, Garbagnate Milanese and Rho.

HVAC standards

In the context of designing libraries where it is not required to preserve, store and conserve historical books, the state of the art in HVAC can be investigated in the literature of the major players in the field of HVAC, such as ASHRAE, AiCarr and various books, manuals, standards, laws and guidelines.

Guidelines from "Manuale del termotecnico"

The third edition of Nicola Rossi's book [4] dates back to 2009 and provides useful insights. In particular, the suggested thermo-hygrometric conditions are:

	T,w in °C	**T,s in °C**	**RU in %**
Reading rooms	20	25	50
Storage rooms	13÷18	13÷18	50
Film storage	15	15	35
Magnetic tapes storage	15	15	40

Table 1: Thermo-hygrometric condition for libraries according to Nicola Rossi's book.

Where:

- T, w is the setpoint of temperature in winter;
- T, s is the setpoint of temperature in summer;
- RU is relative humidity.

The reference data to be assumed for the design of air conditioning systems [4] are:

- Crowding: very variable, up to 1 person each square meter in reading rooms; in storage rooms 1 person each 90 square meter;
- Lighting: in the reading rooms the levels are rather low; 15 to 20 W/m^2; in the storerooms, and only occasionally, up to 10 W/m^2;
- Fresh air renewal: 20 m^3/h per person, unless otherwise specified; in the storage rooms there are practically no people present;
- Total air exchange: 8 to 12 vol/h;
- Air speed in the occupied zone: less than 0.13 m/s;
- Sound level: 35 dB(A).

The author then adds that from the point of view of the type of systems, all-air systems are also preferred for libraries, capable of serving, independently, the different zones so as to achieve constant control of the thermo-hygrometric conditions throughout the day. As far as auditoriums are concerned, the same book gives the following information. Ventilation requirements, dictated by the need to contain ambient air pollution due to the emission of carbon dioxide and other effluents by people, set 30 to 50 $m3/h$ per person for the amount of external air to be supplied by air conditioning systems; it must also be verified that the rooms are over-pressurised towards the outside to avoid unpleasant air ingress from the foyers and entrances. Air quality must also be maintained by using air filtration, which should be done with F5 pre-filters and F7 efficient pocket post-filters. The thermo-hygrometric conditions that it is good to maintain in the different rooms are:

- Theatres: 22°C to 25°C with relative humidity from 40% to 50%, winter to summer.
- Lighting: in the reading rooms the levels are rather low; 15 to 20 W/m^2; in the storerooms, and only occasionally, up to 10 W/m^2;
- Cinemas: 20°C to 24°C with relative humidity from 40% to 50%, winter to summer.

The recommended sound level must be below 30 db(A).[4] With regard to the type of system, the author suggests a single-zone all-air system with constant air flow. One of these two solutions is recommended for air diffusion:

- Supply from the side and return from below.
- Supply from below and return at the top.

The first, uses nozzles, given the required throw. The return air leaves the space from under the seats or from the sides of the space. The second involves supplying air at a temperature a few degrees lower than the ambient temperature and is supplied at the foot of the seats or with special diffusers on the bleachers; the air rises upwards, removing the heat and pollutants emitted by the people. Recovery is carried out at the top. This system is also advantageous in the event of a fire as the air escaping from the floor keeps the lower part of the room free of smoke, while the smoke is evacuated at the top.[4]

ASHRAE 2019 Applications

At chapters 3 and 5 there are some common practices which describe the way air is supplied and extracted. Regarding auditoriums, supply is from auditorium's back and return from stage's fly as most of the internal gain comes from lights; lights are usually placed in the fly and their convective heat portion is brought away by the return air. In atrium, the book suggests two ways of managing air diffusion:

- low supply and high exhaust
- a combination of low and high supply and low and high exhaust.

[5]

ASHRAE 62.1

The reference manual published by ASHRAE that may be of interest to the present case study is Standard 62.1 of 2022 [6]. Section 6.2 provides a prescriptive method for calculating the minimum outdoor air quantity to achieve acceptable air quality. Specifically,

$$V_b z = R_p \times P_z + R_a \times A_z \tag{1}$$

where:

- $V_b z$ represents the air flow rate described above;
- P_z represents the number of people present during the use of that space,
- A_z represents the net occupiable area of that space.
- R_p and R_a are tabular values, which vary from space to space. Table 6-1 provides these values; specifically under the public assembly headings 'Auditorium seating area' and 'Libraries'.

Work goals

The goal of this document is to provide an example of HVAC design in public libraries and/or auditoriums and to accomplish thermohygrometric comfort and good indoor air quality.

1 | Architectural and thermal analysis

1.1. Steps overview

The first step in the analysis of building performance is to assess thermal needs and ventilation needs. Regarding thermal needs, the main goal should be to seek thermal comfort, that is the condition of mind that expresses satisfaction with the thermal environment and is assessed by subjective evaluation. Thermal comfort is linked to thermal neutrality, that is maintained when the heat generated by human metabolism is allowed to dissipate, thus maintaining thermal equilibrium with the surroundings . A library or an auditorium fall in the category of moderate environment, which differs from the severe environment category for the aim of thermal comfort: in severe environment the aim is to prevent stressful situations with even serious consequences while in moderate ones, the degree of discomfort of workers is measured, which does not necessarily cause pathologies. The main factors that influence thermal comfort are those that determine heat gain and loss, namely metabolic rate, clothing insulation, air temperature, mean radiant temperature, air speed and relative humidity. Psychological parameters, such as individual expectations, also affect thermal comfort. The ASHRAE 55-2010[7] Standard defines metabolic rate as the level of transformation of chemical energy into heat and mechanical work by metabolic activities within an organism, usually expressed in terms of unit area of the total body surface. Metabolic rate is expressed in *met* units, which are defined as follows: 1 $met = 58.2 W/m^2$, which is equal to the energy produced per unit surface area of an average person seated at rest. The amount of thermal insulation worn by a person has a substantial impact on thermal comfort, because it influences the heat loss and consequently the thermal balance. Layers of insulating clothing prevent heat loss and can either help keep a person warm or lead to overheating. Generally, the thicker the garment is, the greater insulating ability it has. Depending on the type of material the clothing is made out of, air movement and relative humidity can decrease the insulating ability of the material. 1 *clo* is equal to 0.155 $m^2 K/W$. In the analysis of thermal comfort the following indices are used:

- Predicted Mean Vote (PMV), an integer value ranging from -3 to $+3$ where the first means very cold, the latter means very hot and 0 means neutral.
- Predicted Percentage Dissatisfied (PPD), is a percentage value that, as name says, tries to estimated the amount of people dissatisfied with that specific thermal condition.

According to the EN ISO 7730:2006[8], the formula for PMV is

$$PMV = CT^{0,303e-0,036M+0,028} \tag{1.1}$$

where

- CT is the arithmetic difference between the thermal power given to the ambient by the person and the thermal power exchanged by the same in homeothermic conditions.

and these indices are correlated by this formula:

$$PPD = 100 - 95e^{0,03353PMV^4-0,2179PMV^2} \tag{1.2}$$

Note that even when PMV is equal to 0, $PPD = 5\%$; this means that in optimal conditions there exist a percentage of people that will be unsatisfied by them. This indices have got sense only in certain ranges of M,I_{cl},V_{ar}, and T_a, shown in table 1.1.

M	from 0.8 met to 4 met
I_{cl}	from 0.0 clo to 2 clo
T_a	from 10 C to 30 C
V_{ar}	from 0.0 m/s to 1 m/s

Table 1.1: Applicability ranges of the variables M, Icl, Var, and Ta of the formula of PMV.

To guarantee thermal comfort, the standard recommends that the PPD is less than 10%, it means that no more than 10% of subjects consider the thermal environment to be unsatisfactory. This condition corresponds to a PMV vale between -0.5 and +0.5. Environments characterized by PMV values higher than +0.5 are perceived as too hot by the occupants, while environments in which the PMV is lower than -0.5 are perceived as too cold. EN ISO 7730:2006[8] takes into account local discomfort as well, but in this project work it won't be analized. Moreover, another parameter for thermal comfort is used: operative temperature (T_{op}). It accounts for convective and radiative heat exchanges of the human body and under simplifying conditions, it can be calculated as the average between air temperature and mean radiant temperature. Regarding ventilation needs, designing a ventilation systems promotes occupant health and well-being, and requires a clear understanding of the ways that ventilation airflow interacts with, dilutes, displaces, or introduces pollutants within the occupied space.

After having briefly shown what the intentions and objectives are, to achieve a detailed study of how building behaves, a building model must be created and customized. A building model is a virtual, computer-based representation of the building, where thermal behaviour is simulated using normative. The inherent model's uncertainties means to adopt a safety factor on the results. This analysis is performed using Edilclima. In this section, the building's layout characteristics are shown, thermal requirements and loads are investigated. The software "Edilclima" is used to perform calculations, which is recognized to be a certified software by the Italian State. The analysis will use the most recent UNI/TS 11300:2019[9] to perform heating load calculus and Carrier-Pizzetti[10] method to perform cooling load calculus.

Figure 1.1: Photo of the library after work completion.

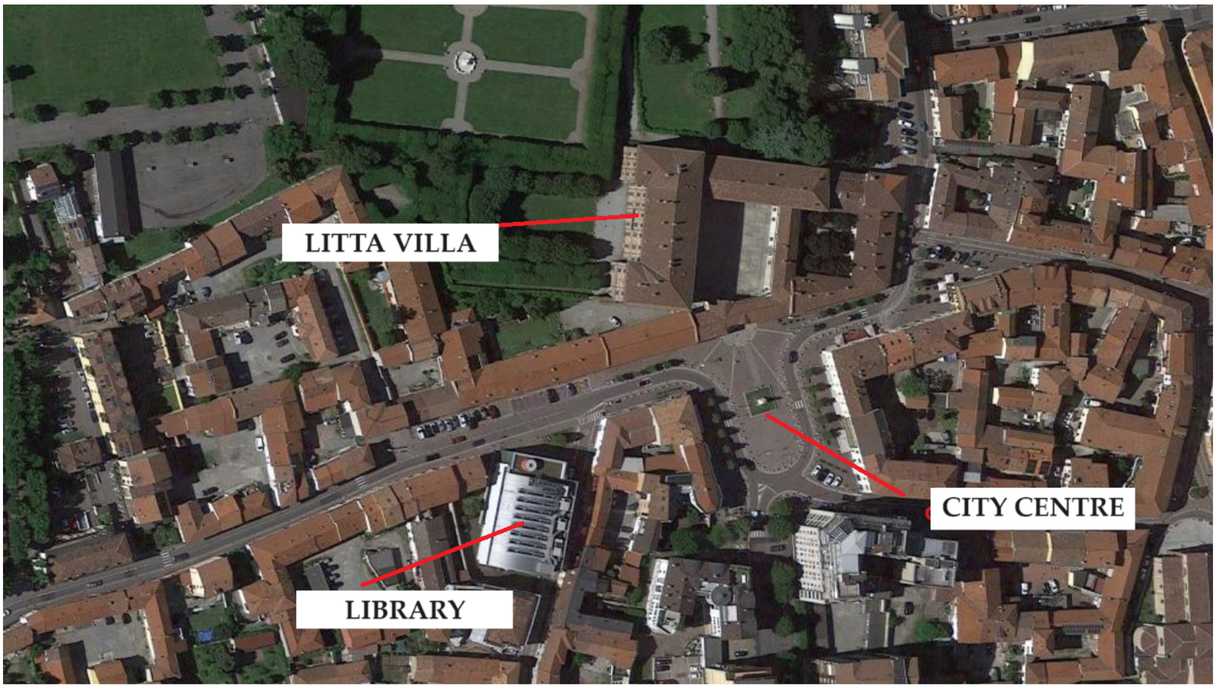


Figure 1.2: Eagle view of the area where the building is located.

The figures 1.1 and 1.2 show the urban context where the building is immersed and provide a photo of the finished works. The main glass façade faces north with an angle of 17° toward west. According to legislation DPR 412/93[11] , this building is categorized as:

- E.4(2) buildings destined to recreative use or similar as cinemas, theatres, meeting rooms for congress.
- E.4(1) buildings destined to recreative use or similar as exhibitions, museums, libraries or worship assemblies.

Destination of use is a criteria that can group various buildings under a category, through which legislation assigns values for the project about the limit of temperature and the limit of time usage of thermal plants for example.

1.2. Overview of the spaces and their functionality

In this subsection the layout is presented.

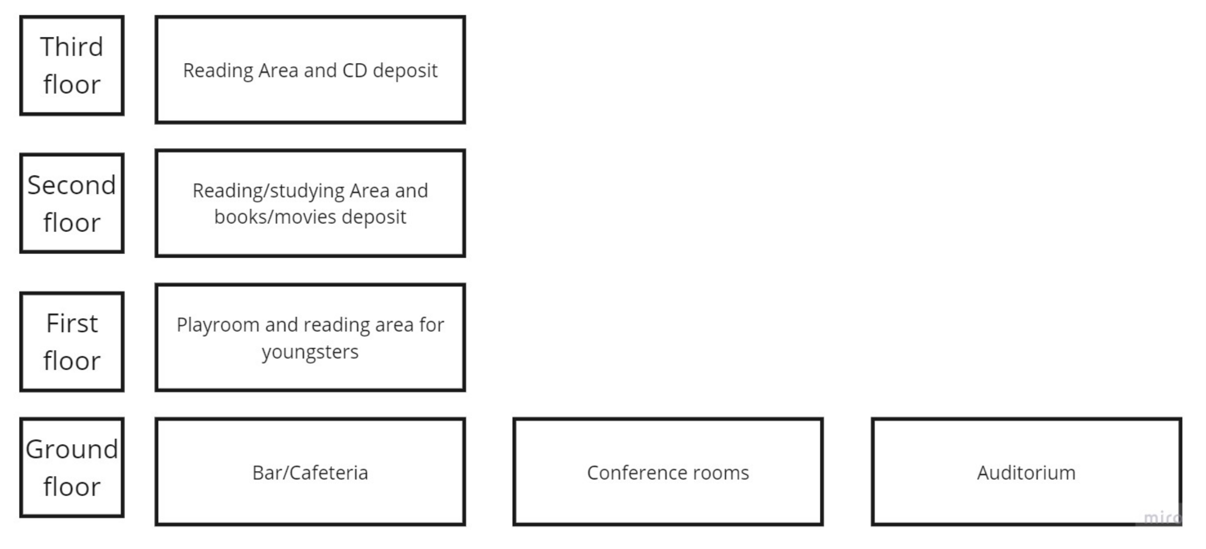


Figure 1.3: Spaces functionalities scheme.

In figure 1.3 there are the destinations of use of each floor. In the following pictures, those are highlighted with colors on the planimetry.

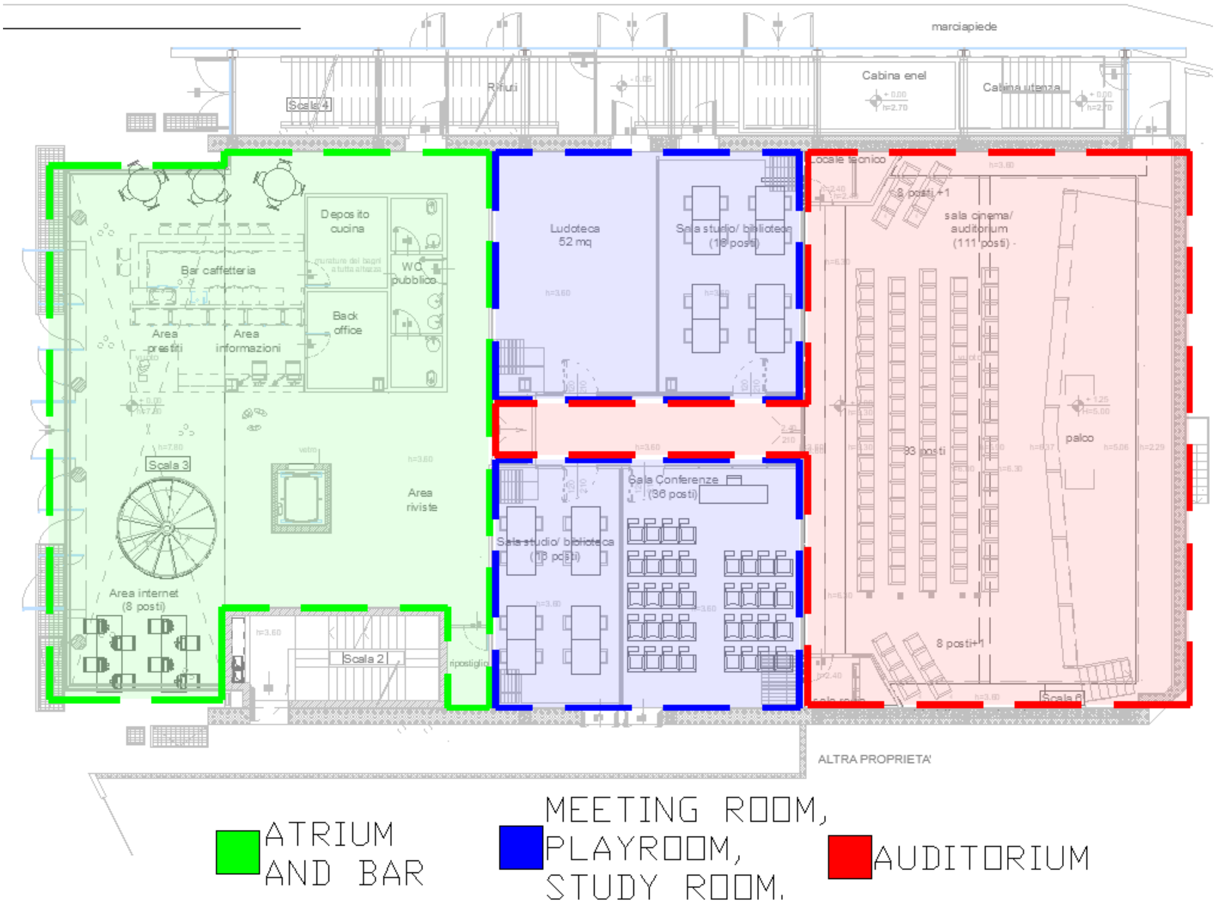


Figure 1.4: Spaces located at ground floor scheme.

