UNIVERSITÀ DEGLI STUDI DI PADOVA

Dipartimento di ingegneria industriale
Corso di laurea magistrale in ENERGY ENGINEERING

Dynamic thermal investigation of office building and HVAC system in Debrecen (HU)

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Abstract

The ever-increasing amount of time spent indoors obliges us to pay great attention on the systems that ensure comfort under all conditions. The following study by means of a dynamic analysis defines the thermal load requirements of an office in a recently constructed, large building, with the peculiarity of a steel structure and an almost completely glazed cladding. Consequently, solar input becomes a resource and at the same time a problem for the management of winter and summer loads. A system was then designed that centrally provides for the use of an air handling unit for the completely electrical management, including power supply, of heating, cooling, fresh air supply and removal of exhaust air. The study was carried out on the temperature trend of the year 2019 in the city of Debrecen (HU), characterised by harsh winter climate comparable to Italian's F-band and by hot and dry summers that define a total electrical power demand of:

	$kWh_e/year$	$kWh_e/year * m^2$
Total electric consumption	102645.9	54.16

Table 1: Total electric consumption

Finally, this study experimentally analysed the comfort and discomfort conditions of the micro-climate of workstations under air diffusers that can work with a variable flow rate, verifying that the limiting conditions defined by the standards are met.

Contents

Introduction

The need of fresh air is a vital necessity to many organisms present on earth, from plants, animals and of course to the humans. Air is life, it's a basic physiological need, our brains and the cells in our bodies need oxygen in order to develop and perform appropriately. The air handling system could be compared to our own respiratory system, both are extremely important for the correct working of our body and of the building. A "sick building", can lead also to sick people. Eliminating unnecessary heat and pollutants significantly affects our comfort, weel-being, productivity and efficiency. Those who live or work in the building should never be exposed to poor indoor climates,as this may put at risk one's health. There are always a number of laws and official regulations that must be followed when constructing new buildings, or altering an existing one.

The work that will be presented in this paper, will focus on what could be a air ventilation possible solution to be installed in the brand new Forest Office building, located in Debrecen, Hungary. It is therefore a modelling of a system that could lead to an effective solution for air treatment inside a large office. Air treatment, which deals with purification, heating, cooling, the introduction of new air and the removal of exhausted air through forced ventilation. In order to do this, the design of an AHU was used, the operation of which, the energy expended, will become the core of the discussion. HVAC systems, are those that support the activities of the building. Heating system, ventilation and cooling systems determine the indoor climate. The HVAC systems are installed to keep the room air clean and room temperature within desired limits. They can be designed in this way to achieve these function but, most importantly, they must provide the room temperatures and air qualities that have been specified. This is why it is essential to consider functionality first and then choose, design and size. The solutions that best fulfil both the usage requirements and the buildings requirements. It is not justifiable to base a choice purely on different technical solutions and then, as sometimes happens, amend the functional requirements to suit a particular solution. Particularly with regard to air treatment, but more generally with regard to the entire sector, careful design is therefore necessary, as all elements that can reduce energy consumption should not be seen as stand-alone elements, but should be placed and contextualised within the overall building-installation complex. When projecting a new building, at the preliminary architectural design stage, an unbiased feasibility study should be carried out to determine the weather, the client's requirements which can be fulfilled within the technical and economical limits of the project. Choosing the wrong starting point often leads to poorly functioning system. It doesn't matter if the product installed is of very high quality if these primary requirements have not been yet accorded. To fulfill these guide lines, of course in chapter one, the investigation regarding the building will be outlined, including hypothesis in relation to the worst scenario, the losses and the energy required.
Particular consideration should be paid into the most critical season, summer. Energy consumption for space cooling has more than tripled since 1990, with significant implications for electricity grids, greenhouse gas emissions, and urban heat islands. This left abundant room for an ever-increasing growth of machines that were efficient in heating and cooling large and small

spaces. AHUs have had an increasingly wide use and consequently a large growth in parallel with the construction of new buildings. HVAC is the main end energy use of an office building with a weight close to 50%, lighting follows with 15% and appliances with 10%. The building type is a critical parameter in regard to the energy end uses, how the energy is distributed and it's energy intensity. The office are by far on top of the list in the third sector; excluding transport and industry [11].
IEQ is the combination of different parameters affecting our indoor well-being :thermal comfort, IAQ, visual comfort and acoustics. Although all the above-mentioned parameters affect the comfort and productivity of people, in this paper I will focus on the thermal comfort in terms of the inner temperatures of a building. Considering that people nowadays spend 90% of their time indoors, between home, work, gyms, bars and restaurants, having comfortable conditions is a decisive part of people's lives. Comfort therefore being the ultimate goal, it was deemed necessary to investigate through a study of measurements, the conditions to which a worker would be subjected by spending several hours in the office. And being that the machine works at a variable air flow rate, the present study has the aim to define thermal comfort with different air flow rates, and whether this is also verified with lower air flow rates, and again, the positioning and distribution of the air diffusers.
Consequently, a cost analysis was deliberately not discussed as it was deemed unnecessary for the purpose of the drafting.