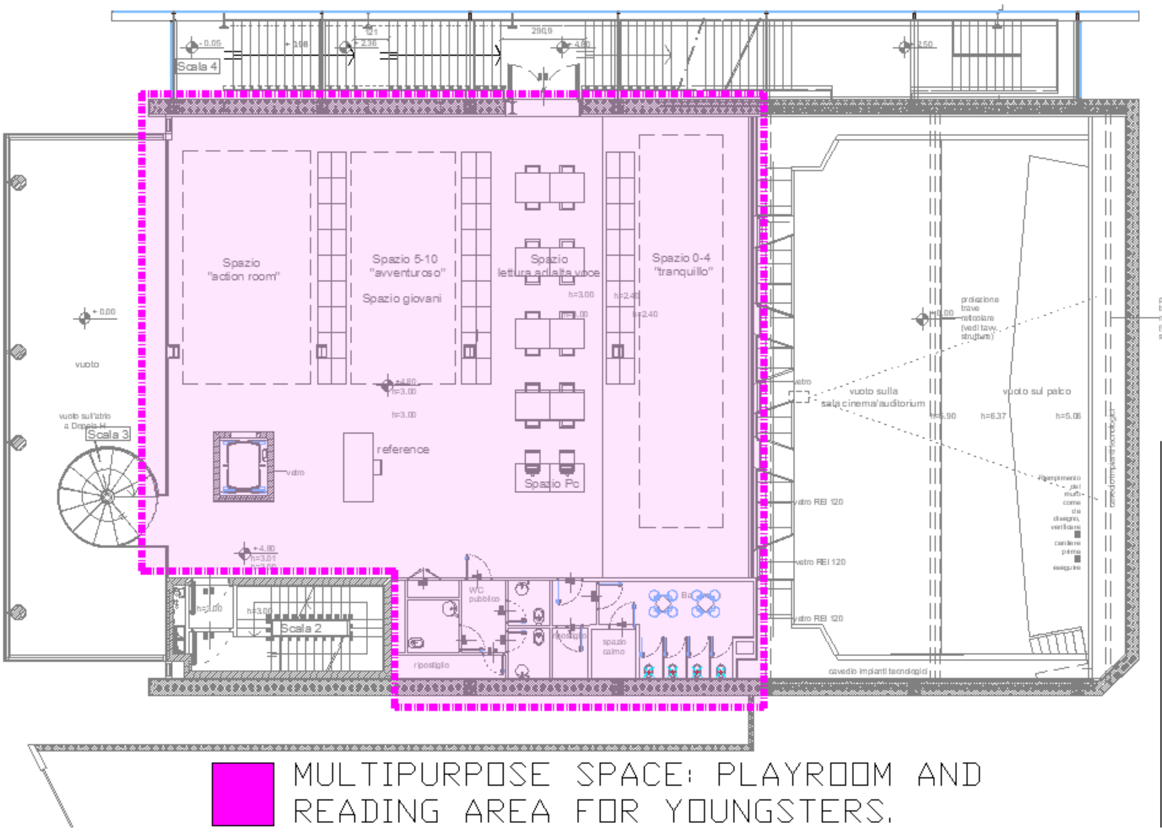


Figure 1.5: Spaces located at first floor scheme.

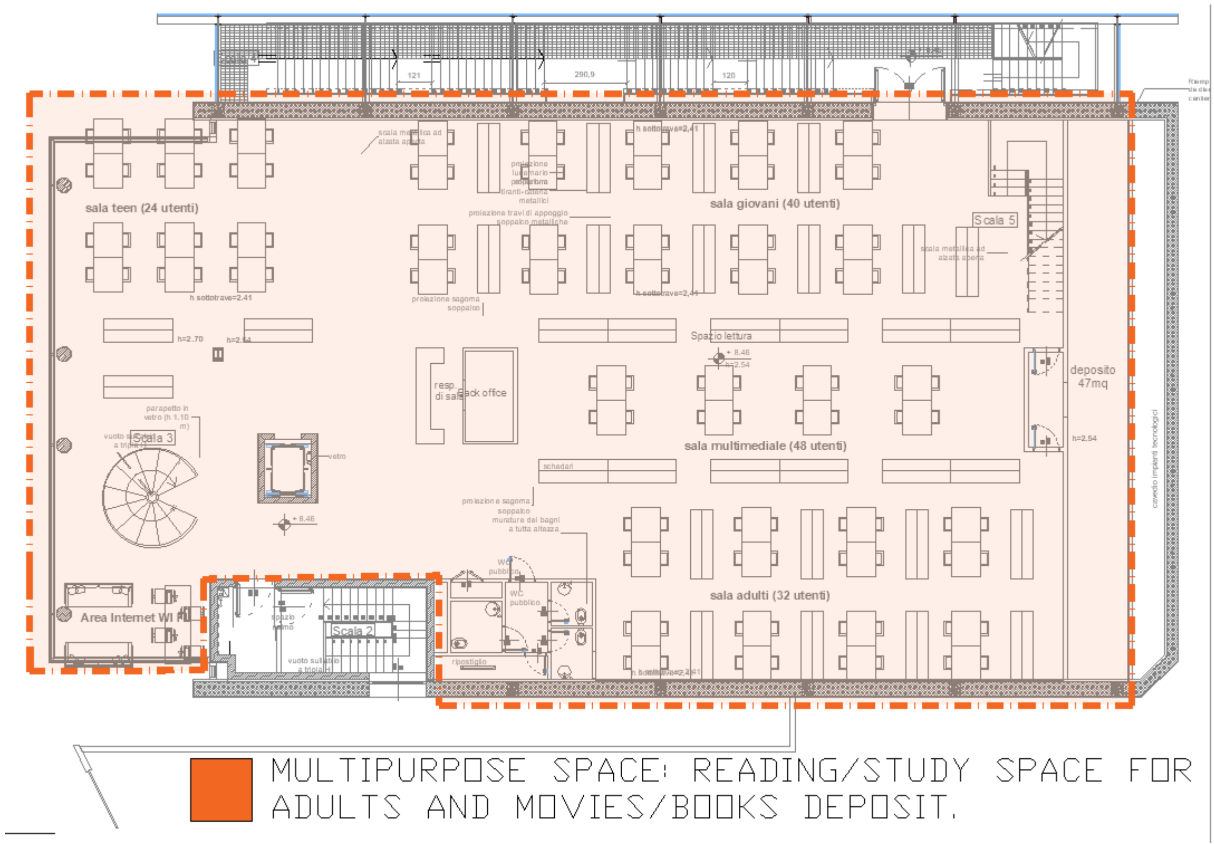


Figure 1.6: Spaces located at second floor scheme.

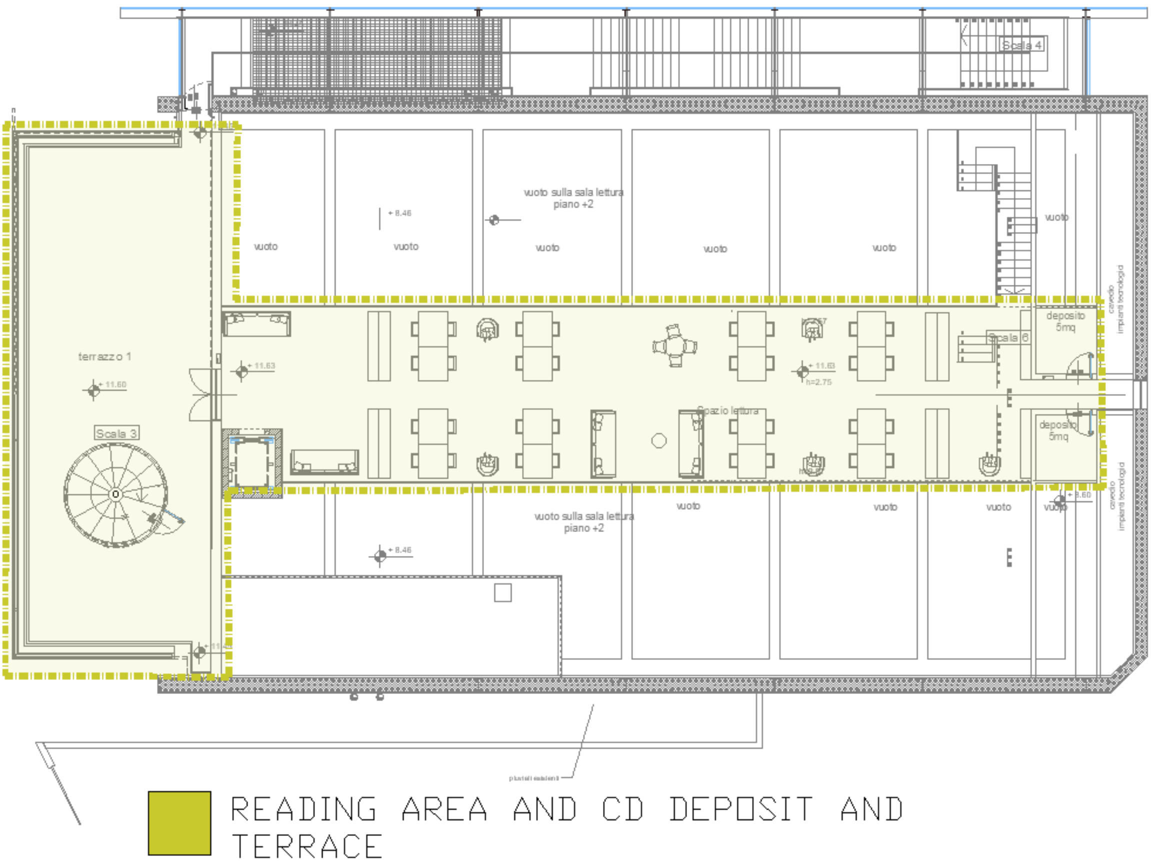


Figure 1.7: Spaces located at third floor scheme.

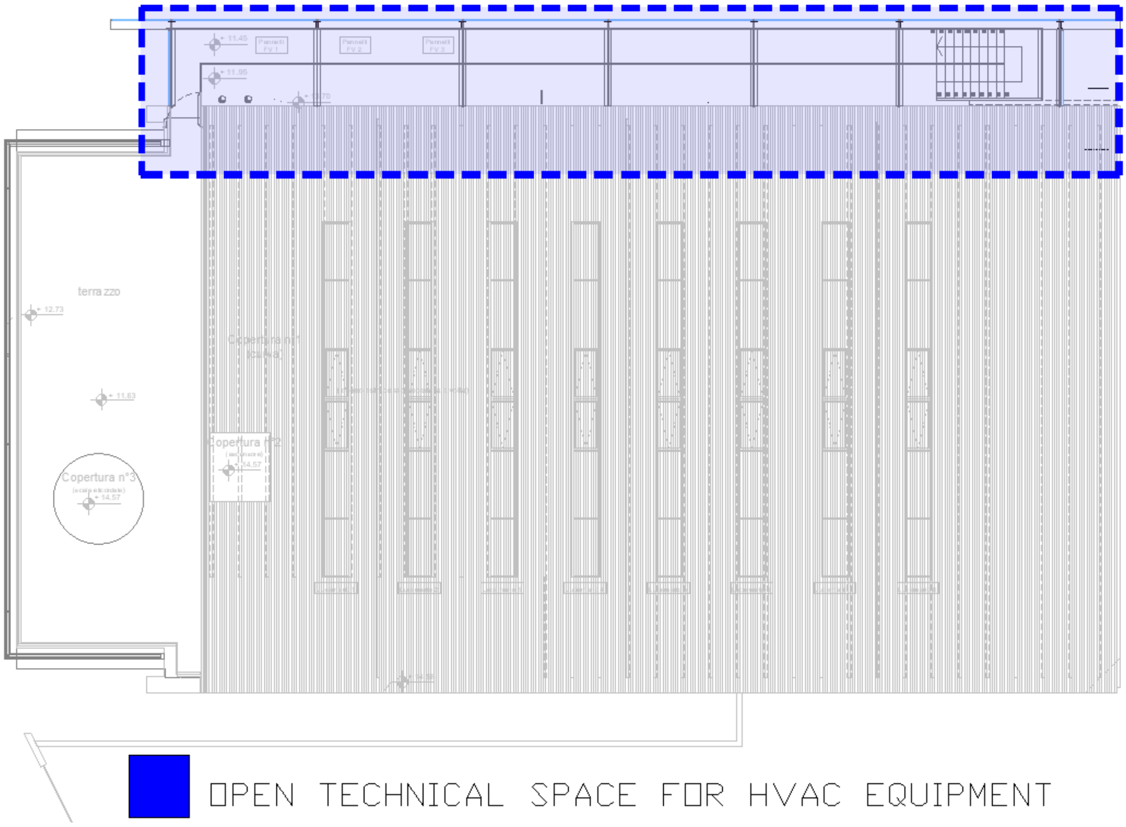


Figure 1.8: Spaces located at last floor scheme.

The setpoints for the thermohygrometric conditions are reported in table 1.2. The reason why those remain equal is the similarity of use conditions: the value of CLO and MET are very similar. The used values are taken from CBE thermal comfort tool [12]:

- In winter: $CLO = 1.0$ clo and MET $= 1.0$ met
- In summer: $CLO = 0.5$ clo and $MET = 1.2$ met

Where 1.2 met corresponds to standing quiet person, 1.0 met corresponds to seated quiet person, 0.5 clo corresponds to a typical summer clothing, and 1.0 clo corresponds to a typical winter clothing. Here are reported the results from the CBE thermal tool in the two scenarios: winter and summer. In both simulations a strong assumption has been made: operative temperature is almost the same of the air temperature. This assumption is a simplistic one but can find truth in the thermal performances of the envelope, as they are very well insulated. Thermal comfort is obtained in the region of the diagram filled with light-blue and the space conditions are reported in the red point. When the red point falls in the light-blue region, PMV is in the acceptable range of -0.5 to $+0.5$, hence PPD is lower than 10%.

Figure 1.9: Summer results of CBE tool simulation.

Figure 1.10: Winter results of CBE tool simulation.

As can be seen, winter simulation yields a negative feedback, giving a PMV of -0.88

and it means that the space is perceived as cold and around 20% of the people will be unsatisfied by the conditions. However, the choice of keeping 20 C as winter setpoint has been done because, differently from summer, people can easily cover furthermore their bodies increasing the reference CLO value and because it means consuming less energy to heat the building.

Cooling Period	Design Temperature: 26°C	Design Humidity: 50%
Heating Period	Design Temperature: 20°C	Design Humidity: 50%
Cooling Period	Maximum external Temperature: 32°C [13]	Humidity: 50%
Heating Period	Minimum External Temperature: -5°C [13]	Humidity: 50%

Table 1.2: Set points for design temperature and relative humidity and reference data for external temperatures and relative humidity.

1.3. Zoning

The aim of the zoning process is to group the spaces into different zones. Each zone definition may be led by key factors like temperature setpoint similarity, destination of use, thermal load similarity. As written in the previous subsections, four zones are chosen. The grouping is done by considering the destination of use and the independency required by each space. For example, grouping the auditorium space into the hall/entrance zone would be useless because requirements in terms of ventilation would be vastly different, considering the intermittence of occupancy during daytime. In this case, choices are reported in table 1.3.

Zone n. 1	Bar, Atrium, Meeting Rooms, Playroom, and Study Room.	See green and blue color in table 1.4.
Zone n. 2	Multipurpose space at 1st floor	See magenta color in table 1.5.
Zone n. 3	Multipurpose space at 2nd floor	See orange color in table 1.6.
Zone n. 4	Auditorium	See red color in table 1.4.

Table 1.3: Zoning choice: space grouping and references to the colored layouts

The characterization of spaces is an essential aspect of building performance analysis, particularly in the context of energy efficiency and occupant comfort. To achieve these goals, various factors need to be considered, including thermal comfort set points, heat gains from internal sources such as:

- equipment;
- people;
- lighting;
- air infiltration rates.

The different characteristics considered will be later discussed, together with the values used for each parameter, and the methodology adopted to estimate these values. The result of thermal load strongly depends on heating and cooling setpoints. Using this data, a computer model in Edilclima software[14] is created and energy performances simulated. The envelope must comply with [15], the Italian normative about the construction of new buildings and/or the major renovation of old ones. The normative dictates the performance of all the climatized spaces' opaque and transparent envelopes that are adjacent to a non climatized space in its III prospectus for each climatic zone. A climatic zone is a group of Italian cities which share similar need in conditioning requirements. The building analized in this project work is contained in the group of climatic zone E. In particular, table 1.4. shows the requirements for climatic zone E, and so for the thesis building.

Intervention typology	**Maximum allowed thermal transmittance U**
Horizontal, opaque, cover	$\leq 0.20\ W/m^2K$
Horizontal, opaque, floors	$\leq 0.25\ W/m^2K$
Vertical, opaque, perimetral	$\leq 0.23\ W/m^2K$
Transparent total U, not only glass' U	$\leq 1.30\ W/m^2K$

Table 1.4: Maximum allowed thermal transmittance for the building's envelope elements.

Instead, in the next table, table 1.5, there are the worst transmittances obtained. As can be seen, they meet the requirement of the Italian normative.

Element	Maximum thermal transmittance U
Horizontal, opaque, cover	0.20 W/m^2K
Horizontal, opaque, floors	0.24 W/m^2K
Vertical, opaque, perimetral	0.20 W/m^2K
Transparent total U, not only glass' U	1.23 W/m^2K

Table 1.5: Maximum thermal transmittance for each installed element of the studied building.

All the properties of the envelopes are calculated according to one of these normatives:

- UNI TS 11300-1:2014[9];
- UNI EN ISO 6946:2018[16];
- UNI EN ISO 10077:2018[17].

To show in a greater detail the main stratigraphies, here are reported the main structures' characteristics.

- In figure 1.14 there are the details of the continuous facade glass.
- In figure 1.12 there are the details of the external wall facing south.
- In figure 1.11 there are the details of the external wall facing east and west.
- In figure 1.13 there are the details of the roof structure, a vaulted scaffolding.

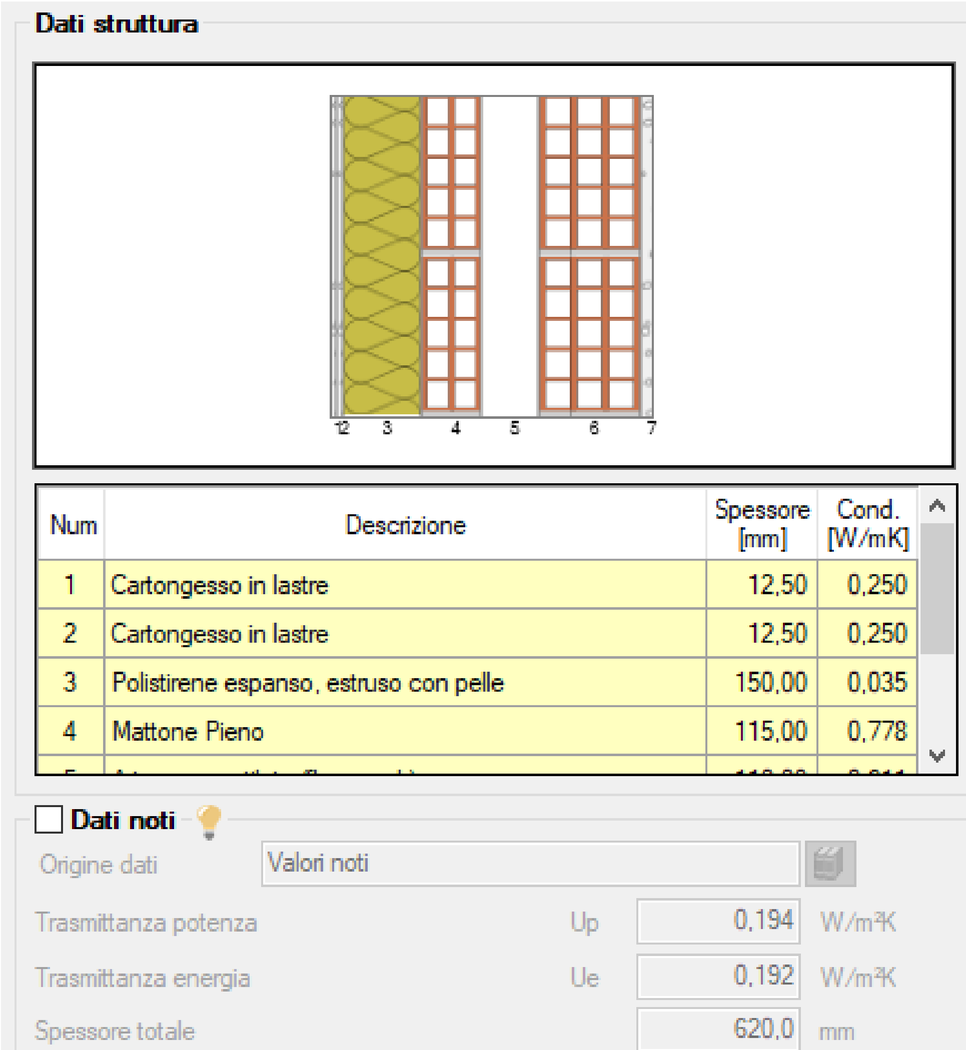
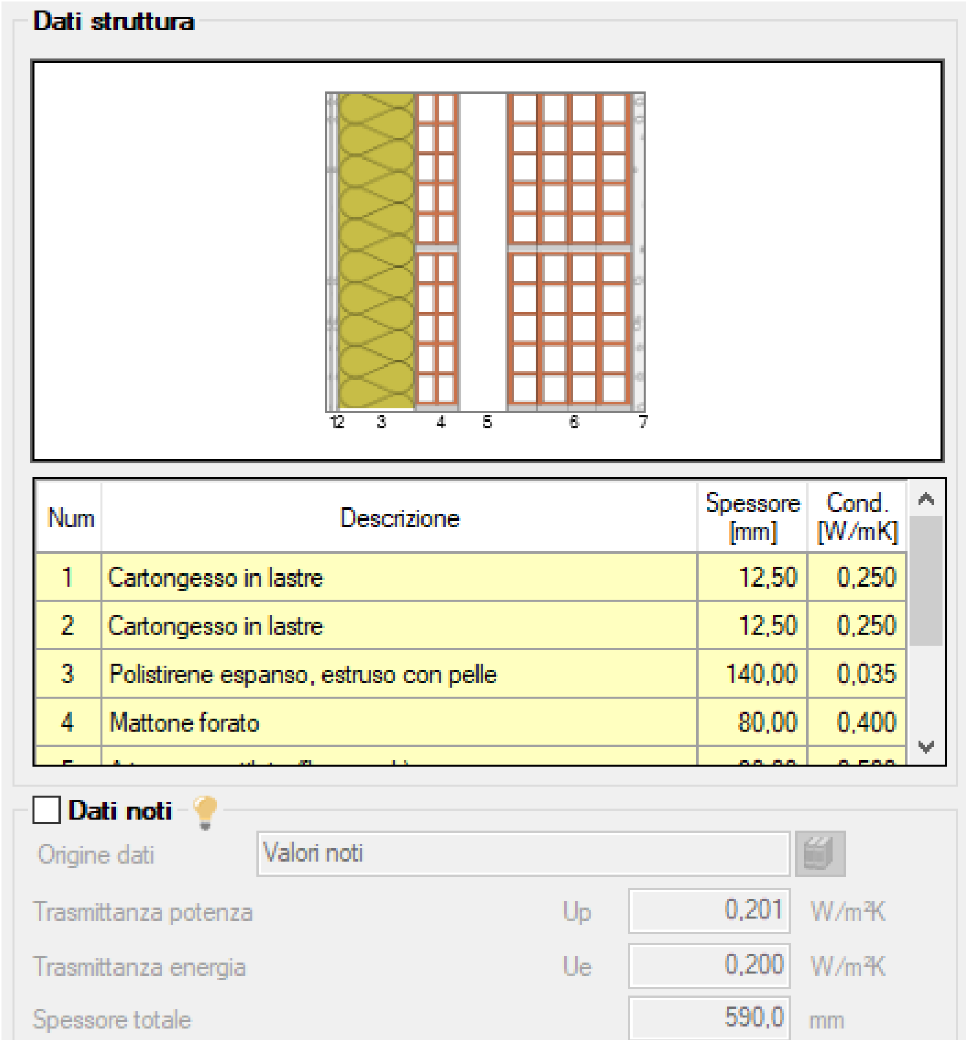


Figure 1.11: Composition and other thermal specifications of the opaque external wall.



Dati struttura

Num	Descrizione	Spessore [mm]	Cond. [W/mK]
1	Cartongesso in lastre	12,50	0,250
2	Cartongesso in lastre	12,50	0,250
3	Polistirene espanso, estruso con pelle	140,00	0,035
4	Mattone forato	80,00	0,400

Dati noti

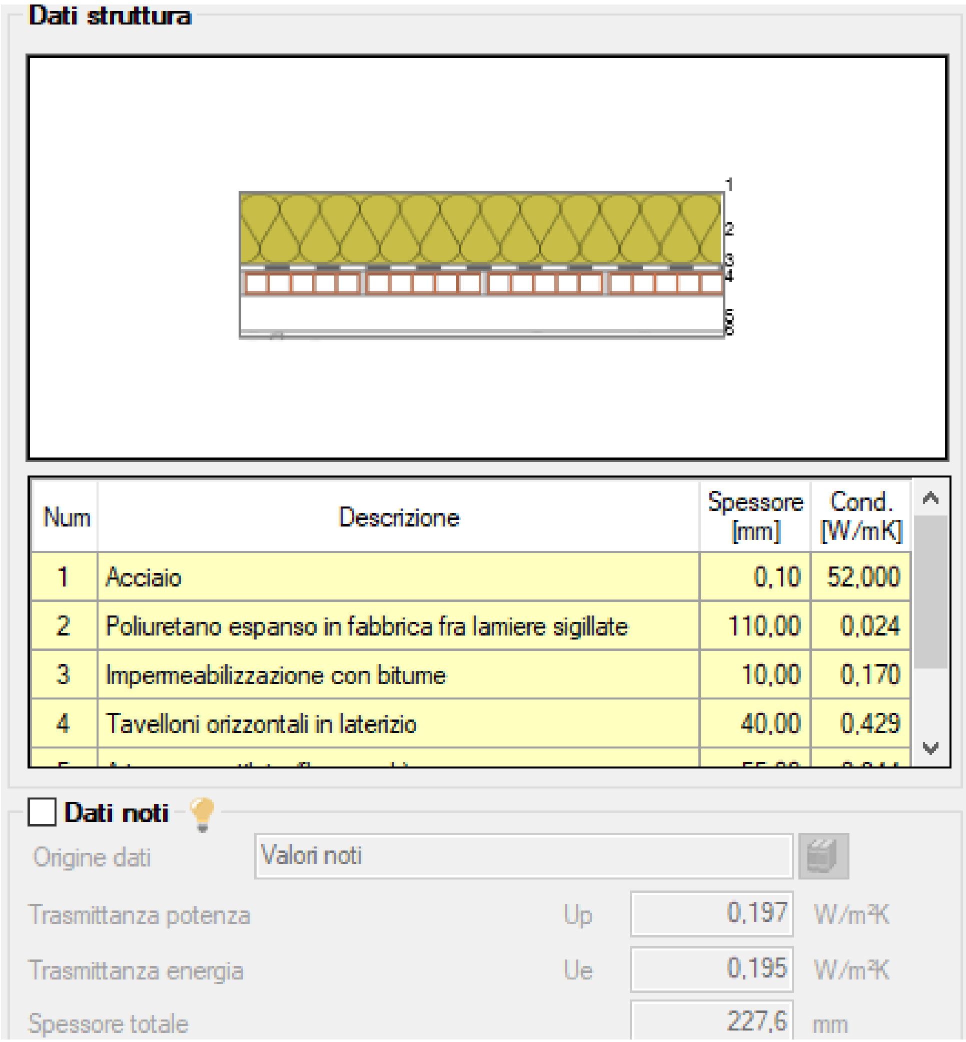
Origine dati: Valori noti

Trasmittanza potenza Up 0,201 W/m²K

Trasmittanza energia Ue 0,200 W/m²K

Spessore totale 590,0 mm

Figure 1.12: Composition and other thermal specifications of the opaque external wall.



Dati struttura

Num	Descrizione	Spessore [mm]	Cond. [W/mK]
1	Acciaio	0,10	52,000
2	Poliuretano espanso in fabbrica fra lamiere sigillate	110,00	0,024
3	Impermeabilizzazione con bitume	10,00	0,170
4	Tavelloni orizzontali in laterizio	40,00	0,429

Dati noti

Origine dati		Valori noti	
Trasmittanza potenza	Up	0,197	W/m²K
Trasmittanza energia	Ue	0,195	W/m²K
Spessore totale		227,6	mm

Figure 1.13: Composition and other thermal specifications of the curved roof slab.

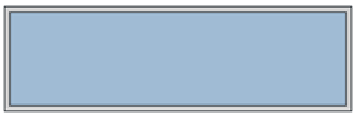
Descrizione della finestra: ***Facciata Continua*** **Codice:** ***W1***

Il serramento è un modulo di facciata continua.

Caratteristiche del serramento

Tipologia di serramento		-	
Classe di permeabilità		*Senza classificazione*	
Trasmittanza termica	U_{cw}	*1,300*	W/m²K
Trasmittanza solo vetro	U_g	*1,300*	W/m²K

Dati per il calcolo degli apporti solari e delle schermature



Emissività	ε	*0,837*	-
Fattore di trasmittanza solare	$g_{gl,n}$	*0,670*	-
Fattore tendaggi (invernale)	$f_{c\ inv}$	*1,00*	-
Fattore tendaggi (estivo)	$f_{c\ est}$	*1,00*	-
Fattore trasmissione solare totale	g_{gl+sh}	*0,300*	-

Caratteristiche delle chiusure oscuranti

Resistenza termica chiusure		*0,00*	m²K/W
f shut		*0,5*	-
Trasmittanza serramento *	$U_{w,e}$	*1,300*	W/m²K

* Valore calcolato considerando l'effetto della chiusura oscurante (UNI EN ISO 10077)

Dimensioni e caratteristiche del serramento

Larghezza	*315,0*	cm
Altezza H	*95,0*	cm

Caratteristiche del telaio

K distanziale	K_d	*0,110*	W/mK
Area totale	A_w	*2,992*	m²
Area vetro	A_g	*2,592*	m²
Area telaio	A_f	*0,400*	m²
Fattore di forma	F_f	*0,87*	-
Perimetro vetro	L_g	*7,800*	m

Caratteristiche del modulo

Trasmittanza termica del modulo	U	*1,300*	W/m²K

Traversi e montanti del modulo di facciata continua

Traversi

Spessore	s_t	*90,0*	cm
Area	A_t	*2,84*	m²

Montanti

Spessore	s_m	*5,0*	cm
Area	A_m	*0,09*	m²

Figure 1.14: Composition and other thermal specifications of the continuous facade.

Continuous facade is composed of double glass with a coupled chassis. Its g-factor g_{glass} is 0.35. Interstitial condensation verification is performed on all the envelope elements with UNI EN ISO 13788:2013[18] and the check is positive for each one.

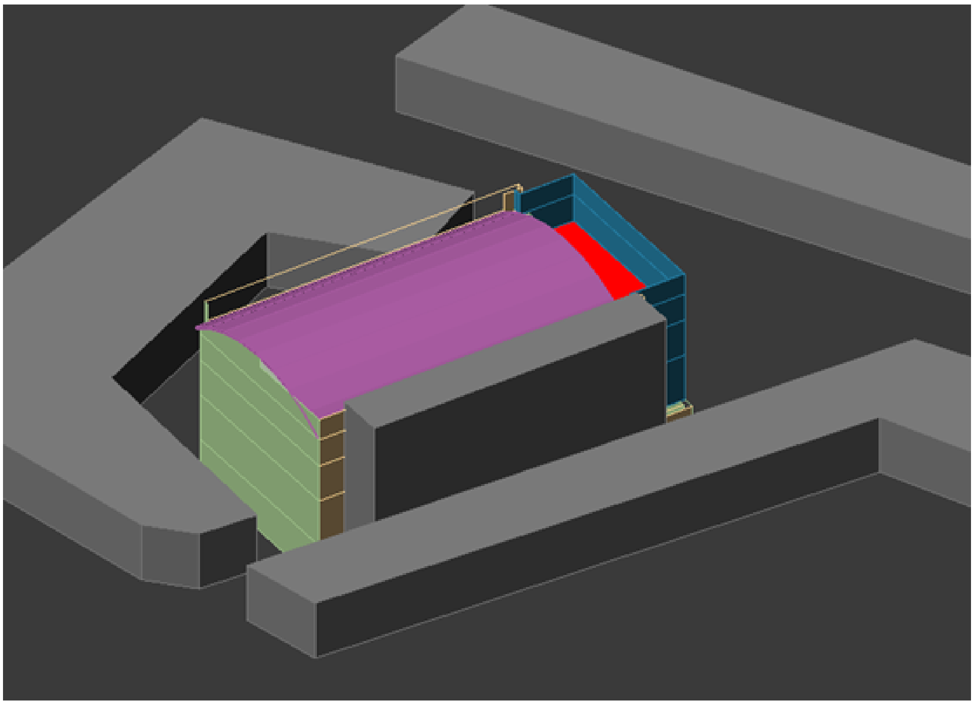


Figure 1.15: 3D model of the building in Edilclima. In purple, the curved roof; in light green, the external wall; in grey, external obstacles; in blue, the continuous glazing facade.

In the next section the calculated heat and cool loads in respect to the area tag are criticized.

1.4. Thermal analysis

1.4.1. Heating loads

To obtain a conservative estimate of the heating load requirements for the building, it is necessary to consider the worst-case scenario by excluding internal heat gains from the calculations. By doing so, the energy required to condition the indoor space will be maximized, providing a more accurate estimate of the heating load. In this chapter, we will analyse the results obtained using the heat gains calculation tool from the Edilclima software, performed considering UNI EN 12831:2018[19]. It considers various factors such as outdoor design conditions and thermal characteristics of the building envelope, intending to understand the heating performance of the building and identify potential areas for improvement. Mechanical ventilation load is not considered because will later be. A correction on the winter powers for intermittent heating is adopted with 25 W/m^2, and furthermore a safety factor of -25% on the global is adopted. As can be seen,the estimate of winter heat loads will be precautionary. The most dispersive envelope element is, as foreseen, the transparent envelope because of its high thermal transmittance. Diagram in figure 1.16 shows the contributions of heat dispersion from the envelope.

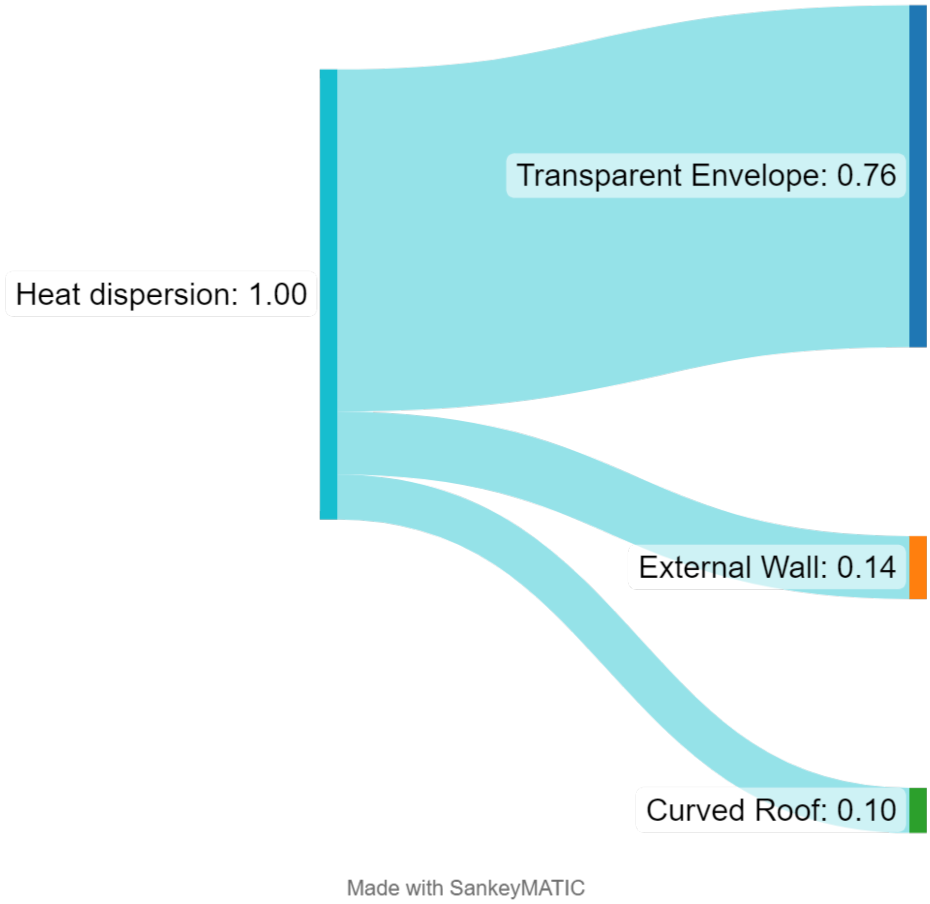


Figure 1.16: Sankey diagram of the heat dispersion of the whole building by component, where 1 corresponds to the total.

1.4.2. Cooling loads

To estimate the cooling requirements of a building, it is important to consider the various sources of heat gain. Among these sources, internal and external solar gains are the most significant contributors to the overall cooling load. Moreover, the impact of humidity on indoor comfort must also be considered, particularly in spaces where moisture generation is high. In this sections, the intention is, for each one of the four zones, to show the peak hour loads by load type. The important assumptions used to run this simulation are:

- Dry bulb temperature setpoint of 26 °C.
- Wet bulb temperature setpoint of 19 °C.

- Peak infiltration rate of 1.0 vol/h.
- Sensible heat gain for each person of 64 W.
- Latent heat gain for each person of 46 W.
- Value of people number determined from architectural layout.
- Heat generated by lighting and/or electrical equipment calculated considering LED fixtures of 7 W/m^2 plus estimated heat gain from other equipment, like computers, divided by the area of the space.

Zone n. 1, Atrium

In figure 1.17 there is a pie chart which detailes the values of the peak hour (4 p.m.) heat loads and the percentage over the total of each contribution. Where:

- Qirr is the load due to irradiation.
- QTr is the load due to heat transmission.
- Qv is the load due to infiltration rate.
- Qc is the load due to internal gains.