Chapter 1

Thermal comfort and IAQ

1.1 Definition

Thermal comfort is something that the humans have never stop searching, for obvious reasons, and so far, remains one of the most important parameters, as well as one of the aims to strive for a new building construction. The definition of thermal comfort may appear straight forward, however, it hides different intricacies, due to the several parameters that influence its functionality. By definition, the thermal comfort is regarded as: "the condition of mind that expresses satisfaction with the thermal environment and is assessed by subjective evaluation" [5], from the ASHRAE55 or the: "Mental condition of satisfaction related to the thermal environment" [6] from the ISO7730 standard. Owling to the literal definition, comfort, it's by nature subjective and therefore it's a construct which is difficult to convert into physical parameters. The environment has a strong effect, as well as does the activity. This may appear strange, but the thermal comfort could be percieved by two people in opposite situations. For example, the evaluation of the thermal comfort conditions for one person that is running, maybe outside, and for another one that is relaxing on the sofa, will produce completely different results, which are entirely subjective. Thus, the activities involved also play a key role with regard to the thermal comfort. So far, we said that it's a personal evaluation, but we have not touched on the process behind its evaluation, and how our body can respond to different conditions. A brief overview with regard to how we evaluate the thermal environment will be outlined below.

1.2 Body regulation

Man considers the environment comfortable if no type of thermal discomfort is present. The first comfort condition is thermal neutrality, which means that a person feels neither too warm or too cold. And in order to maintain this condition, the temperature of the body should be always around 37 °C. What happen when this condition fails?

Ian Campbell in his article, Body temperature and its regulation [2], explains which are the consequences when we interact with hot and cold conditions. He explains that, when the outside temperature is too **cold**, two different phenomenons arise, the first one is the vasoconstriction, which reduces the bloody flow to the periphery areas, creating a shell around the chest, with the intent to keep warm the vital organs, as shown in the figure 1.1.a The second possibility, is renown as shivering, almost all muscle groups are activated (in an involuntary response) increasing the energy generation within the body that usually increases around three or four times the normal state. In the other words, when the body perceives a **hot** environment, the first reaction is the vasodilation, which leads to a higher blood flow all around the surface in the periphery area, an increase of the skin temperature and a higher heat exchange with

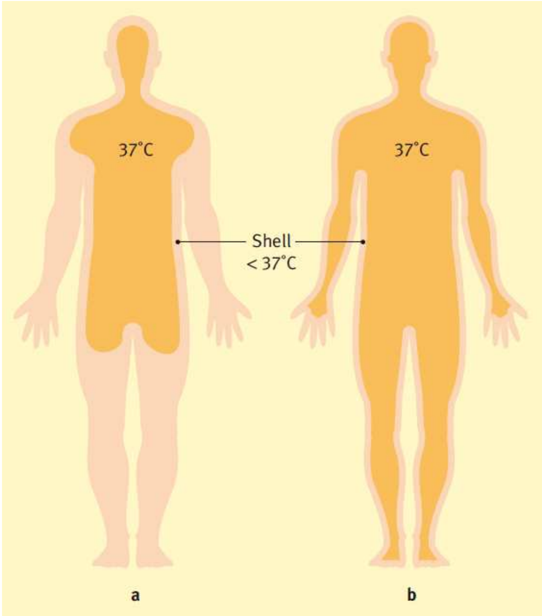


Figure 1.1: Body reaction to cold conditions (a) and hot conditions (b)

the air. Therefore, the body starts sweating, functioning as an effective cooling tool, as the energy required for the sweat to evaporate is taken from the skin, the so-called latent heat of evaporation. So far, it can be hypothesized that the temperature plays a key role in the comfort conditions, we will analyze later more in details the role of the temperature in the thermal comfort. Everything said so far, is the initial condition to reach the thermal comfort of a person, the combination of the skin temperature and the body's core temperature must provide a sensation of thermal neutrality, but it's not enough, because it only consider the bodily evaluation.
Let's see which are the conditions and the other parameters in order to reach the thermal comfort.

1.3 Subject conditions for the thermal comfort

Coupled with the thermal neutrality just described, in order to define the construct of thermal comfort, it is necessary to take into account the body's energy balance: the heat produced by the metabolism should be equal to the amount of heat lost from the body. So, we must consider, beyond the skin temperature and core body temperature, also the activity. The formula which describes the energy balance of the body is the following:

$$S = M - (W + E_{res} + C_{res} + C + R + E + K) \tag{1.1}$$

- S = energy stored
- M = metabolic rate
- W = external mechanical power - Depends on the type of activity, like walking, or swimming or just chill. In the most cases can be set equal to zero.
- E_{res} = latent respiration heat loss
- C_{res} = dry respiration heat loss (sensible term)
- R = heat loss by convection