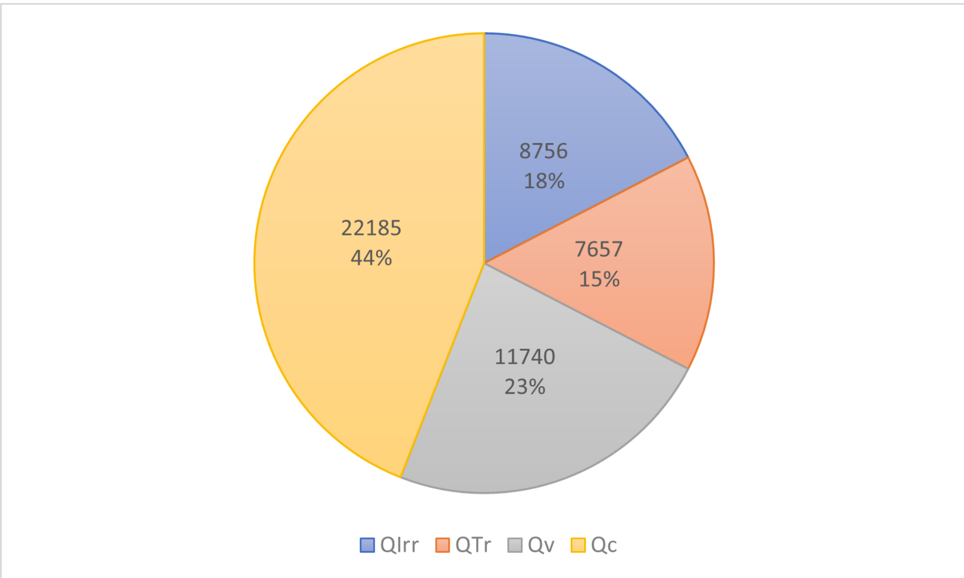


Figure 1.17: Pie chart of the zone n. 1 heat loads divided by source type. Values in W.

The sensible heat factor, from now on abbreviated to SHF, calculated as the ratio of sensible heat load over the total load is 0.71. The major contribution, as can be seen, come from internal gains. Transmission, ventilation and irradiative gain are secondary but not negligible. No corrective observations can be applied as to internal heat gains apart from reducing people number. Air leakage, also called air infiltration, is the unintentional introduction of outside air into the space and it is often associated with envelope elements like windows, doors, wiring, plumbing, and ducts. It plays a non irrelevant role and must be reduced using sealing materials during the construction phase. Transmission can be even reduced by improving opaque and transparent element's transmittance U. Irradiative heat gain could be reduced using proper external or internal shading like louvres or curtains; unluckily this was not the case because of architectural decisions.

Zone n. 2, multipurpose space at first floor

In figure 1.18 there is a pie chart which detailes the values of the peak hour (4 p.m.) heat loads and the percentage over the total of each contribution. The nomenclature used is the same as the previous subsection.

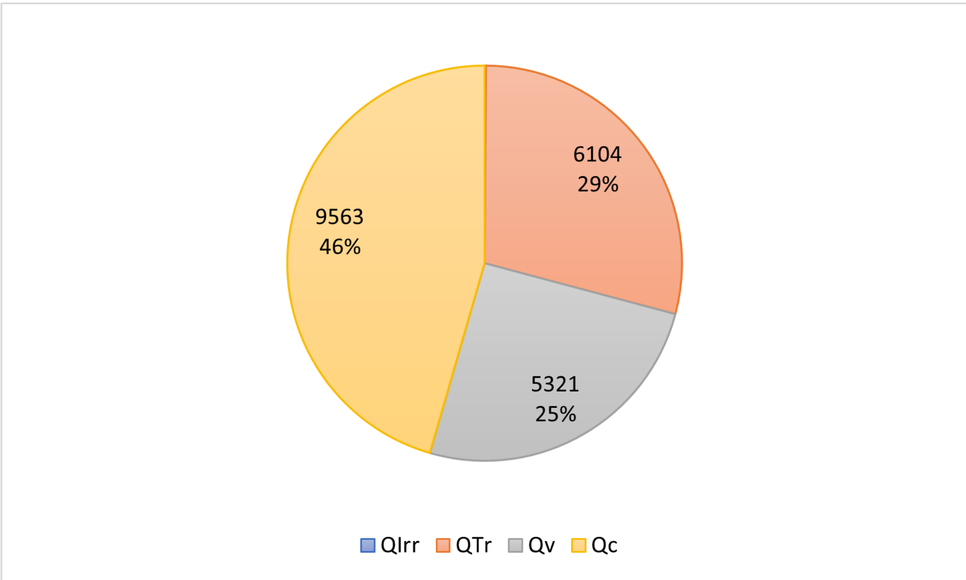


Figure 1.18: Pie chart of the multipurpose area at first floor heat loads divided by source type. Values in W.

The sensible heat factor is 0.72. The considerations done for multipurpose space at second floor written in previous subsection apply here as well. Furthermore, irradiative load is nearly zero because the space is closed and relies on artificial lighting only.

Zone n. 3, multipurpose space at second floor

In figure 1.19 there is a pie chart which detailes the values of the peak hour (4 p.m.) heat loads and the percentage over the total of each contribution. The nomenclature used is the same as the previous subsection.

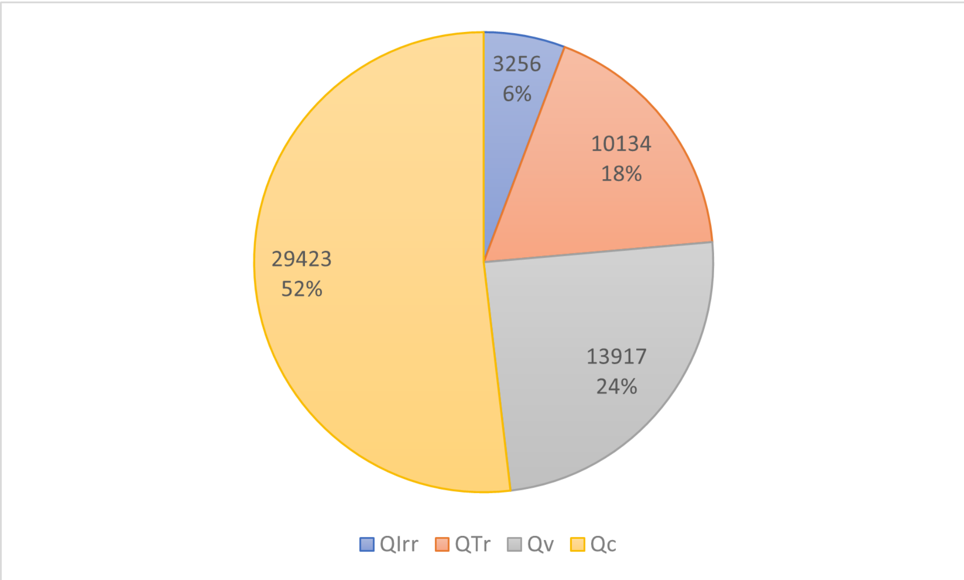


Figure 1.19: Pie chart of the multipurpose area at second floor heat loads divided by source type. Values in W.

The sensible heat factor is 0.70. The three major contribution, as can be seen, come from transmission, ventilation and internal gains. No corrective observations can be applied as to internal heat gains apart from reducing people number. Regarding air infiltration, same reasoning applies as before. Transmission can be even reduced by improving opaque and transparent element's transmittance U. Irradiative heat gain could be reduced using proper external or internal shading like louvres or curtains; unluckily this was not the case because of architectural decisions.

Zone n. 4, auditorium

In figure 1.20 there is a pie chart which detailes the values of the peak hour (4 p.m.) heat loads and the percentage over the total of each contribution. The nomenclature used is the same as the previous subsection.

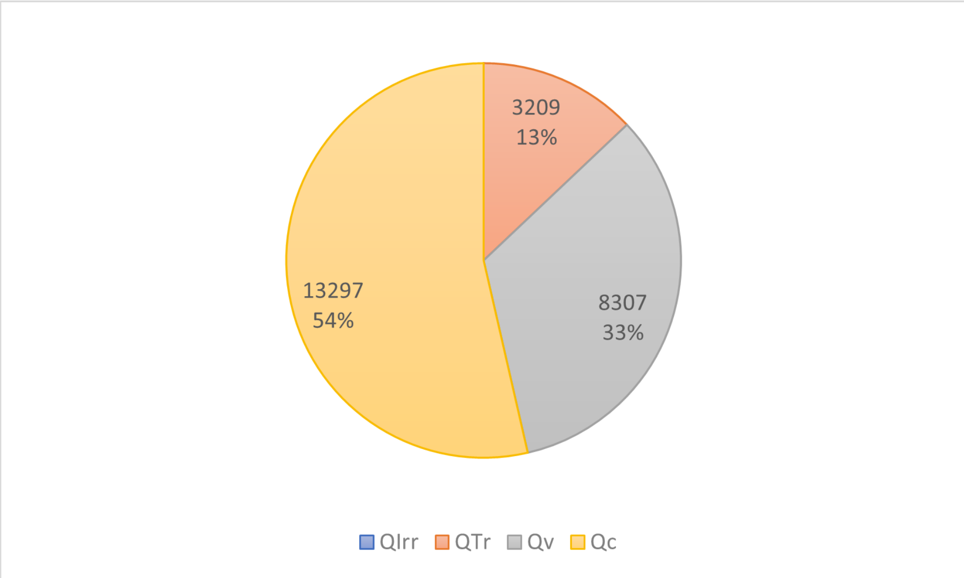


Figure 1.20: Pie chart of the auditorium zone's heat loads divided by source type. Values in W.

The sensible heat factor is 0.60. The three and only contribution, as can be seen, come from transmission, ventilation and internal gains. The considerations expressed in the previous subsections apply here as well.

In table 1.6 there are summarized the heating loads, the sensible cooling loads, and the latent cooling loads for each space.

Zone number	**Space**	Q_{htg} **in** W	$Q_{clg,sens}$ **in** W	$Q_{clg,latent}$ **in** W
1	Ground floor atrium	31600	25000	7800
1	Ground floor bar deposit	500	400	200
1	Ground floor reception deposit	400	450	180
1	Ground floor WC	130	nc	nc
1	Ground floor playroom n. 1	2350	2100	1400
1	Ground floor study room n. 1	2880	2400	1300
1	Ground floor study room n. 2	2800	2100	1300
1	Ground floor meeting room	3110	3600	2400
2	First floor multipurpose area	17670	15300	5900
2	First floor disable WC	400	nc	nc
2	First floor WC	400	nc	nc
3	Second floor multipurpose area	39500	39700	17000
3	Second floor disable WC	300	nc	nc
4	Auditorium technical room	150	250	150
4	Auditorium	18250	14800	11000
Totals		$\approx$ 128440	$\approx$ 106000	$\approx$ 48000

Table 1.6: Thermal loads of the climatized spaces.

Where:

- Q_{htg} is the heating load;
- $Q_{clg,sens}$ is the cooling sensible load;
- $Q_{clg,latent}$ is the cooling latent load;
- *nc* means not conditioned space;
- the first number after the name tag refers to the level of the building.

2 | Ventilation requirements

For each zone, ventilation requirements have been computed, considering the normative UNI 10339 [20]. In particular," prospect III "External air flows in civil use destined buildings", under the category of "Buildings used to recreative activities". For reading spaces, UNI 10339 gives at chapter 9.1.1.1 a method to determine the ventilation requirements.

Condition	Criteria
if $V/n \leq 15$	$Q_{ope} = Q_{op}$ found in prospect III
if $15 \leq V/n \leq 45$	Method B is applied
if $V/n \geq 45$	Method A is applied

Table 2.1: Schematic of the procedure to determine the ventilation requirement.

Where:

- V is the space volume;
- n is the occupancy;
- Q_{ope} is the effective air flow;
- Q_{op} is the air flow found in the prospect III.

Method A expects that $Q_{ope} = 4 \times 10^{-3}\ m^3/s$. Method B is obtained using the following formula:

$$Q_{ope} = Q_{op} + m(V/n - 15) \tag{2.1}$$

where m is $\frac{Q_{opmin} - Q_{op}}{30}$ and where Q_{opmin} is the air flow obtained with method A.

Another criteria to determine outdoor air need comes from ASHRAE 62.1 [6] and in particular the VRP (Ventilation Rate Procedure) that is a prescriptive one. As shown in the introduction, the reference manual published by ASHRAE that may be of interest to the present case study is Standard 62.1 of 2022. Section 6.2 provides a prescriptive method for calculating the minimum outdoor air quantity to achieve acceptable air quality. Specifically,

$$V_b z = R_p \times P_z + R_a \times A_z \tag{2.2}$$

where:

- $V_b z$ represents the air flow rate described above;
- P_z represents the number of people present during the use of that space,
- A_z represents the net occupiable area of that space.
- R_p and R_a are tabular values, which vary from space to space. Table 6-1 provides these values; specifically under the public assembly headings 'Auditorium seating area' and 'Libraries'.

In picture 2.1, the reported scheme outlines the logical path to determine and fix a goal value of airflow.

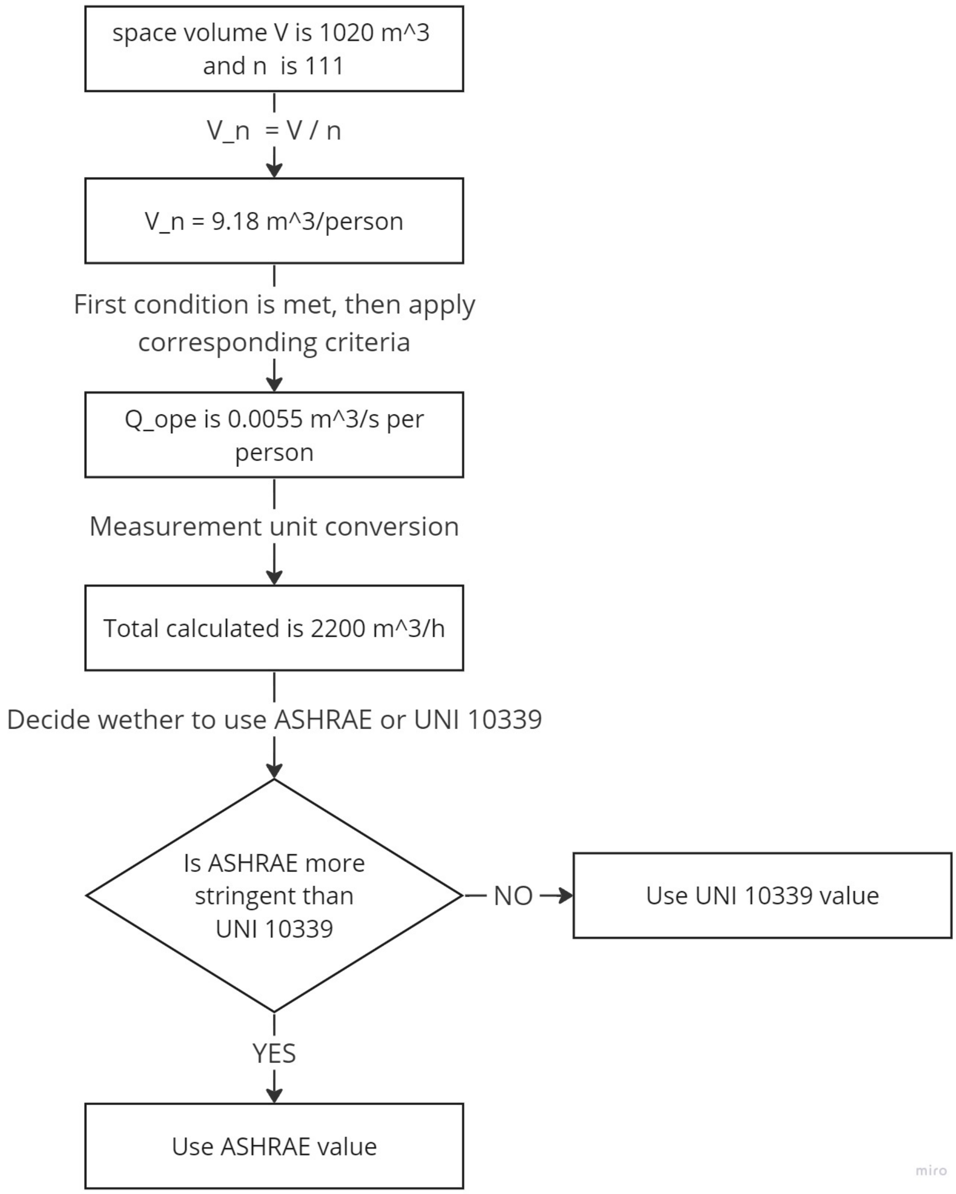


Figure 2.1: Block diagram explaining the criteria used to establish the airflow value.

The procedure is carried over for all the spaces because all of them fall on this category apart from restrooms. In table 2.2 the value used are reported.

		ASHRAE 62.1 values for ventilation assessment						UNI 10339:1995 for ventilation assessment						Result
Zone n.	Space	R_p [L/(s person)]	R_a [L/(s m2)]	P_z [person]	A_z [m2]	V_bz [L/s]	V_bz [m3/h]	V [m3]	n [person]	V_n [m3/person]	Q_op [10^-3 m3/s person]	Q_ope [10^-3 m3/s person]	Q_ope [m3/h]	Worst case scenario
1	Atrium	2.5	0.3	75	245	261	940	1020	75	14	ESTRAZIONI		4080	UNI 10339
1	Playroom	2.5	0.3	16	50	55	198	225	16	14	5.5		317	UNI 10339
1	studyroom #1	2.5	0.3	16	40	52	187	184	16	12	5.5		317	UNI 10339
1	studyroom #2	2.5	0.3	16	40	52	187	174	16	11	5.5		317	UNI 10339
1	meeting room	2.5	0.3	36	50	105	378	235	36	7	5.5		713	UNI 10339
2	multipurp 1st floor	2.5	0.6	58	336	346.6	1248	1110	58	19	METODO B	5.3	1105	ASHRAE 62.1
3	multipurp 2nd floor	2.5	0.6	190	655	868	3125	2900	190	15	5.5		3762	UNI 10339
4	auditorium	2.5	0.3	111	215	342	1231	1731	111	16	METODO B	5.5	2186	UNI 10339

Figure 2.2: Comparative table of Values of outdoor air need according to ASHRAE and UNI 10339 for each space of the building.

Regarding all the restrooms of the building, the need of ventilation consist of air extraction. The value of air to be extracted is determined using, again, either ASHRAE 62.1 or UNI10339. The first one suggests that, for each water closet, a minimum of 90 m^3/h shall be extracted , the second one suggests indicates the following formula: $Q_{extracted} = 8[vol/h] \times V_{space}[m^3]$. All the building's restrooms are subject to the worst rule, which in this case is the ASHRAE 62.1 one.

An additional note about ventilation requirements is about book deposits in libraries; according to UNI 10339, they must comply with 1.5 vol/h extracted. Hence, first floor deposit must face extraction of continuous 150 m^3/h.

2.1. Selection of system type

Once the assessment of building's performance is carried out, system selection is the next step. Here multiple considerations group to establish the system/systems type/s.

The two main macro types of systems available are:

- Primary air and water (and/or freon) terminals, abbreviated to "PA+WT".
- All air system, abbreviated to "AA".
- A combination of the previous two.

The difference between PA+WT and AA systems mainly is the equipment responsible to conditionate the space. In particular, in the first type of system the heating/cooling medium is water (or freon) while in the second type it is air. It can be demonstrated that air is inherently more inefficient than water as energy medium. In facts, air implies large ducts, high parasitic power (energy consumed by fans), and non negligible air leakages. On the other hand AA systems have a quicker response on heat loads change or occupancy change and some other secondary pros like simplified maintenance and typically greater air filtration quality.

Overall, the rapid variation of occupancy, as well as the possibility to prevent virus infections (see CoVid 19) in multipurpose spaces and auditorium are the key factors that leads to this decision of using all air systems to provide cooling, heating and ventilation in very crowded spaces like auditorium and reading areas.

An additional possibility is to use water terminals and/or VRV systems where possible in order to downsize the all air system equipment.

In table 2.2 are summarized the system types used.

Space	System types	Useful effect
Atrium	VRV units + HR for PA	H, C, V, D
Playroom	"	"
Study room 1	"	"
Study room 2	"	"
Meeting room	"	"
Multipurpose space 1st floor	Rooftop + VRV units	H, C, V, D, U
Multipurpose space 2nd floor	"	"
Auditorium	"	"

Table 2.2: Schematic of the procedure to determine the ventilation requirement.

Where:

- The acronym "HR for PA" means Heat Recovery equipment for Primary Air. In apparent contradiction with the statements made before, all air system is not used. This is because of the lack of space technological rooms or on the roofs.
- The acronyms "H, C, V, D, U" are respectively Heating, Cooling, Ventilation, Dehumidification, and Humidification. Dehumidification is performed with the cold battery and the linked reheating is done with the recovery hot battery of the rooftop. Humidification is performed with an accessory from the rooftop manufacturer and is a immersed electrodes humidifier.

2.2. Sizing of all air system

In this section, the aim is to explain the general sizing criteria of the rooftops, and parallel to this, explaining how to choose the temperature of the air supplied to the spaces in the two main seasons. For every zone served by a rooftop, the data used is:

- Primary air required in m^3/h.
- Space heating load in W.
- Space cooling sensible load in W.
- Maximum number of people in the space.

Using psychrometric transformations formulas, and setting supply temperature, volumetric air flow is obtained.

2.2.1. Auditorium space all air system sizing

A starting example is the auditorium space.

Attribute	Value
Primary air required in m^3/h	2200
Space heating load in W	18250
Space cooling sensible load in W	14800
Maximum number of people in the space	111

Table 2.3: Ventilation requirements to satisfy loads in the auditorium.

Considering the heating period, given this set of data:

- Heating load;
- Supply temperature;
- Humidity of air at setpoint condition;
- Specific isobaric heat capacities of air for moist and dry air.

It is possible to estimate the airflow needed to provide that heating effect, using this formula.

$$Q_{heating} = m_{air} \times (C_{p,dry} + C_{p,wet} X_{room}) \times (T_{supply} - T_{heating,indoor}) \tag{2.3}$$

This formula represents a sensible heating process in a psychrometric transformation. The same estimation can be performed to investigate ventilation requirement to satisfy sensible cooling load and separately to satisfy latent cooling load. Formulas are respectively:

$$Q_{cooling,sensible} = m_{air} \times (C_{p,dry} + C_{p,wet} X_{room}) \times (T_{supply} - T_{cooling,indoor}) \tag{2.4}$$

So, considering the setpoint values and a reasonable inlet condition, one solution might be this one. Please refer to table 2.4.

Variable	Value	
Heating supply temperature in °C	28	
Cooling supply temperature in °C	20	
Cooling supply relative humidity in %	55	
Air amount to satisfy heating load in kg/s and m^3/h, respectively.	2.20	6700
Air amount to satisfy sensible cooling load in kg/s and m^3/h, respectively.	2.31	6900

Table 2.4: Ventilation requirements to satisfy loads in the auditorium/cinema.

Since the most ventilation demanding load is the cooling sensible, the maximum air flow at design condition is 6900 m^3/h. The same reasoning can be done with all the other air volumes served by all-air systems. Regarding the humidification load in winter, the method used to assess its magnitude has been to consider the worst condition. That is when a minimum of 20 people is present in the auditorium. A resting person emits approximately 50 g/h of water vapor. Hence, calculating the humidification need with the following formula:

$$m_{air} \times (X_{room} - X_{outdoorair}) = m_{watervapour} \tag{2.5}$$

and subtracting the free humidification from people inside the space, the following results are obtained.

Variable	Value
X_{room} in g/kg	7.25
$X_{outdoorair}$ in g/kg	2.5
$m_{outdoorair}$ in kg/h	2640
$m_{watervapour}$ need kg/h	12.5
$m_{watervapour}$ from people in kg/h	1

Table 2.5: Humidification load calculation for the auditorium space.

Hence, the humidification load is estimated to be 11.5 kg/h.

2.2.2. First floor multipurpose space all air system sizing

Regarding the one at first floor, in table 2.6 are summarized the data input used.

Attribute	Value
Primary air required in m^3/h	1250
Space heating load in W	17670
Space cooling sensible load in W	15300
Maximum number of people in the space	58

Table 2.6: Ventilation requirements to satisfy loads in the first floor multipurpose space.

Feeding the same formulas as before and making the same underlying assumptions, results are:

Variable	Value	
Heating supply temperature in °C	28	
Cooling supply temperature in °C	16	
Air amount to satisfy heating load in kg/s and m^3/h, respectively.	2.17	6500
Air amount to satisfy sensible cooling load in kg/s and m^3/h, respectively.	1.43	4300

Table 2.7: Ventilation requirements to satisfy loads in the multipurpose space at first floor.

Here, instead the limiting ventilation rate is 6500 m^3/h, deriving from heating load. The same calculation is performed here with the only difference about the minimum number of people, considered to be null. Since during the library activity the space must be conditionate even if no person is present, differently from the auditorium, a space used only on scheduled events. Feeding the same equation used before with the data of multipurpose space at first floor, the following is obtained.

Variable	Value
X_{room} in g/kg	7.25
$X_{outdoorair}$ in g/kg	2.5
$m_{outdoorair}$ in kg/h	1500
$m_{watervapour}$ need kg/h	7.2
$m_{watervapour}$ from people in kg/h	0

Table 2.8: Humidification load calculation for the first floor multipurpose space.

Hence, the humidification load is estimated to be 7.2 kg/h.

2.2.3. Second floor multipurpose space all air system sizing

Regarding the one at first floor, in table 2.9 are summarized the data input used.

Attribute	Value
Primary air required in m^3/h	3760
Space heating load in W	10000
Space cooling sensible load in W	25000
Maximum number of people in the space	190

Table 2.9: Ventilation requirements to satisfy loads in the second floor multipurpose space.

Feeding the same formulas as before and making the same underlying assumptions, results are:

Variable	Value	
Heating supply temperature in °C	30	
Cooling supply temperature in °C	16	
Air amount to satisfy heating load in kg/s and m^3/h, respectively.	0.98	2940
Air amount to satisfy sensible cooling load in kg/s and m^3/h, respectively.	2.34	7050

Table 2.10: Ventilation requirements to satisfy loads in the multipurpose space at second floor.

Again, the limiting ventilation rate is 7050 m^3/h, deriving from cooling load. The magnitude of cooling sensible load and heating load are relatively big. That's why only half will be served by all-air system and the other by some VRF technology; the chosen VRF must be capable of perform 25 kW cooling power and the remaining 25 kW will be satisfied by the all-air system. The same applies in winter, where has been chosen to let the rooftop provide 10kW of heating power while the other part is provided by VRV internal units.

2.2.4. Rooftop overview

The chosen equipment is a Clivet rooftop of the serie named "CSNX-XHE2". It can produce heat and cool thanks to its two refrigerant circuits equipped with scroll compressors. Here is reported a functional scheme to help the reader understand the stages that the air is going to be subject to.

Figure 2.3: CLIVET Rooftop serie CSNX presentation.

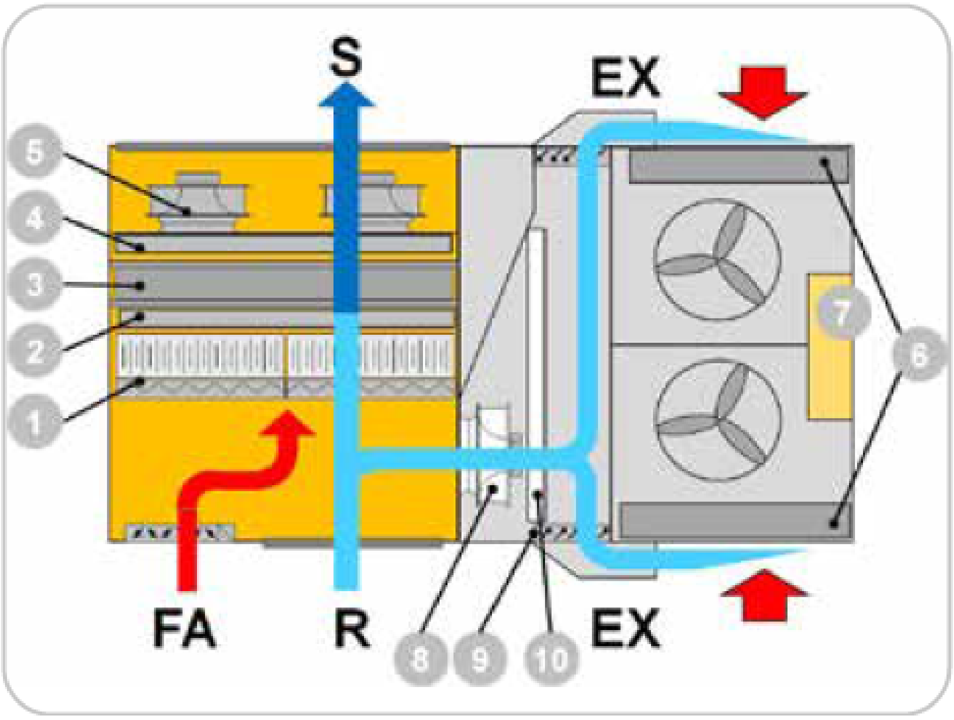


Figure 2.4: Functional scheme of the chosen rooftop.

Where:

- 1 is the position of the first and second stage of filtration.
- 2 is an optional electrical resistance heat exchanger.
- 3 is the main heat exchanger for the refrigerant circuit (evaporator or condenser depending on the functioning).
- 4 is the post treatment heat exchanger.
- 5 is the ventilating section of supply and return.
- 6 is the source side heat exchanger.
- 7 is the electrical unit.
- 8 is the expulsion fan.
- 9 is the over-pressure damper.
- 10 is the heat recovery heat exchanger.
- FA, S, EX, and R are respectively fresh air, supply air, expulsion air, and return air.

To assess the indoor air quality problem related to air recirculation, especially in public assembly spaces, a portion of the air treated by the rooftop will be fresh outdoor air. Coupled with this setting, the filtration (see point n. 2 in figure 2.4) will be provided with two filters: the first is a F7 (ISO16890:2017 [21] ePM1 55%) and the second is a FIFD (ISO16890:2017 [21] ePM1 90%). The indicated filters will eventually block particulate matter, virus, and bacteria.

The selection is performed considering the data sheets and the space needs. In particular, the engineer must perform the following considerations to make sure that the machine will work once installed.

- Heating and cooling need at worse conditions external conditions can be supplied.
- Primary air setpoint is actually met.

Regarding the first point, it must be specified that rooftop must be able to meet not only space loads but also the thermal power to cool or heat the fresh outdoor air (in facts, it has not been considered previously in thermal loads), both in cooling and heating. The following assumption are considered to perform calculations.

- As calculation method, a sensible heating psychrometric transformation is considered.
- Outdoor air temperature in the worst condition is -5 °C with absolute humidity of 2.5 g/kg in winter.
- Outdoor air temperature in the worst condition is 35 °C with absolute humidity of 14.25 g/kg in summer.
- density of air equal to 1.2 kg/m^3.
- a immersed electrodes humidifier is positioned between point n.4 and point n. 5 of figure 2.4.

The calculation sheet is here presented, where the humidification with water vapour, for the sake of simplicity, is considered a iso-thermal transformation in the psychrometric diagram.

Coupling the following results with technical data sheet, the following sizes of rooftop have been chosen.

Space	Rooftop size
Auditorium	16.4
1st floor multipurpose space	15.3
2nd floor multipurpose space	16.4

Table 2.11: Rooftop model for each space.

2.3. Sizing and design of VRV system

As discussed in the previous chapters, in zone n. 1 and zone n.3, direct expansion circuits will provide heating and cooling. In zone n. 1, primary air is provided by heat recovery device. Ducted internal units will be mounted to satisfy heating and cooling need. The chosen model for the use is "PEFY-VMM-E" from the manufacturer "Mistubishi".

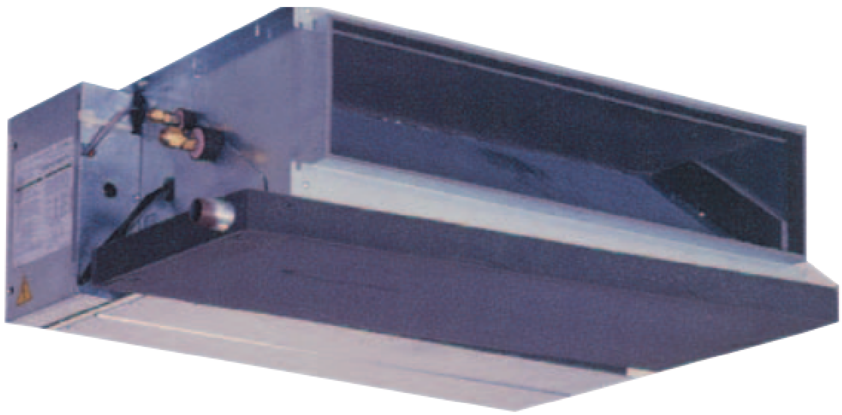


Figure 2.5: Ductable fan coil from Mitsubishi.

The size selection of the ducted internal unit must meet thermal performance requirements (see table 1.6) and to do so, the technical documentation from the manufacturer has been investigated.

1. SPECIFICATIONS

R410A Data G4

Model			PEFY-P20VMM-E	PEFY-P25VMM-E	PEFY-P32VMM-E	PEFY-P40VMM-E
Power source			1-phase 220-240V 50Hz			
Cooling capacity (Nominal)	*1	kW	2.2	2.8	3.6	4.5
	*1	kcal / h	1,900	2,400	3,100	3,900
	*1	Btu / h	7,500	9,600	12,300	15,400
	*2	kcal / h	2,000	2,500	3,150	4,000
	Power input	kW	0.15	0.15	0.17	0.19
	Current input	A	0.73	0.73	0.81	0.92
Heating capacity (Nominal)	*3	kW	2.5	3.2	4.0	5.0
	*3	kcal / h	2,200	2,800	3,400	4,300
	*3	Btu / h	8,500	10,900	13,600	17,100
	Power input	kW	0.15	0.15	0.17	0.19
	Current input	A	0.73	0.73	0.81	0.92
External finish			Galvanized			
External dimension H x W x D		mm	295 x 815 x 700	295x815x700	295x815x700	295x935x700
		in.	11-5/8" x 32-1/8" x 27-9/16"	11-5/8" x 32-1/8" x 27-9/16"	11-5/8" x 32-1/8" x 27-9/16"	11-5/8" x 36-13/16" x 27-9/16"
Net weight		kg (lb)	27 (60)	27 (60)	27 (60)	33 (73)
Heat exchanger			Cross fin (Aluminum fin and copper tube)			
FAN	Type x Quantity		Sirocco fan x 1	Sirocco fan x 1	Sirocco fan x 1	Sirocco fan x 2
	External static press.	Pa	30- 50- 100	30- 50- 100	30- 50- 100	30- 50- 100
		mmH_2O	3.1- 5.1- 10.2	3.1- 5.1- 10.2	3.1- 5.1- 10.2	3.1- 5.1- 10.2
	Motor type		1-phase induction motor			
	Motor output	kW	0.075	0.075	0.075	0.075
	Driving mechanism		Direct-driven by motor			
	Airflow rate (Low-Mid-High)	m^3 / min	6.0 - 7.2 - 8.5	6.0 - 7.2 - 8.5	7.5-9.0-10.5	10.0-12.0-14.0
		L / s	100 - 120 - 142	100 - 120 - 142	125-150-175	167-200-233
		cfm	212 - 254 - 300	212 - 254 - 300	265-318-371	353-424-494
Noise level (Low-Mid-High) (measured in anechoic room)		dB <A>	27-30-32	27-30-32	28-32-35	31-34-37
Insulation material			Polyethylene foam, Urethane foam			
Air filter			PP honeycomb fabric (washable)			
Protection device			Fuse			
Refrigerant control device			LEV			
Connectable outdoor unit			R410A, R407C, R22 CITY MULTI			
Diameter of refrigerant pipe	Liquid (R410A)	mm (in.)	ø6.35 (ø1/4") Flare	ø6.35 (ø1/4") Flare	ø6.35 (ø1/4") Flare	ø6.35 (ø1/4") Flare
	Liquid (R22, R407C)	mm (in.)	ø6.35 (ø1/4") Flare	ø6.35 (ø1/4") Flare	ø6.35 (ø1/4") Flare	ø6.35 (ø1/4") Flare
	Gas (R410A)	mm (in.)	ø12.7 (ø1/2") Flare	ø12.7 (ø1/2") Flare	ø12.7 (ø1/2") Flare	ø12.7 (ø1/2") Flare
	Gas (R22, R407C)	mm (in.)	ø12.7 (ø1/2") Flare	ø12.7 (ø1/2") Flare	ø12.7 (ø1/2") Flare	ø12.7 (ø1/2") Flare
Field drain pipe size		mm (in.)	Socket(R1)+R1(PT1)[O.D. 34mm(1-11/32")] or O.D.32mm(1-1/4") if using optional Drain pump.			
Drawing	External		IU - W65-3948			
	Wiring		IU - W65-3958			
	Refrigerant cycle		-			
Standard attachment	Document		Installation Manual, Instruction Book			
	Accessory					

Figure 2.6: Internal units data sheet from Mitsubishi.

The selection procedure led to this arrangement.

Space tag	Indoor Unit tag	Nominal heating capacity [W]	Nominal total cooling capacity [W]
Playroom	PEFY-P40VMM-E	5000	4500
Study room n. 1	PEFY-P40VMM-E	5000	4500
Study room n. 2	PEFY-P40VMM-E	5000	4500
Meeting Room	PEFY-P63VMM-E	8000	7100
Auditorium technical space	PKFY-10VLM-E	1400	1200
Atrium	2 x PEFY-P125VMM-E	16000	14000
Multip. space 2nd floor	2 x PEFY-P125VMM-E	16000	14000

Table 2.12: Table of VRV internal units' capacities by space.

The scheme that follows in figure ?? explains both the criteria and the positioning of heat recovery equipment.

2.3.1. Cross flow heat exchanger coupling with internal units

The spaces ventilation need is provided with the model "LGH-xxxRVXT-E" by Mitsubishi.

Figure 2.7: Recuperator LGH-RVXT-E by Mitsubishi.

There are mainly two choices about the coupling of internal units with heat recovery equipment.

- 1. The recuperator supply line has got a dedicated air terminal to reach ambient.
- 2. The recuperator supply line merges with the supply line of the internal units.

The following two figures explain the above mentioned choices n.1 and n.2.