Activity	**Met**	**W/m2**
Sleeping	0.7	40.6
Sitting	0.8	46.4
Typing	1.1	63.8
Standing short sleeves and light underwear	1.4	81.2
Ordinary standing work in shop, laboratory, and kitchen	1.6-2	92.8-116
Slow walking (3 km/h)	2	116
Normal walking (5 km/h)	2.6	150.8
Fast walking (7 km/h)	3.8	220.4
Sports - Running (15 km/h)	9.8	568.4
Ordinary carpentry and brick-laying work	3	174

Table 1.1: Met values

- E = heat loss by evaporation of sweat from the surface of the skin
- K = heat loss by conduction

Equation (1.1) represent a balance that describes which are the parameters that play a role in the stored energy (S), is equal at the main input, and defined as the metabolic rate (that in turn is the principal heat producer) minus all the losses through the body. To match the thermal comfort conditions, the term S must be equal to zero.
The equation also wanted to stress the importance of the metabolic rate and introduce it.
The metabolic rate encompasses the chemical and physical processes that take place in the cells and tissues of the human body and depends on the amount of the muscular activity.
From different studies [8], metabolism is measured in Met; 1 Met = 58.15 W/m^2 of the body surface. And on average, we can take 1.7 m^2 for a skin surface of an adult. So, for example, a person in thermal comfort conditions with an activity level of 1-1.2 Met, which corresponds to a normal employee's activity, will have a heat loss into the environment of around 100-120 W. We meet a minimum when the person is sleeping (0.8 Met) and a maximum while doing sports, where it's possible to reach also 10 or even more Met. The main values are reported in table 1.2 Another parameter that must be considered, decreases the energy losses through the body, are clothes. Therefore, clothing is classified according to its insulation value, which has its own measure unit which is Clo; 1 Clo = 0.155 m^2C/W.
The Clo value can be calculated by summing the values of the single garments which are tabulated in table 1.1
In conclusion, the most important parameters that we have to take into account are the metabolic rate and the clothes, or at least, these are the parameters related to the subject.
But what should be measured? This is exhibited in table 1.2.

1.4 SBS - Sick building syndrome

Sick Building Syndrome (or Sick Building Syndrome or SBS) is a set of symptoms manifested by those who stay for a prolonged time in unhealthy environments or where indoor pollution has been established. The main culprit of Sick Building Syndrome is poor indoor air quality, which is most commonly found in modern buildings, recently renovated buildings, and where there is poor maintenance of ventilation and air conditioning systems

Garment description	Clo	m2 °C/W
Underpants plus:		
- Shirt (short sleeves), lightweight trousers, light socks, and shoes	0.5	0.078
- Shirt, lightweight trousers, socks, and shoes	0.6	0.093
- Boiler suit, socks, and shoes	0.7	0.11
Underwear (short sleeves/legs) plus:		
- Tracksuit (sweater and trousers), long socks, and training shoes	0.75	0.116
- Boiler suit, insulated jacket and trousers, socks, and shoes	1.4	0.217
- Shirt, trousers, jacket, quilted jacket and overalls, socks, and shoes	1.85	0.287
Bra and pants plus:		
- T-shirt, shorts, light socks, and sandals	0.3	0.05
- Stockings, blouse (short sleeves), skirt, and sandals	0.55	0.09
- Shirt, skirt, sweater, thick socks (long), and shoes	0.9	0.14

Table 1.2: Clo values

1.5 Environmental measures

As underlined from the energy balance, we must bear in mind that the feeling of warm or cold, doesn't come from the room temperature, but from the amount of heat losses from the body, so, in light of this, we must measure the following parameters:

- Air temperature t_a [°C]
- Mean radiant temperature $\bar{t_r}$ [°C]
- Air velocity v_a [m/s]
- Humidity[1] [g/m^3]

The Mean Radiant Temperature of an environment is defined as that uniform temperature of an imaginary black enclosure which would result in the same heat loss by radiation from the person as the actual enclosure. As a hypothesis, we should imagine a person standing in the middleof an empty room (i.e. with no forniture), whom exchanges heat through radiation with each surfaces.
The equation to calculate the mean radiant temperature is the following:

$$\bar{t_r} = \sqrt[4]{\textstyle\sum_{k=1}^{n} F_{p-i}(t_i + 273)^4} - 273 \tag{1.2}$$

Where:

- t_i = surface temperature of surface i [°C]
- F_{p-i} is the angle factor between the person and surface i

It must be noticed, that the measurements of all the surface temperature of the room, and also the angle, are significantly time consuming; for this reason the calculation of the mean radiant temperature it's avoided where it is possible. But can be measured, as the Operative Temperature t_o can be measured as well, which is usually used combined with the Mean Radiant Temperature.