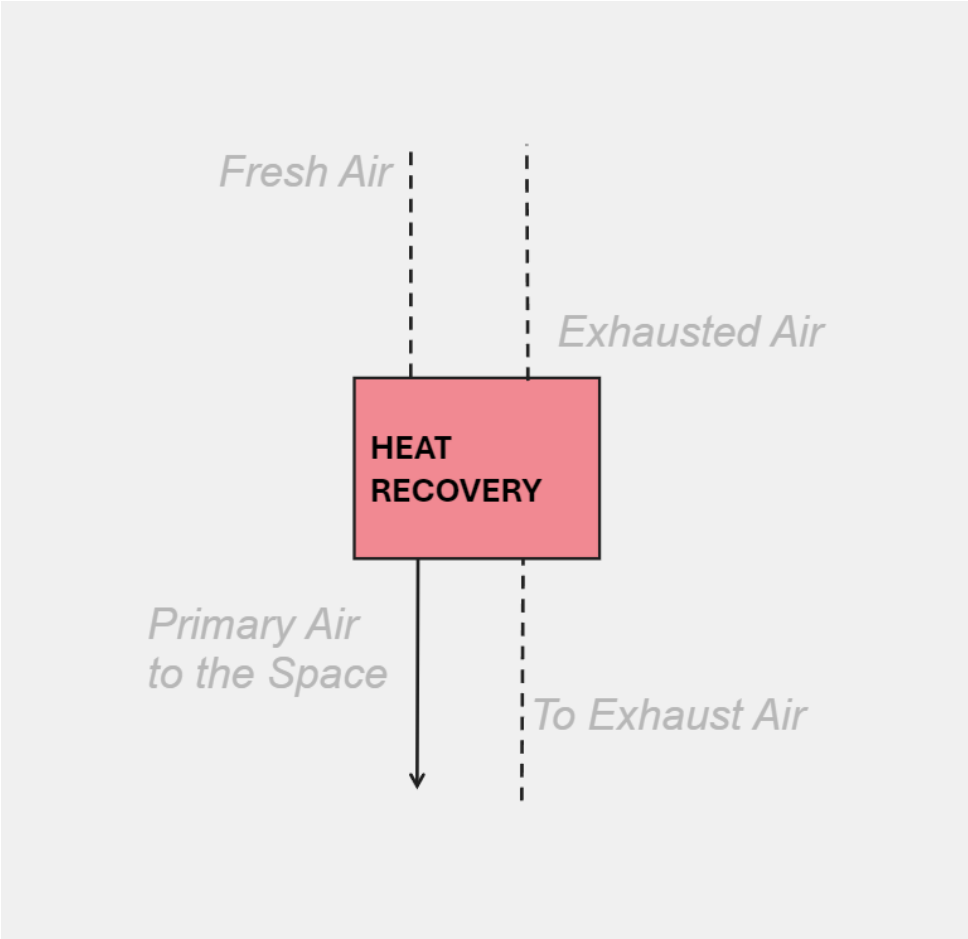


Figure 2.8: Recuperator supply line connection to the space; option n.1.

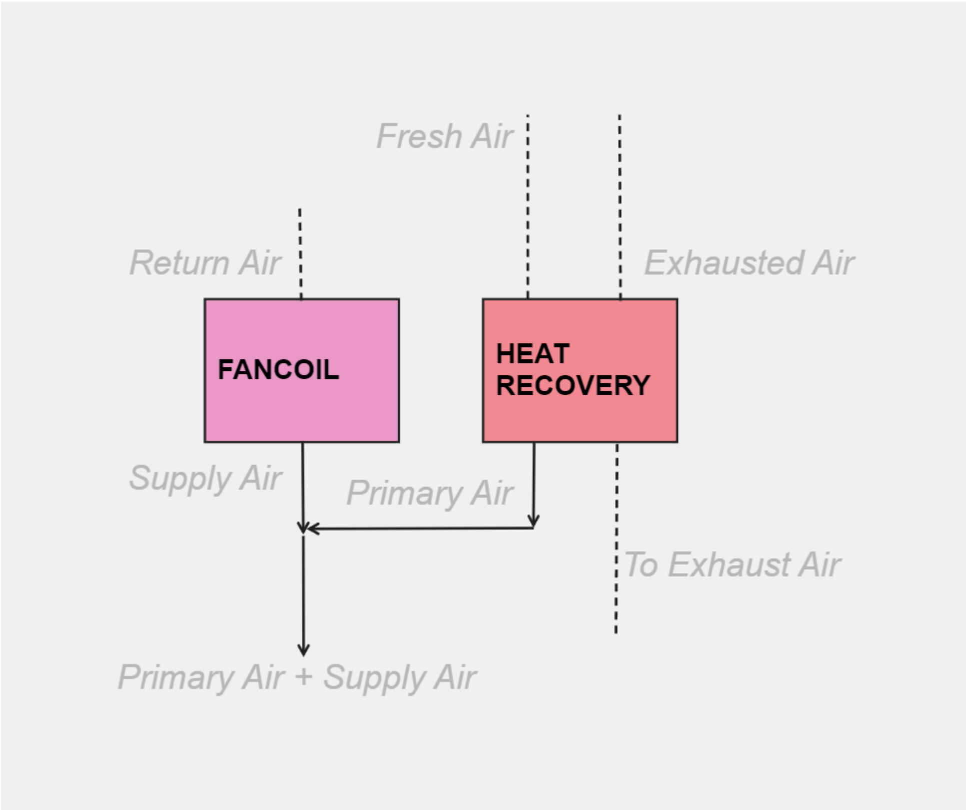


Figure 2.9: Recuperator supply line connection to the space; option n.2.

The logic of work of the option n.2 is this: fresh outdoor air enters the heat recovery machine, exchanges sensible heat with extracted air from the treated space and enters supply ducts of internal units' supply line. There, fresh air mixes with treated air from internal units and reaches diffusers.

Option n.2 can enhance thermal comfort avoiding cold/hot drafts, that is why the project woll contemplate this arrangement.

2.3.2. External units

This solution is composed of an outdoor air-to-air unit together with a controller that sorts the fluid refrigerant (liquid or gas) to the various indoor units. External units chosen are PURY-P500 and PURY-P400, the first will serve each internal unit but the ones at second floor and the latter the missing.

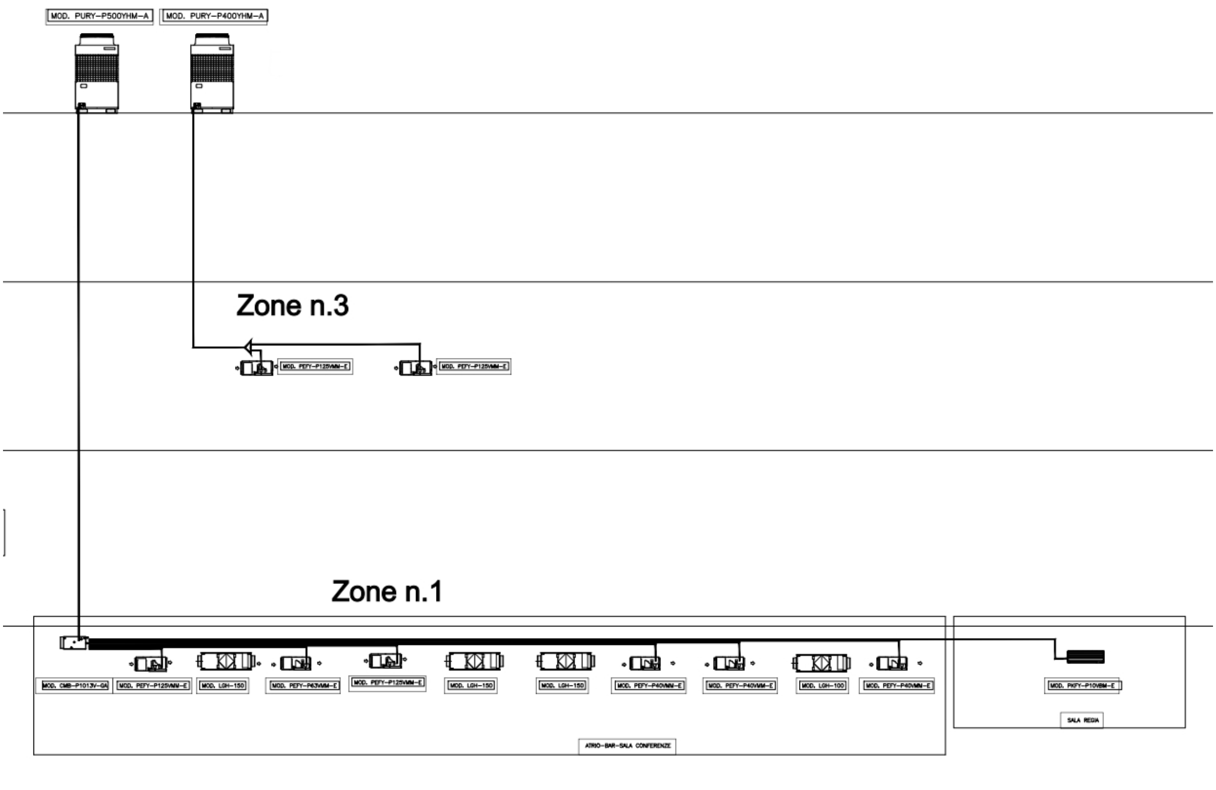


Figure 2.10: Altimetric Scheme of the VRV circuit.

Figure 2.11: Outdoor unit PURY from Mitsubishi installed in the last floor.

3 | Materials selection and aeraulic system dimensioning principles

Ductwork will be in galvanized steel sheet as there is no need to use antimicrobial materials. Key dimensioning criteria to ensure performance and occupants comfort are velocity, noise, and pressure losses. Since libraries must perform under 35 dB(A), the typical range of velocities is from 4 m/s to 5 m/s. The higher the velocity, the higher the pressure losses and the higher the noise generated, and vice versa. Regarding noise, duct thermal insulation acts as noise attenuation. Designing ducts to minimize air turbulence helps to reduce noise generation. Regarding pressure losses a limit of 0.8 Pa/m is set to maintain efficient airflow and prevent compromised system performance. There are two different dimensioning criteria: one that sizes ducts to maintain a pressure loss in a small range of values and the other that sizes ducts to maintain velocity in a small range of values. The former has the advantage of keeping the circuit balanced but at the expense of increased material usage while the latter has the advantage of controlling noise but may require additional measures to ensure proper system balancing. For the dimensioning process, Caleffi guide has been used[22], which provides comprehensive guidelines and charts for designing ductworks. The calculations and charts in the guide are based on air properties at 20°C and sea level conditions, ensuring accurate and reliable results. Considering both supply and return ductwork and considering a maximum pressure loss of 0.8 Pa/m, the limiting factor is only the velocity. As shown in the next figure, the red line is always above the other two diagonal lines.

The above-mentioned diagram gives the diameter of a circular duct and therefore another diagram must be used to get the equivalence[22]. Furthermore, good practice is to avoid sharp turns. Hence, where possible, soft turns must be used. An example in the following picture. Pressure drops calculation is performed to check if the designed ductwork is compatible with the equipment attached to it. As an example, indoor unit's fan can have a limit on the net static pressure that they can give, or rooftops for auditorium could not be able to supply the desired amount of air because of the same reason. Calculation method used is the one reported in the "Caleffi" guide mentioned above, which estimates

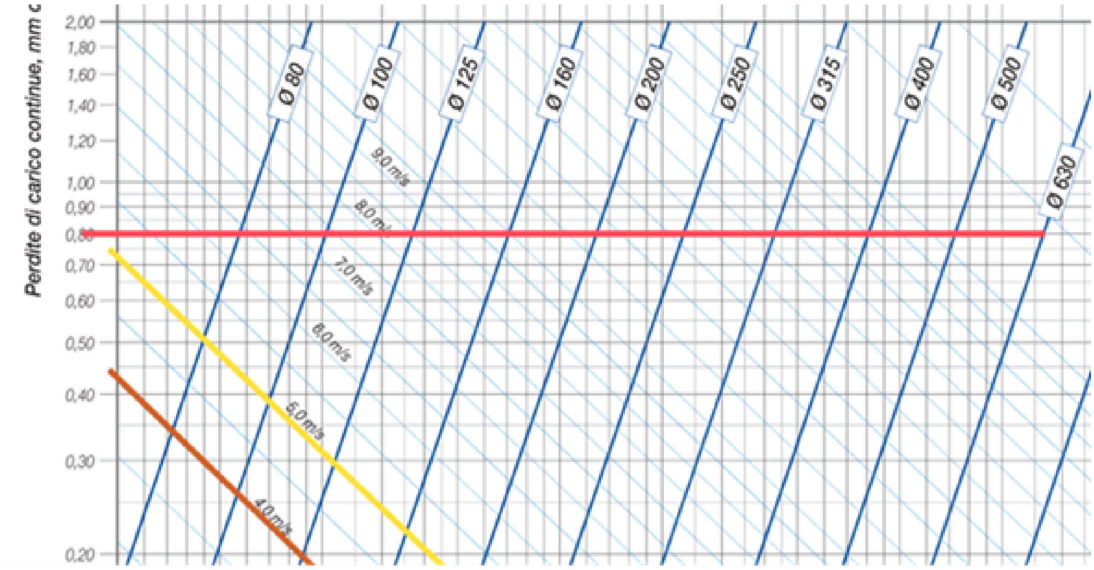


Figure 3.1: Pressure losses diagram for smooth ducts , sea level, and 20 °C.

continuous and localized pressure drops using practical tables and diagrams.

Canali rettangolari: diametri equivalenti per la determinazione delle perdite di carico continue

a, b = dimensioni rettangolo/quadrato, mm *$Ø_e$ = diametro equivalente, mm* *f = fattore correttivo velocità*

b	a	100	150	200	250	300	350	400	450	500	550	600	650	700	750	800	a	b
100	$Ø_e$	109	133	152	169	183	195	207	217	227	236	245	253	261	268	275	$Ø_e$	100
	f	0,94	0,93	0,91	0,89	0,87	0,86	0,84	0,82	0,81	0,80	0,79	0,77	0,76	0,75	0,74	f	
150	$Ø_e$	133	164	189	210	229	245	260	274	287	299	310	321	331	341	350	$Ø_e$	150
	f	0,93	0,94	0,93	0,92	0,91	0,90	0,89	0,87	0,86	0,85	0,84	0,83	0,82	0,81	0,80	f	
200	$Ø_e$	152	189	219	244	266	286	305	321	337	352	365	378	391	402	414	$Ø_e$	200
	f	0,91	0,93	0,94	0,94	0,93	0,92	0,91	0,90	0,89	0,88	0,87	0,86	0,86	0,85	0,84	f	
250	$Ø_e$	169	210	244	273	299	322	343	363	381	398	414	429	443	457	470	$Ø_e$	250
	f	0,89	0,92	0,94	0,94	0,94	0,93	0,93	0,92	0,91	0,90	0,90	0,89	0,88	0,87	0,87	f	
300	$Ø_e$	183	229	266	299	328	354	378	400	420	439	457	474	490	506	520	$Ø_e$	300
	f	0,87	0,91	0,93	0,94	0,94	0,94	0,93	0,93	0,92	0,92	0,91	0,90	0,90	0,89	0,89	f	
350	$Ø_e$	195	245	286	322	354	383	409	433	455	477	496	515	533	550	567	$Ø_e$	350
	f	0,86	0,90	0,92	0,93	0,94	0,94	0,94	0,93	0,93	0,93	0,92	0,92	0,91	0,91	0,90	f	
400	$Ø_e$	207	260	305	343	378	409	437	464	488	511	533	553	573	592	609	$Ø_e$	400
	f	0,84	0,89	0,91	0,93	0,93	0,94	0,94	0,94	0,94	0,93	0,93	0,92	0,92	0,92	0,91	f	
450	$Ø_e$	217	274	321	363	400	433	464	492	518	543	567	589	610	630	649	$Ø_e$	450
	f	0,82	0,87	0,90	0,92	0,93	0,93	0,94	0,94	0,94	0,94	0,93	0,93	0,93	0,92	0,92	f	
500	$Ø_e$	227	287	337	381	420	455	488	518	547	573	598	622	644	666	687	$Ø_e$	500
	f	0,81	0,86	0,89	0,91	0,92	0,93	0,94	0,94	0,94	0,94	0,94	0,93	0,93	0,93	0,93	f	
550	$Ø_e$	236	299	352	398	439	477	511	543	573	601	628	653	677	700	722	$Ø_e$	550
	f	0,80	0,85	0,88	0,90	0,92	0,93	0,93	0,94	0,94	0,94	0,94	0,94	0,94	0,93	0,93	f	
600	$Ø_e$	245	310	365	414	457	496	533	567	598	628	656	683	708	732	755	$Ø_e$	600
	f	0,79	0,84	0,87	0,90	0,91	0,92	0,93	0,93	0,94	0,94	0,94	0,94	0,94	0,94	0,93	f	
650	$Ø_e$	253	321	378	429	474	515	553	589	622	653	683	711	737	763	787	$Ø_e$	650
	f	0,77	0,83	0,86	0,89	0,90	0,92	0,92	0,93	0,93	0,94	0,94	0,94	0,94	0,94	0,94	f	
700	$Ø_e$	261	331	391	443	490	533	573	610	644	677	708	737	765	792	818	$Ø_e$	700
	f	0,76	0,82	0,86	0,88	0,90	0,91	0,92	0,93	0,93	0,94	0,94	0,94	0,94	0,94	0,94	f	
750	$Ø_e$	268	341	402	457	506	550	592	630	666	700	732	763	792	820	847	$Ø_e$	750
	f	0,75	0,81	0,85	0,87	0,89	0,91	0,92	0,92	0,93	0,93	0,94	0,94	0,94	0,94	0,94	f	
800	$Ø_e$	275	350	414	470	520	567	609	649	687	722	755	787	818	847	875	$Ø_e$	800
	f	0,74	0,80	0,84	0,87	0,89	0,90	0,91	0,92	0,93	0,93	0,93	0,94	0,94	0,94	0,94	f	
850	$Ø_e$	282	359	424	482	534	582	626	668	706	743	778	811	842	872	901	$Ø_e$	850
	f	0,74	0,79	0,83	0,86	0,88	0,89	0,91	0,92	0,92	0,93	0,93	0,93	0,94	0,94	0,94	f	
900	$Ø_e$	289	367	435	494	548	597	643	686	726	763	799	833	866	897	927	$Ø_e$	900
	f	0,73	0,79	0,82	0,85	0,87	0,89	0,90	0,91	0,92	0,92	0,93	0,93	0,93	0,94	0,94	f	
950	$Ø_e$	295	376	445	506	561	612	659	703	744	783	820	855	889	921	952	$Ø_e$	950
	f	0,72	0,78	0,82	0,85	0,87	0,88	0,90	0,91	0,92	0,92	0,93	0,93	0,93	0,94	0,94	f	
1000	$Ø_e$	301	384	454	517	574	626	674	719	762	802	840	876	911	944	976	$Ø_e$	1000
	f	0,71	0,77	0,81	0,84	0,86	0,88	0,89	0,90	0,91	0,92	0,92	0,93	0,93	0,93	0,94	f	
1100	$Ø_e$	313	399	473	538	598	652	703	751	795	838	878	916	953	988	1.022	$Ø_e$	1100
	f	0,70	0,76	0,80	0,83	0,85	0,87	0,88	0,89	0,90	0,91	0,92	0,92	0,93	0,93	0,93	f	
1200	$Ø_e$	324	413	490	558	620	677	731	780	827	872	914	954	993	1.030	1.066	$Ø_e$	1200
	f	0,69	0,74	0,79	0,82	0,84	0,86	0,87	0,89	0,90	0,90	0,91	0,92	0,92	0,93	0,93	f	
1300	$Ø_e$	334	426	506	577	642	701	757	808	857	904	948	990	1.031	1.069	1.107	$Ø_e$	1300
	f	0,67	0,73	0,77	0,80	0,83	0,85	0,86	0,88	0,89	0,90	0,90	0,91	0,92	0,92	0,92	f	
1400	$Ø_e$	344	439	522	595	662	724	781	835	886	934	980	1.024	1.066	1.107	1.146	$Ø_e$	1400
	f	0,66	0,72	0,76	0,79	0,82	0,84	0,86	0,87	0,88	0,89	0,90	0,91	0,91	0,92	0,92	f	
1500	$Ø_e$	353	452	536	612	681	745	805	860	913	963	1.011	1.057	1.100	1.143	1.183	$Ø_e$	1500
	f	0,65	0,71	0,75	0,79	0,81	0,83	0,85	0,86	0,87	0,88	0,89	0,90	0,91	0,91	0,92	f	
1600	$Ø_e$	362	463	551	629	700	766	827	885	939	991	1.041	1.088	1.133	1.177	1.219	$Ø_e$	1600
	f	0,64	0,70	0,74	0,78	0,80	0,82	0,84	0,85	0,87	0,88	0,89	0,89	0,90	0,91	0,91	f	
1700	$Ø_e$	371	475	564	644	718	785	849	908	964	1.018	1.069	1.118	1.164	1.209	1.253	$Ø_e$	1700
	f	0,64	0,69	0,74	0,77	0,79	0,81	0,83	0,85	0,86	0,87	0,88	0,89	0,89	0,90	0,91	f	
1800	$Ø_e$	379	485	577	660	735	804	869	930	988	1.043	1.096	1.146	1.195	1.241	1.286	$Ø_e$	1800
	f	0,63	0,69	0,73	0,76	0,79	0,81	0,82	0,84	0,85	0,86	0,87	0,88	0,89	0,90	0,90	f	
1900	$Ø_e$	387	496	590	674	751	823	889	952	1.012	1.068	1.122	1.174	1.224	1.271	1.318	$Ø_e$	1900
	f	0,62	0,68	0,72	0,75	0,78	0,80	0,82	0,83	0,85	0,86	0,87	0,88	0,88	0,89	0,90	f	
2000	$Ø_e$	395	506	602	688	767	840	908	973	1.034	1.092	1.147	1.200	1.252	1.301	1.348	$Ø_e$	2000
	f	0,61	0,67	0,71	0,74	0,77	0,79	0,8	0,83	0,84	0,85	0,86	0,87	0,88	0,89	0,89	f	
2200	$Ø_e$	410	525	625	715	797	874	945	1.013	1.076	1.137	1.195	1.251	1.305	1.356	1.406	$Ø_e$	2200
	f	0,60	0,66	0,70	0,73	0,76	0,78	0,80	0,81	0,83	0,84	0,85	0,86	0,87	0,88	0,88	f	

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Figure 3.2: Rectangular to circular duct conversion table.

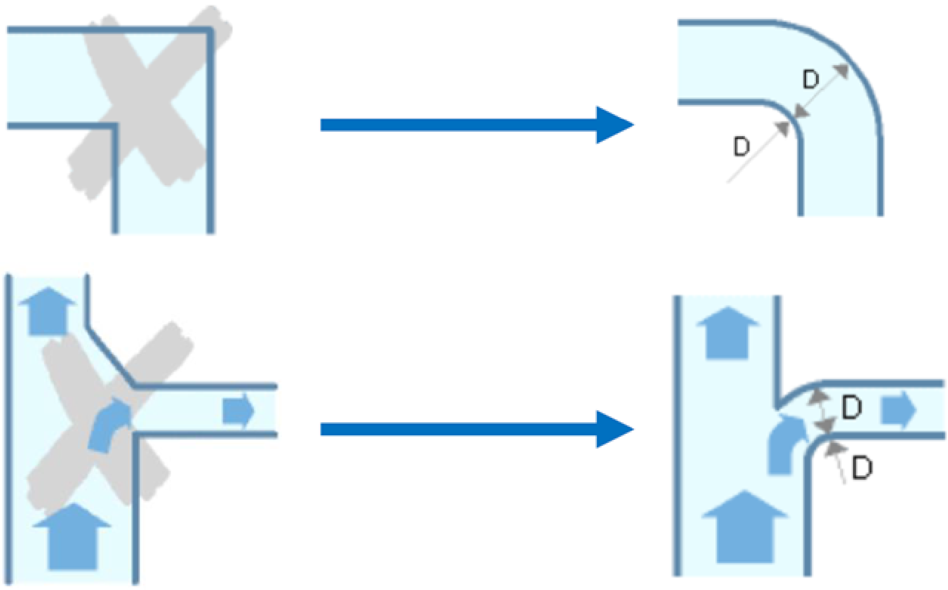

Figure 3.3: Figure that shows common solutions to avoid pressure losses in fluid deviations.

4 | Sizing

In this chapter, dimensioning procedure and other choices regarding air terminals are explored.

4.1. Zone n.1 space dimensioning

This section explain the dimensioning criteria of the spaces in zone n.1

4.1.1. Playroom and adjacent reading room

Both in playroom and reading room, heating or cooling is performed with VRV internal units (see figure 2.5). On the other hand, for these tow spaces, ventilation is performed with one heat recovery equipment. Both ventilation needs are of 317 $m3/h$ of fresh air, hence the total is 634 $m3/h$. Looking at the datasheet of "LGH-RVXT-E", the model that best fits the job is "LGH-150RVXT-E" running at a specific power curve called "SP3", that according to the static pressure-volumetric flow rate diagram, can ensure 150 Pa on the supply side and 125 Pa on the return one at about 650 m^3/h.

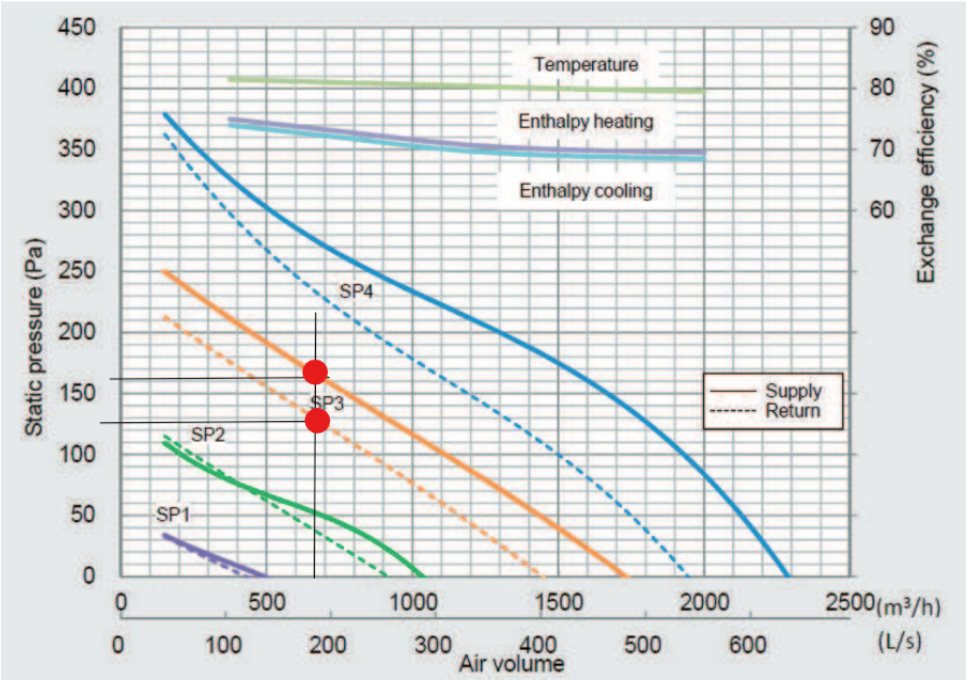

Figure 4.1: Recuperator size "150" curve for thermal exchange efficiency, and power.

One might want to check that there is a pressure gradient that allows fresh outdoor air from heat recovery device to flow towards diffusers, otherwise some air will flow backwards. This check is performed here as follows. Internal unit is designed to work at maximum velocity when starts again from an inactivity period.

Both the spaces are provided with internal unit "PEFY-P40" model, for which the net static pressure is, according to datasheets, 100 Pa at design volumetric flow rate. So, the heat recovery device is correctly sized. A calibration damper is put to regulate the airflow when testing the built system.

The sizing of the ducts of the ductable internal unit is performed under the assumption of:

- Full speed air flow rate.
- Duct must bring both primary air and conditionate air.
- Air velocity inside ducts is maximum 5 m/s.

At maximum speed of the internal unit, the airflow is 840 m^3/h and considering the other 317 m^3/h by the heat recovery device, the maximum airflow in the main supply duct is therefore 1157 m^3/h. The dimensions of the main supply duct are obtained using the conservation of mass, yielding 350mm x 250 mm. The main supply duct divides in two branches each one bringing 680 m3/h to the space. The dimensions are obtained with the same reasoning, maintaining the same height, yielding 200mm x 250mm. They reduce in section while they bring air to the space.

The air diffusion inside the auditorium is realized as in the following scheme in figure 4.16.

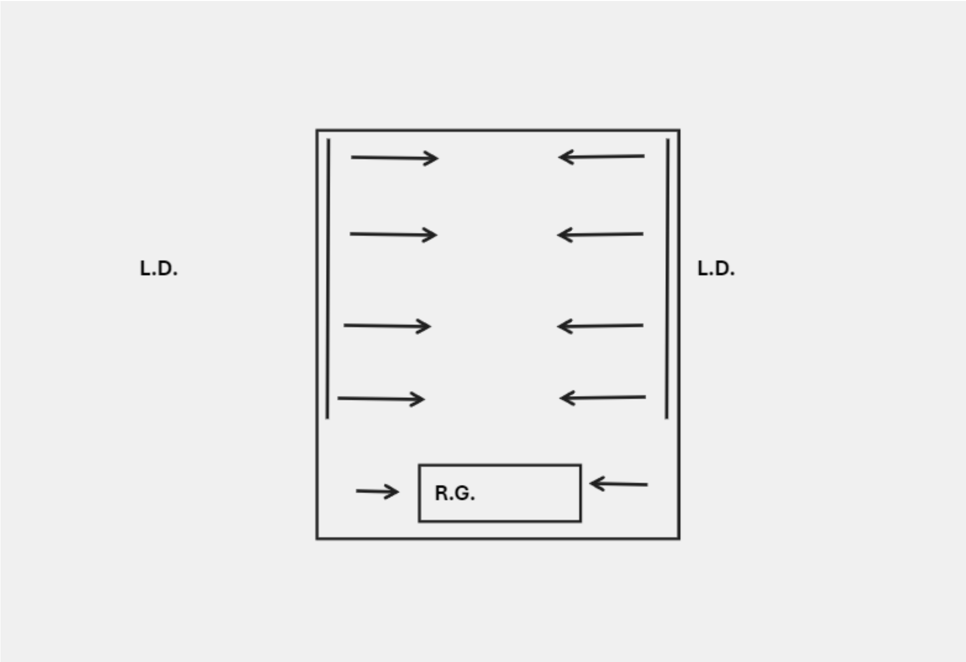

Figure 4.2: Air diffusion scheme in playroom and reading space.

Where:

- L.D. indicates the used Linear Diffusers.
- R.G. indicates the Return Grilles.

Terminals are linear diffusers placed on the side of the room. In this case, each branch can host six meters of linear diffusers, hence each meter must bring 113 m^3/h. Model selected is "FLH10" from "Europair", which according to datasheets, has got:

- A value of noise criterion NC = 15.
- A throw of 3.7 meters at 0.25 m/s.

Given this data, the diffuser must be oriented with a vertical throw so that it does not affect the velocity parameter on the human comfort.

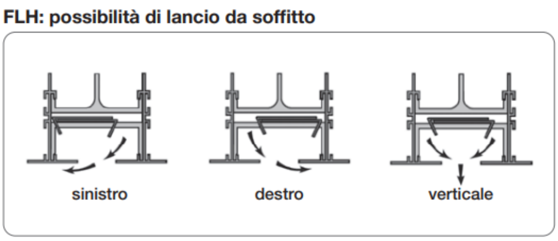

Figure 4.3: Enter Caption

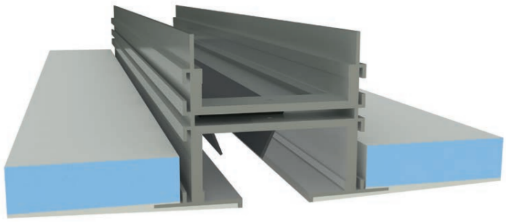

Figure 4.4: Linear diffusers from Europair.

Regarding the return air terminals, since internal unit at full speed crosses the maximum noise level, reaching 37 dB(A), and since a cut in the false ceiling to gather return air would transmit the noise and the vibrations to the space, an aphonic return grille is able to reduce the noise. The following image clarifies the model used.

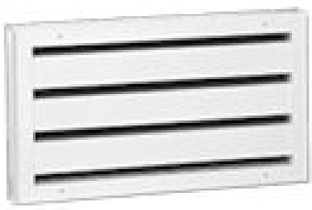

Figure 4.5: Render of the aphonic grille used in the auditorium.

The adjacent study hall shares the same exact principles, and casually, the same dimensioning of air terminals. The following picture shows the mechanical layout.

4.1.2. Meeting room and adjacent reading space

Here, as discussed in previous sections, two internal units will provide cooling and heating effect. Ventilation rate is ensured with the heat recovery equipment. The ventilation

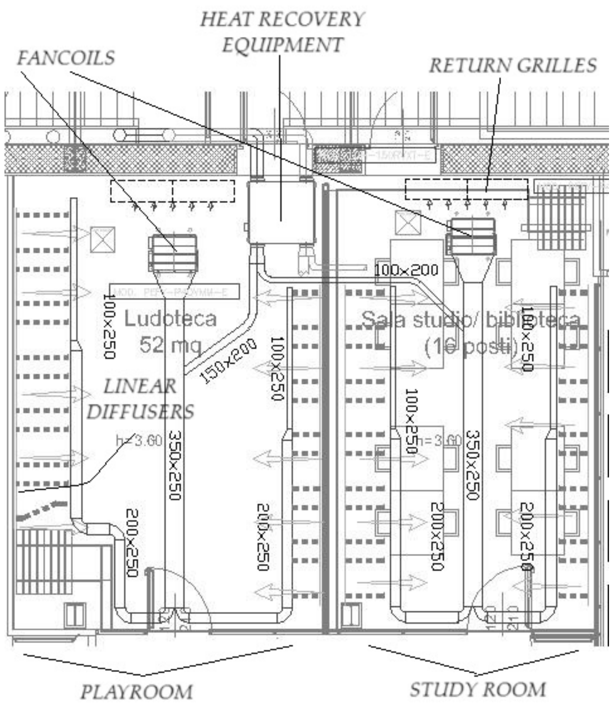

Figure 4.6: Mechanical Layout of Playroom and Reading room

consists of providing fresh air for 712 m3/h to the meeting room and 317 m3/h to the adjacent study hall. The total is, therefore, 1029 m3/h. Looking at the datasheet of “LGH-RVXT”, the model that best fits the job is “LGH-150RVXT-E" running at “SP3”, that according to the static pressure-volumetric flow rate diagram, can ensure 115 Pa on the supply side and 75 Pa on the return one. One might want to check that there is a pressure gradient that allows fresh outdoor air from heat recovery device to flow towards diffusers, otherwise some air will flow backwards. This check is performed here. Internal unit is designed to work at maximum velocity when restarted. For PEFY-P40 and PEFY-P63 the net static pressure is, according to datasheets, 100 Pa. So, the heat recovery device is correctly sized. A calibration damper is put to regulate the airflow when testing the built system. Air diffusion logic is the same of the previous subsection, and the same is true for the air terminals. The only exception is for the meeting room, in which the linear diffuser is FLH20 and double slot (each meter of this linear diffuser must bring 310 m3/h to the space).

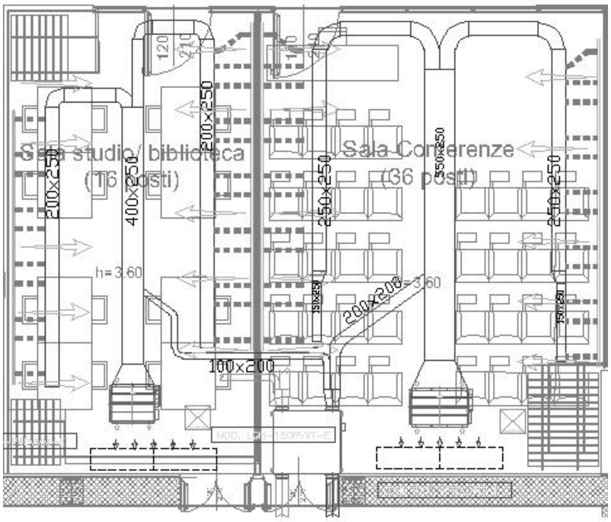

Figure 4.7: Mechanical Layout of the meeting room and the adjacent reading room.

4.1.3. Atrium space

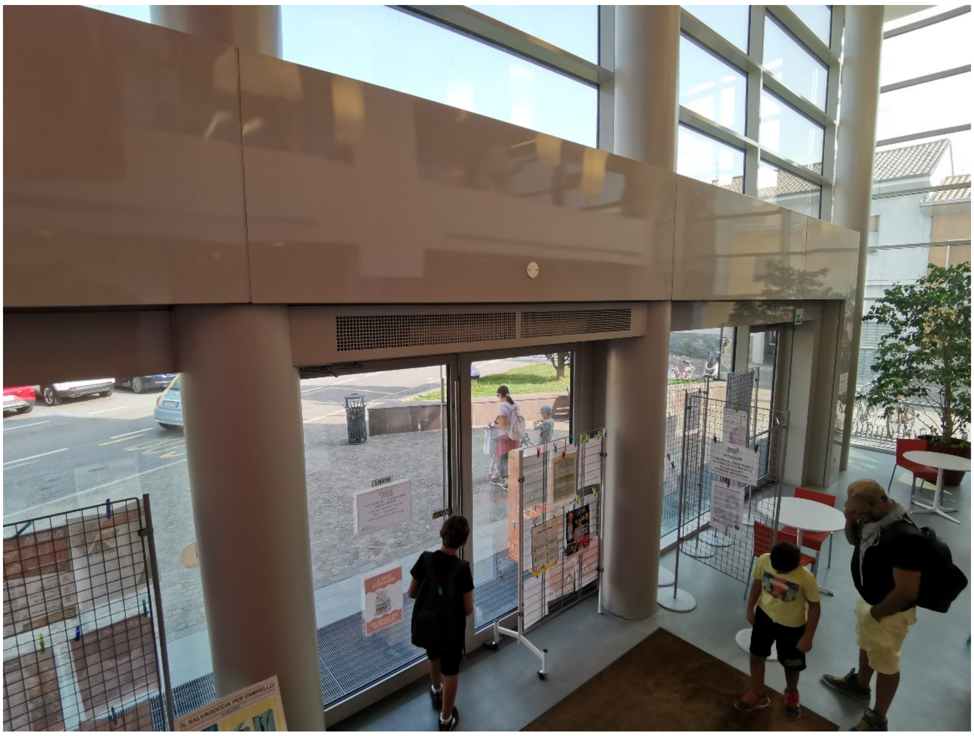

Figure 4.8: Photo of the air blade in the atrium.