[1]There are several studies that show that the humidity doesn't play a relevant role in the thermal comfort, so I didn't measure it

The operative temperature, permits to calculate the comfort in thermal environment with just one parameter, (in terms of temperature) and also, as a difference with the mean radiant temperature, it's considered not only the heat transfer by radiation, but also the convective term too.
The general equation for the operative temperature is the following one:

$$t_o = \frac{h_r \bar{t_r} + h_c t_a}{h_r + h_c} \tag{1.3}$$

Where:

- h_r = heat transfer coefficient by radiation
- h_c = heat transfer coefficient by convection
- t_a = air temperature
- $\bar{t_r}$ = mean radiant temperature

But the question of the matter is, how we can calculate the two heat transfer coefficients? Usually, we don't have to calculate h_c and h_r because the ASHRAE55 standard give us the following simplified equation:

$$t_o = (1 - A)\bar{t_r} + At_a \tag{1.4}$$

The A term, it's tabulated in table 1.3, and it's proportional to the air velocity.

v_r	**<0.2 m/s**	**from 0.2 to 0.6 m/s**	**from 0.6 to 1 m/s**
A	0.5	0.6	0.7

Table 1.3: A values

And also considering that usually in residential buildings, the air velocity should be kept below 0.2 m/s, we can notice that the operative temperature is equal to the average between the air temperature and the mean radiant temperature.

$$t_o = \frac{\bar{t_r} + t_a}{2} \tag{1.5}$$

In light of everything said so far, it is possible to state a unique condition that allows to keep everyone in the room at comfort conditions doesn't exist, especially if they're carrying out different activity and wearing different types of clothes. For these reasons that it's useful talking about PMV and PPD.

1.5.1 PMV - Predicted Mean Vote & PPD - Predicted Percentage of Dissatisfied

Thinking to reach the right thermal comfort condition, it's like an utopia, but it's allowed try to get closer as much as possible, and in order to answer the question if we are not in the thermal comfort condition, how far away we are from it? In other words, which are the limits to ensure a reasonable thermal comfort?.
The answer, can be found within the parameters of the PMV, which from standard definition EN ISO 7730 [6] is the "index that predicts the mean value of the thermal sensation votes of a large group of a person on a 7-point scale".

Scale	Thermal Sensation
+3	Hot
+2	Warm
+1	Slightly Warm
0	Neutral
-1	Slightly cool
-2	Cool
-3	Cold

Table 1.4: PMV Values

It is important to highlight, even when the PMV-index is equal to 0, there will be some people that will be not satisfied with the temperature level, regardless of the clothes and the activity. The standard [6], on the basis of the parameters above-mentioned, has been developed three different categories of thermal comfort:

- (A): with
$$-0.2 < PMV < +0.2$$
- (B): with
$$-0.5 < PMV < +0.5$$
- (C): with
$$-0.7 < PMV < +0.7$$

Where (B) is an average comfortable environment.
In order to identify the categories, we need to calculate the PMV, and this can be carried out from the following Fanger's equation [3]:

$$PMV = [0.303e^{0.036M} + 0.028] * [(M - W) - (C + R + E_{sw} + E_d + C_{res} + E_{res})] \tag{1.6}$$

Which is implemented by standards in:

$$\begin{aligned} PMV = [0.303e^{0.036M} + 0.028] * \{[(M - W) - 3.05 * 10^{-3} * [5733 - 6.99 * (M - W) - p_a] \\ - 0.42 * [(M - W) - 58.15] - 1.7 * 10^{-5} * M * (5867 - p_a - 0.0014 * M * (34 - t_a) \\ - 3.96 * 10^{-8} * f_{cl} * [(t_{cl} + 273)^4 - (t_{mr} + 273)^4] - f_{cl} * h_c * (t_{cl} - t_a]\} \end{aligned} \tag{1.7}$$

Where:

$$h_c = 2.38 * (t_{cl} - t_a)^{0.25} \quad if \quad h_c = 2.38 * (t_{cl} - t_a)^{0.25} > 12.1(v_a)^{0.25}$$
$$h_c = 12.1(v_a)^{0.25} \quad if \quad h_c = 2.38 * (t_{cl} - t_a)^{0.25} < 12.1(v_a)^{0.25}$$

$$f_{cl} = 1 + 0.2 * I_{cl} \quad if \quad I_{cl} < 0.5clo$$
$$f_{cl} = 1.05 + 0.1 * I_{cl} \quad if \quad I_{cl} > 0.5clo$$

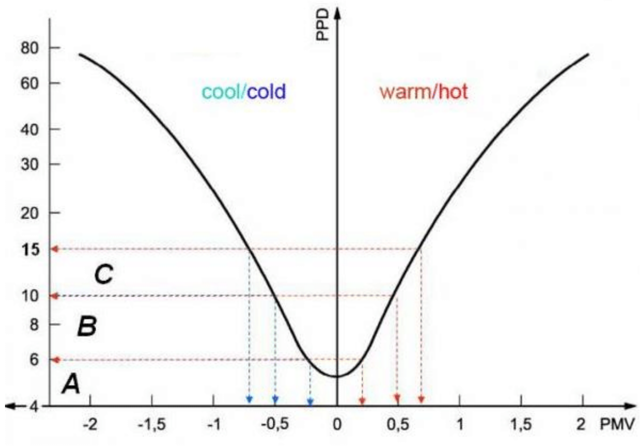


Figure 1.2: Relationship between PMV and PPD

The solution of the equation, comes from an iterative process.
The equation highlights the six variable that affect the PMV parameters, that as we have already saw, are: air temperature, mean radiant temperature, air velocity, relative humidity, activity and clothing resistance.

Another parameter that help us to evaluate how far we are from the satisfaction of people into the environment, is the PPD index - predicted percentage of dissatisfied, calculated from the equation 1.8; that provides an estimate of how many occupants in a space would feel dissatisfied by the thermal conditions, that is also strictly correlated with the PMV index as showed in the figure 1.2

$$PPD = 100 - 95exp(-0.03353PMV^4 - 0.2179PMV^2) \tag{1.8}$$

It's interesting to notice that as mentioned above, it's impossible to satisfy everyone present in the room, and that is showed in the graphs looking at the point where PMV is equal to zero; which corresponds more or less 5% of the PPD. That means that, even in the best thermal comfort conditions, there will be 5% of people that will not be satisfied.
The figure, shows also which are the maximum limits of the three different categories of thermal comfort, at which we will add the four types of local discomfort.
Table 1.3, reports the three categories of the thermal comfort [6], where usually, at least the class B is required for an acceptable comfort condition.

1.6 Local thermal discomfort

Even when the thermal neutrality is achieved, some trouble can arise if some parts of the body are exposed to a condition that could result in thermal discomfort. And these type of problems can't be solved just by adjusting the heating or cooling level, or the other parameters; they must be solved at the root of the problem. The standards groups the local thermal discomfort into these four headings:

- Asymmetric radiant field
- Vertical temperature difference
- Floor temperature
- Draught risk

C	Thermal Sensation		Local discomfort			
	Predicted percentage of dissatisfied PPD %	**Predicted mean vote PMV**	**Percentage of dissatisfied due to the draught DR** %	**Percentage of dissatisfied due to air temperature difference** %	**Percentage of dissatisfied due to warm or cool floor** %	**Percentage of dissatisfied due to radiant asymmetry** %
A	<6	-0.2<PMV<0.2	<15	<3	<10	<5
B	<10	-0.5<PMV<0.5	<20	<5	<10	<5
C	<15	-0.7<PMV<0.7	<25	<10	<15	<10

Figure 1.3: Thermal comfort categories

1.6.1 Asymmetric radiant field

If you stand in front of a bonfire on a cold day, after a period of time your back will begin to feel uncomfortably cold. This discomfort can not be relieved by moving closer to the fire, resulting in an increased body temperature. This is recognized as the asymmetric radiant field, which occurs when we have different parts of the body at different temperature.
In turn, this type of discomfort, is divided into four different problems, and usually just the two high-lighted entries are the more significant:

- **Warm ceiling**

$$PD = \frac{100}{1 + exp(2.84 - 0.174\Delta t_{pr})} - 5.5 \qquad if \quad \Delta t_{pr} < 23°C \tag{1.9}$$

- **Cool wall**

$$PD = \frac{100}{1 + exp(6.61 - 0.345\Delta t_{pr})} \qquad if \quad \Delta t_{pr} < 15°C \tag{1.10}$$

- Cool ceiling

$$PD = \frac{100}{1 + exp(9.93 - 0.5\Delta t_{pr})} \qquad if \quad \Delta t_{pr} < 15°C \tag{1.11}$$

- Warm wall

$$PD = \frac{100}{1 + exp(3.72 - 0.052\Delta t_{pr})} - 3.5 \qquad if \quad \Delta t_{pr} < 35°C \tag{1.12}$$

Experiments exposing people to changing degrees of radiant temperature asymmetry have proved that warm ceilings and cold windows cause the greatest discomfort [9].
In figure 1.5, we can see the different relationship between the radiant temperature asymmetry Δ_{pr}, and the percentage of dissatisfied calculated with the equations outlined above; and as previously mentioned, the figure shows that there are two discomforts more pronounced than others, at which should be paid more attention. Furthermore, table 1.5, shows for each type of discomfort, which are the limits of the different categories.

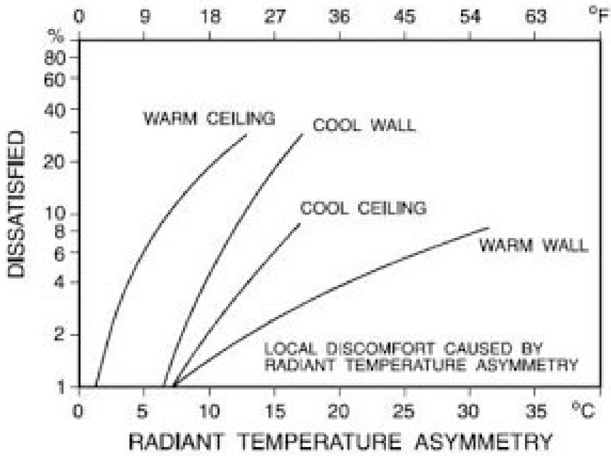


Figure 1.4: Radiant temperature asymmetry

Category	Radiant temperature asymmetry [°C]			
	Warm ceiling	Cool wall	Cool ceiling	Warm wall
A	<5	<10	<14	<23
B	<5	<10	<14	<23
C	<7	<13	<18	<35