Figure 4.9: Linear diffuser in the atrium.

Figure 4.10: Another photo of the diffuser in the atrium.

As discussed before, two internal units powered by the main external one will provide heating and cooling effect. Ventilation is provided by the two heat recovery machine. It will supply fresh air to the space that will mix with treated air from internal units. Both internal units are positioned in in the false ceiling at ground floor. Air terminals are linear diffusers as the ones used in playroom but this time they are vertically mounted and the throw is not horizontal but pointed at 45° towards the floor.

Regarding air flow calculations, cafeteria internal unit ductwork must provide at maximum 4400 m3/h (sum of primary air and airflow at maximum velocity), which according to the same criteria used before, can be brought by an 850mm x 300 mm duct. Same applies for the other internal unit. In this space there is no need to keep sound pressure level below 35 dB(A), so there is no need for aphonic grilles that usually come for a greater cost. Ducts dimensions are calculated according to the previous criteria, i.e. mass conservation law.

4.2. Zone n.2 dimensioning: first floor multipurpose space

Figure 4.11: Air diffuser at first floor space.

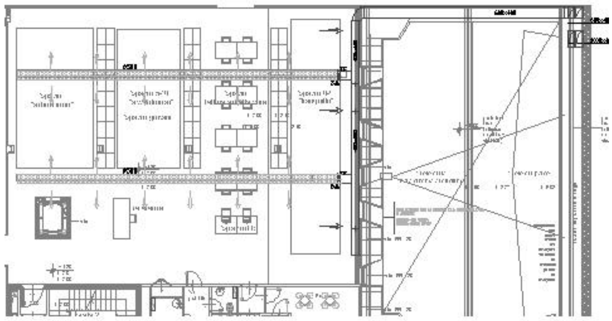

Figure 4.12: Mechanical Layout of the first floor multipurpose space.

Supply and return ductwork will pass through the mechanical and electrical shaft behind the stage of the auditorium/cinema and will reach the studied space. Air terminals will be micro-perforated ducts in red fabric. As previously calculated, the maximum airflow that ductwork must bring is about 4300 m^3/h.

Consider fixing one dimension of the main duct to 400 mm due to architectural constraints of the shaft, then 600 mm is the other dimension calculated according to mass balance equation keeping 5 m/s. According to the manufacturer data sheet of the fabric micro-perforated duct, two main ducts of the diameter of 500 mm can ensure diffusion and low noise.

4.3. Zone n.3 dimensioning: second floor multipurpose space

Figure 4.13: Photo of the second floor.

In the worst condition, 7050 m^3/h is the airflow that flows in the ductwork serving the second floor. According to this data and keeping one dimension of the main duct fixed to 400 mm for the same reason of the previous section, then the other dimension must be 1000 mm to reach at maximum 5 m/s inside the rectangular duct. The main duct will not run in the same mechanical and electrical shaft that hosts ductwork serving first floor spaces and auditorium spaces. The main duct splits into two secondary ducts, one letting 4700 m^3/h flow with rectangular ducts with dimension 700 mm x 400 mm and the other 2350 m^3/h. Again, according to the manufacturer datasheet of the fabric micro-perforated duct, three main ducts of the diameter of 500mm can ensure diffusion and low noise.

4.4. Zone n.4 dimensioning: auditorium

Figure 4.14: Photo of the built auditorium.

The air coming from the rooftop reaches the space through a insulated galvanized duct running in the mechanical/electrical shaft present in the back of the auditorium's stage. On the other hand, the return air from the space reaches back the rooftop via another insulated galvanized duct line running in the same shaft. Regarding the connection of ductwork from the shaft to the rooftop unit, a vibration dumping joint reduces the vibrations transmitted to rigid rectangular ducts. The dimensions of supply and return air from and to the rooftop will be done using the equation of conservation of mass in a duct:

$$\frac{Q[m^3/h]}{3600} = v[m/s] \times A[m^2] \tag{4.1}$$

For the auditorium Q is 7000 m^3/h, and fixing as velocity 5 m/s, the obtained area is 0.388 m^2. Considering the assumption of having one side of the duct of 400 mm, the other dimension is easily obtained: 1000 mm.

Figure 4.15: Photo of the auditorium's stage.

The air diffusion inside the auditorium is realized as in the following scheme in figure 4.16.

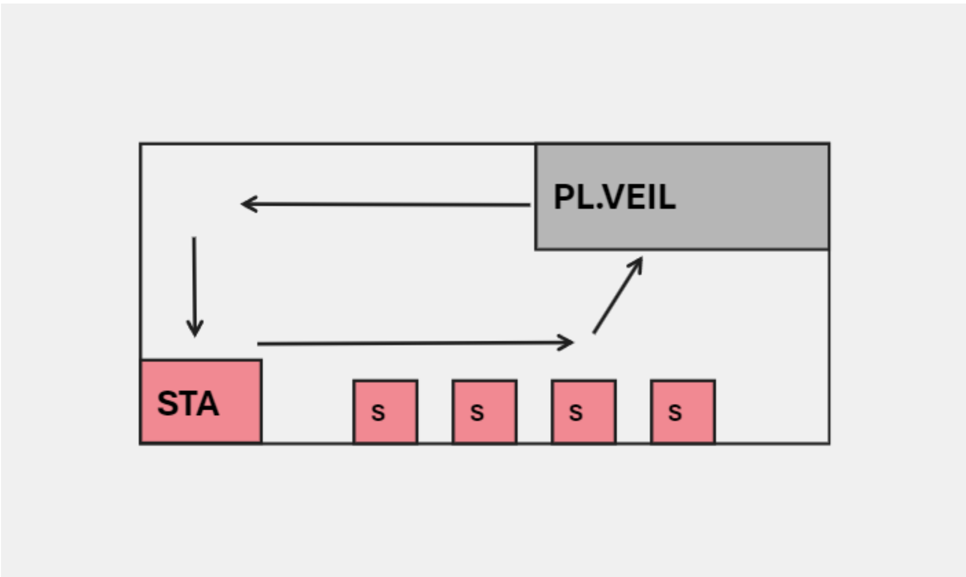

Figure 4.16: Scheme of the air diffusion in the auditorium.

Where:

- S indicates the seat.

- STA indicates the stage.
- PL.VEIL indicates the plasterboard veil where mechanical equipment like ducts is hidden.

The air supply will be performed with nozzle diffusers because of their throw, while the gather of vicious air through aphonic grilles on the plasterboard veil horizontal surface. In the sizing of these air terminals it has been considered that accepted velocities at +1.80 m from the ground are usually between 0.15 m/s and 0.25 m/s. Aphonic grilles help to avoid airborne noise transmission of the duct to the space. Those are designed so that air passes with velocity of about 0.5 m/s which has been a compromise between pressure loss and dimensions.

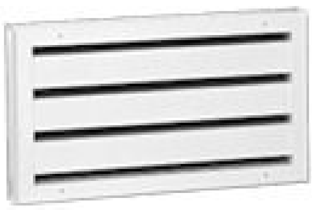

Figure 4.17: Render of the aphonic grille used in the auditorium.

In order to obtain a quiet place, nozzle diffusers selection is based on the throw and the sound pressure levels given by the manufacturer datasheet.

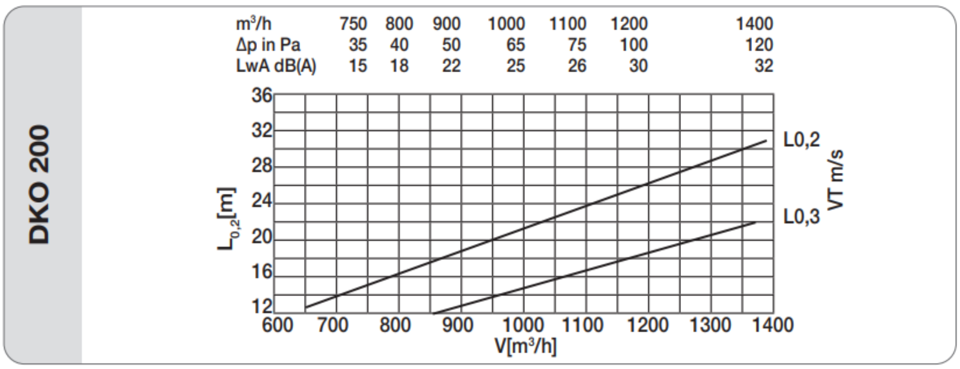

Figure 4.18: Characteristic's curve of nozzle terminals from its manufacturer.

Even though the nominal airflow of 700 m^3/h is almost at the lower end of the curve, this can ensure a throw at 0.2 m/s of about 14 meters (compared to the 8 meters needed) and less than 35 Pa and 15 $dB(A)$ of weighted sound pressure level. During summer they are designed to have their axis horizontal while during winter they are designed to have their axis facing downwards to let hot air reach people's bodies avoiding the buoyancy effects.

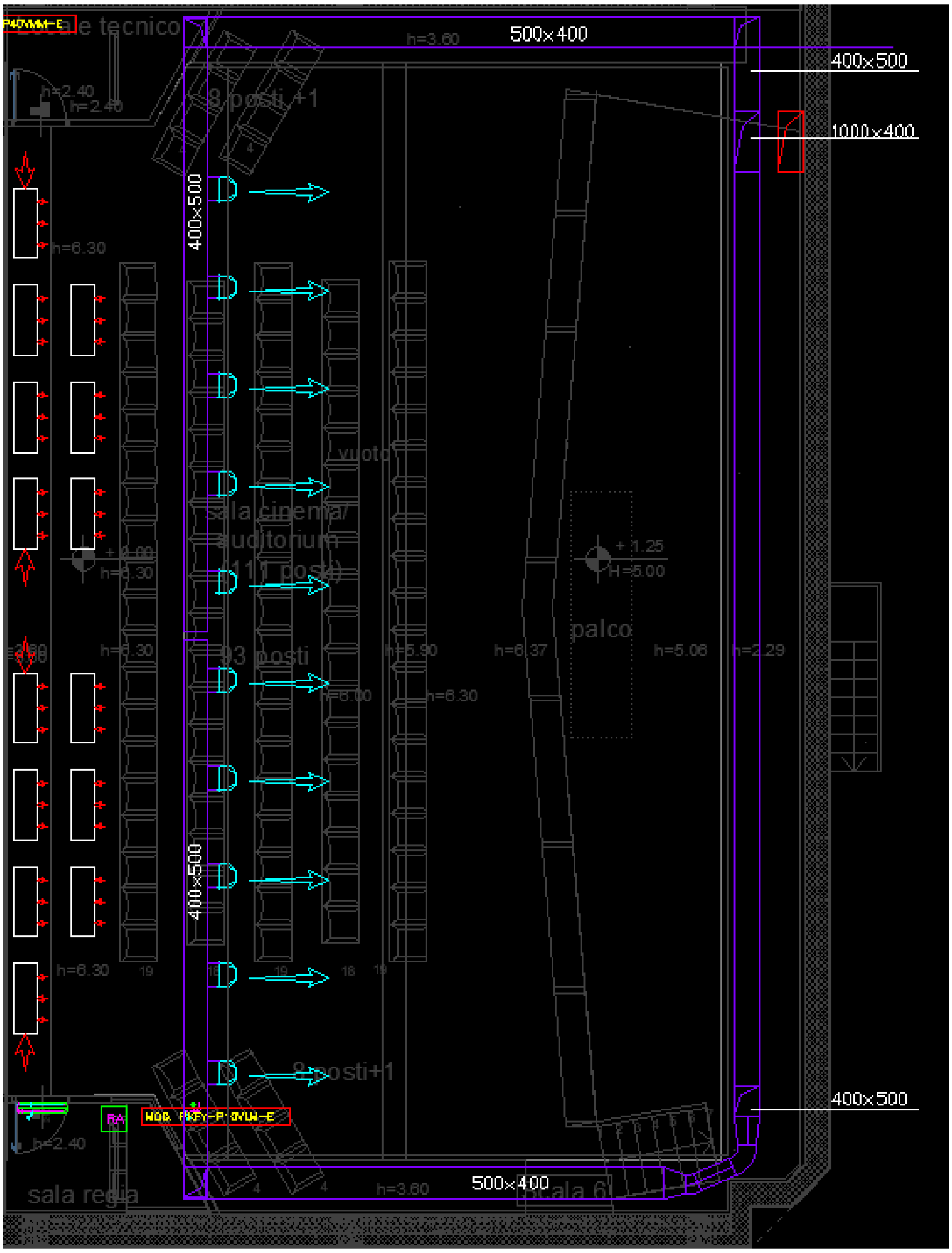

Figure 4.19: Planimetry of the mechanical system of the auditorium.

5 | Control logic

Control logic is the criteria that equipment follow during their operation. Control equipment is practically mandatory as nominal conditions for a zone do not occur every day and most often there are times when, as an example, people occupancy is not as design or outdoor temperature is different from reference conditions, hence heat transfer entity is less powerful. In the following sections, control logic is outlined.

5.1. Rooftop served spaces

Just to recall, multipurpose spaces at first and second floor and auditorium are served by rooftops. Fresh outdoor air volumetric flow rate can be handled by rooftops. This parameter is controlled by a dedicated damper that is able to open or close. Damper must be seen as an actuator and its degree of opening is decided by a 10V signal, sent by a controller that receives data about concentration of CO_2 in the space by a CO_2 sensor. Reference setpoint is 1000 parts per million (ppm).. Although there is a setpoint, a dead-band from 700 ppm to 1500 ppm is used to prevent continuous on/off cycle. When indoor CO_2 concentration reaches 1500 ppm, damper opens until 700 ppm is reached, then closes again. The choice of not having always full outdoor air flow is to save energy.

Figure 5.1: User interface connection with bus to the Unit.

CO_2 concentration is a second priority when compared to temperature and humidity control. Temperature control is done in winter acting on the amount of gas refrigerant flowing in the condenser and in summer acting on the amount of liquid refrigerant flowing in the evaporator. Humidity is controlled in summer acting on the temperature at which liquid refrigerant is sent to the evaporator. The lower, the more humidity is subtracted from the air. While in winter a water humidifier is activated according to the setpoint.

5.2. VRF served spaces

`Images/$CO_2$_sensor.png`

Figure 5.2: Mitsubishi's CO_2 sensor.

Ventilation in these spaces is performed thanks to heat recovery equipment. A CO_2 sensor placed in ambient will send a signal to the controller, which acts on a damper, i.e. the actuator. UNI EN 13779:2008[23] states that the best performance, called as "IDA1" is obtained with CO_2 concentrations less than 400 ppm. "IDA1" is one of the four category of indoor air quality. In this case, being a public library in a urban center where usually CO_2 levels do not undergo 450 ppm as a rough indication, the performances that can be targeted is 800, falling in the "IDA3" category. Although there is a setpoint, a dead-band from 700 ppm to 900 ppm is used to prevent continuous on/off cycle. When indoor CO_2 concentration reaches 900 ppm, damper opens until 700 ppm is reached, then closes again. The choice of not having full outdoor air flow always is to save energy. CO_2 concentration is a second priority when compared to temperature and humidity control. Temperature control is done in winter acting on the amount of gas refrigerant flowing in the heat exchange battery in the internal unit by a three way valve, which communicates with distributor BC. In summer, the same applies but liquid refrigerant flow is the regulated variable. Humidity is not controlled but fluctuates depending on the on/off state the heat recovery device and on the flow and temperature of cold/hot refrigerant.

Conclusions

The aim of this project work was to develop a HVAC system for a public library comprehending an auditorium and some multipurpose spaces. The engineering challenge has been to apply all the best practices, setting as priority a good IEQ, comprehending day light levels, noise and IAQ, which comprehends ventilation and thermohygrometric comfort. The buildings hosts spaces with different functionalities and with different use frequencies, and occupancy frequencies. The methodology to meet the priorities mentioned above are the calculation of winter and summer heat loads and the comparison of two prescriptive methodologies that estimate and prescribe values of outdoor air to be introduced into the room. Regarding one of the priorities which is thermohygrometric comfort, it was achieved through the estimation of heat loads and through the CBE calculation tool to get an estimate of whether the conditions in the room were acceptable. With regard to ventilation, which is another priority, two prescriptive methods were used: the first is UNI 10339, which prescribes guidelines for the amount of outdoor area and clean fresh air to be brought into the room the other is ASHRAE 62.1. This too provides numbers for assessing the amount of ventilation with outside air. In almost all spaces, the former was more stringent, so it was decided to adopt it. The importance of a good environment is the effect of providing a space where studying is aided. After that, there was also a whole selection of machines sizing and terminals sizing to take noise into account but that is not covered in depth in this paper. Before taking the step on the choice of terminals there was to figure out what kind of system could best serve these spaces within the budget. The frequency of use and occupancy are highly variable over time for most of the spaces in this building and are also dependent on the season of the year as there are times when, for various reasons, library use increases. Faced with this, there were mainly three choices: that of coupling primary air with room terminals (hydronic or direct expansion) or an all-air system or a hybrid of the two. The choice of an all-air system fell for spaces such as the second floor and the auditorium, which have the highly variable frequency of use and occupancy and thus a highly variable ventilation demand over time. In contrast, for spaces such as the atrium, it was decided to use primary air plus terminals, which in this case were chosen with a direct expansion circuit. This choice was also carried forward for other spaces also on the ground floor, such as the meeting room, reading rooms, and playroom. It remains clear that the choice of air conditioning done with a system that provides air as the heat transfer fluid is worse than one that provides water or freon gas because the latter greatly reduces the technical spaces associated with distribution.

Summarizing, the procedural iter has passed through considering the state of art and the present normative as first step; then, the energy analysis of thermal loads and ventilation requirements as second step; then the evaluation of the equipment size and rooms' air distribution layout as third and last step.

Using international standards and tools to assess comfort it was found that not always a simple and standardized solution exist for all buildings, especially this case, where multiple needs coexist.

Overall, the main challenges of design were the rapid occupancy change and the air distribution design.

Often the design work must be performed with strong deadlines and time constraints by the client, that's why some strong assumptions are considered in this project work, which can be improvable points. For example, operative temperature T_{op} is equal to room air temperature. This can impact CBE thermal comfort calculations and therefore occupants comfort. Furthermore, the energetic consumption analysis could be unfolded, which currently is not treated. The last point on this list but not in the general one, is the BMS (Building Management System) aspect, a serious and salient puzzle piece that coordinates and systematize the entire system to lower energy expenditure.

Overall, wether final users of the librarys' spaces will experience comfort or not depends on numerous factors such as maintenance, state of art installation, for example with roftop filters, and others.

Engineering design process is not a precise science.

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