Table 1.5: Permissible radiant temperature asymmetry for the three categories

1.6.2 Vertical temperature difference

Difference in temperature between head and ankles, can cause discomfort, regardless of this being caused by radiation or convection. The temperature difference is defined as Radiant Temperature Asymmetry $\Delta t_{a,v}$, and according to an experiment carried out on people being in thermal neutrality, it's showed that a 3°C air temperature difference between head and feet gave a 5% dissatisfaction level. Therefore, the 3°C has been chosen from standard [6] as an acceptance level for a sitting person at sedentary activity.
Figure 1.5, shows the percentage of dissatisfied as a function of the vertical air temperature difference between head and ankles, and also, tab 1.6 shows which are the limits of temperature difference for the three category of comfort.

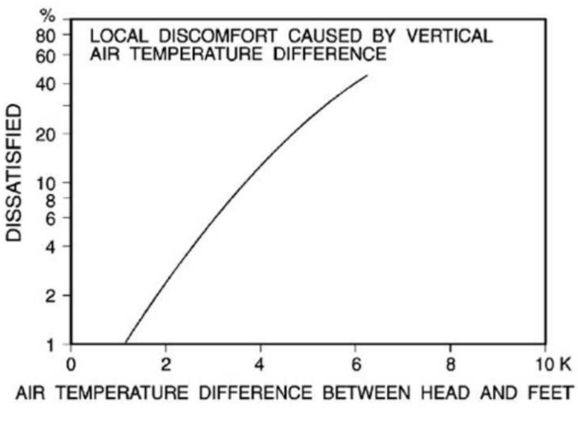


Figure 1.5: Local discomfort caused by vertical temperature difference

$$PD = \frac{100}{1 + exp(5.76 - 0.856\Delta t_{a,v})} \tag{1.13}$$

Category	**Air temperature difference [°C]**
A	<2
B	<3
C	<4

Table 1.6: Permissible range of the floor temperature for the three categories

1.6.3 Floor temperature

Due to the contact between floor and feet, discomfort could arise when the floor temperature is too high or too low.
If people wear normal indoor footwear the floor material is less significant. Therefore, it has been possible to set some comfort levels for this normal situation. Again, standard sets rules for comfort levels at sedentary activity, and for this specific case uses 10% as value of dissatisfied. This leads to acceptable Floor Temperatures ranging from 19°C to 29°C. Quite different recommendations are valid for floors occupied by people with bare feet. In a bathroom the optimal temperature is 29°C for a marble floor and 26°C for hard linoleum on wood. Also in this case, the table 1.7, shows which are the limits for the comfort level category.

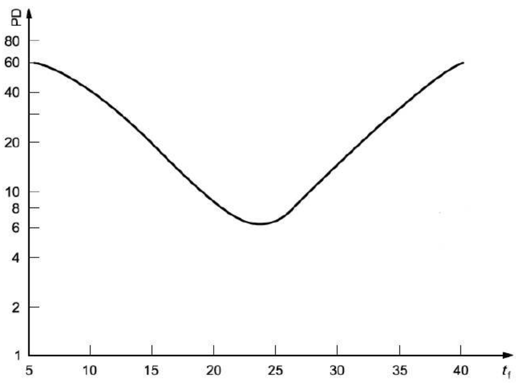


Figure 1.6: Local discomfort caused by floor temperature

$$PD = 100 - 94exp(-1.387 + 0.118t_f - 0.0025t_f^2) \tag{1.14}$$

Category	**Range of surface temperature of the floor [°C]**
A	19-29
B	19-29
C	17-31

Table 1.7: Permissible range of the floor temperature for the three categories

1.6.4 Draught risk

Draught is an unwanted local cooling of the body caused by air movement and temperature" [5], and it's also the most common complaint when talking about indoor climate in air conditioned buildings and ventilated space in general. People are most sensitive to draught in the unclothed parts of the body, therefore, draughts are usually only felt on the face, hands and lower legs. The draught, like for the other causes of discomfort, cay be expressed by evaluating the percentage of people dissatisfied due to the draught as follow:

$$DR = (34 - t_a)(v - 0.05)^{0.62}(0.37vTu + 3.14) \tag{1.15}$$

Where:

- DR = is the draught rating, i.e. the percentage of people dissatisfied due to the draught
- t_a = local air temperature [°C]
- v = mean local air velocity [m/s]
- T_u = local turbulence intensity

The DR could also have a positive side, because is lower for the people that feel warmer than the neutral conditions by, cooling a little bit the body an creating a more comfortable condition. The permissible mean air velocity is given in the figure for the categories A and C.

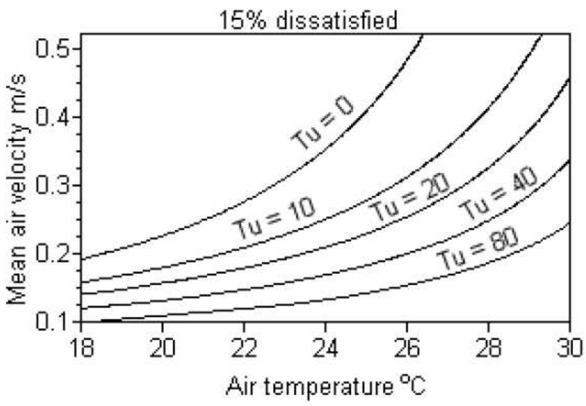


Figure 1.7: Category A - DR = 15%

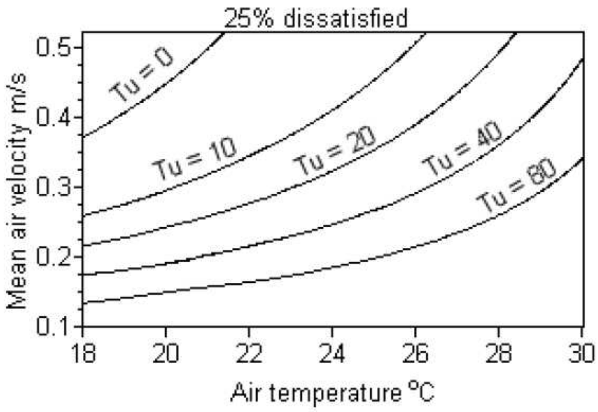


Figure 1.8: Category C - DR = 25%

Figure 1.8 shows that the mean air velocity, is function of the air temperature and turbulence intensity. Still from the standards, we know that turbulence, should stay between 30% and 60% in spaces with mixing flow air distribution.

Table 1.3, also shows, for the four categories of discomfort, which are the percentage limits of dissatisfied in order to assure to remain in the category A, B or C.

Chapter 2

Measurements

2.1 Introduction

Ahmed Benabed and Amir Boulbair in their article [1], investigate the performance of a multi-cone ceiling diffuser equipped with lobed inserts, this is assessed and compared with a traditional multi-cone ceiling diffuser, with a different mass flow rate, but focusing especially on the lobed. Also Fabio Nardecchia, Laura Pompei and Fabio Bisegna with their article [10] investigate the effect of a new type of ceiling diffuser especially used in exhibition rooms, with variable mass flow rate, but that kind of diffuser it's totally different from the one's that we used. The closest research with our studies, is the one made by Marek Jaszczur and it's team, which has investigate a single square ceiling diffuser [7],slightly different from ours but that can give us a good comparison between the results. Furthermore, they compare the experimental data with the numerical analysis carried out by means of the CFD obtaining similar results with the experimental ones.

2.2 Equipment

In order to get the measurements, a specific equipment has been used, more specifically, we used all the instrumentation of one of the biggest manufacturer in this field, which is "TESTO". Let's see, one by one which are the instruments:

- **Testo 400 - Universal IAQ instrument**

This is a high precision tool, that, coupled with different probes, can measure all the IAQ-related parameters, in accordance to the standards ASHRAE55, ISO7730, such us: flow velocity, temperature, humidity, pressure, illuminance, radiant heat, turbulence, CO2 and CO. The technical data are the following in the figure 2.1
The figure reports the technical data of this tool. It can be observed that the measuring range is significally wide, for both the temperature and pressure. Moreover, the Testo 400 is a really accurate instrument in every range of temperature and pressure, as it never goes beyond an error of 0.5% for temperature, and 1.5% for pressure; therefore ensuring significantly precise measurements.

- **Comfort probe**

As already discussed in reference to the others probes, this one must also be coupled with the Testo400 in order to get the measurements, and in particular, this one could be used to determine air velocity, draught risk and turbulence. This is very useful, because the draught risk and the

Differential pressure (internal sensor) - Piezoresistive	
Measuring range	0 to +200 hPa
Accuracy	±(0.3 Pa + 1 % of mv) ±1 Digit (0 to 25 hPa) ±(0.1 hPa + 1.5 % of mv) ±1 Digit (25.001 to 200 hPa)
Resolution	0.001 hPa

Absolute Pressure (internal sensor and external probe)	
Measuring range	700 to 1100 hPa
Accuracy	±3 hPa
Resolution	0.1 hPa

Temperature - NTC	
Measuring range	-40 to +150 °C
Accuracy	±0.2 °C ±1 Digit (-25 to +74.9 °C) ±0.4 °C ±1 Digit (-40 to -25.1 °C) ±0.4 °C ±1 Digit (+75 to +99.9 °C) ±0.5 % of mv ±1 Digit (Remaining Range)
Resolution	0.1 °C

Figure 2.1: Technical data Testo 400 - testo.com

turbulence, are the main parameters that we will investigate, since they are, together with the air velocity, the principal variables responsible for the discomfort in the micro-environment like the workstation. Technical data in figure 2.2.
It's true that all these instruments can be used ones the volumetric flow rate has been set, but in order to ensure an acceptable mass flow rate, two other important tools have been used in order to asses it. The volumetric flow rate, comes from the AHU[1] that is the heart of the air conditioning present in the lab, but that's not enough to ensure the correct flow rate, so, a "**dumper**" inside the ducts, has been used (figure 2.3).
This simply works like a barrier for the flow inside the pipe, by closing or opening the space inside.
Instead, for the proper measurement and then for the setting of the volumetric flow rate, the **KIMO DBM 610**, is employed, an instrument that measures the volumetric flow rate just below the diffuser, allowing the regulations, in order to get the exact flow rate; reported in the figure 2.4

2.3 Displacement or mixing ventilation?

The mechanism of displacement ventilation. In displacement ventilation the fresh air is supplied from the bottom part of the room while the exhaust intakes are usually placed in the superior section of the room. The air is supplied close to the heat and contaminant sources, here the heated air rises upwards due to the buoyancy effect and reaches the highest part of the room. The room is vertically stratified into two zones, the lower part up to the highest, which is called neutral zone, wherein the air is adequate for humans, whereas the top part is characterized by a concentration of contaminants and the air temperature is 2÷3 °C higher than

[1] short for AIR HANDLING UNIT

Temperature - NTC	
Measuring range	0 to +50 °C
Accuracy	±0.5 °C
Resolution	0.1 °C

Absolute Pressure	
Measuring range	+700 to +1100 hPa
Accuracy	±3 hPa
Resolution	0.1 hPa

Velocity - Hot wire	
Measuring range	0 to +5 m/s
Accuracy	±(0.03 m/s + 4 % of mv)
Resolution	0.01 m/s

Figure 2.2: Technical data comfort probe

the supply air.
The main **advantages** of this system, is that if it's well designed, it has a high efficiency, allowing saving energy and money, ensuring a good air quality , and a low pollutants density. These characteristics, lead to a high thermal comfort. On the other hand, the **disadvantages**, except for the size of the pipes[2], are that is not suitable for large cooling load request, and usually a backup system is often necessary to help the main one. Also, this kind of ventilation is mostly used for cooling applications.

The mechanism of mixing ventilation. Mixed flow ventilation is characterized by the room air being inducted into a high-speed air jet. The ventilation pattern chose for the conditioned space is a trade-off between a correct mixing of the air and a reduced risk of drafts due to the high velocity of the air supplied to the room. High turbulence inside the spaces ensures good mixing, leading to uniform temperature and pollutants concentration. However, it is necessary to ensure that after a certain vertical distance y, where the air flow hit the occupied zones, the velocity of the air flow, is lower than v=0.2 m/s, value imposed from the every standard regarding ventilation system like [6]. Air is supplied by the ceiling or the wall (upper part) through grills in the case of the upper part of the wall, or through a diffusers, where a false ceiling, should be considered to take into account the space needed for the ducts and the diffusers itself. Return grills are usually placed in the lower part of the room. Cool air streams and warm air streams behave differently when they are at a temperature lower than the room temperature and due to the difference in density between the room air and the supply air it is possible to observe two phenomena. With regard to warm air streams we can observe the buoyancy effect, this can create problems of stratification inside the space. On the other hand, with reference to cool air streams, an air drop occurs, so it is necessary to avoid the direction of the air flow to upon the occupants' heads. It is important to foresee how much the jet will

[2]which is irrelevant because for the power that we will have to face, every system will have a huge size of the pipes

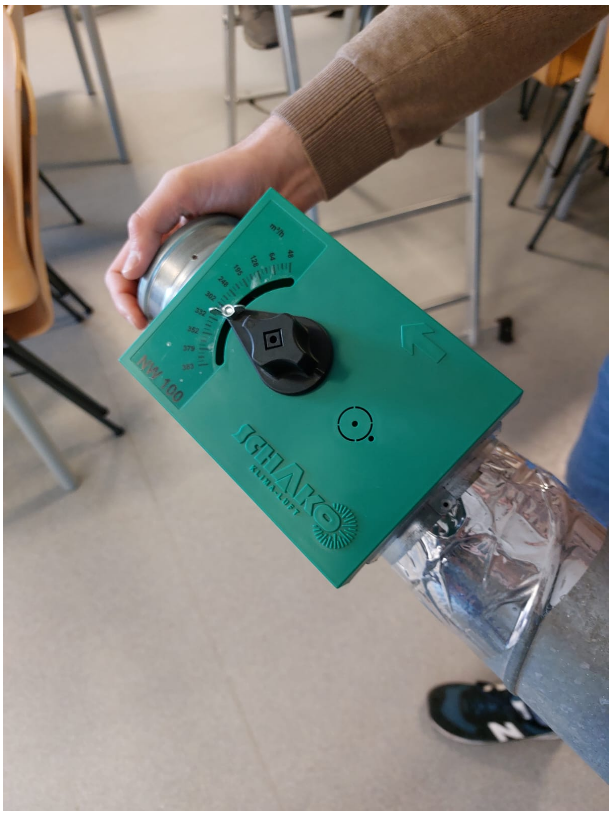


Figure 2.3: Stopper

Figure 2.4: KIMO DBM610

rise or fall, and due to this problem, the diffusers are supplied of blades that can be placed in different modes, for heating or cooling.
A short cut of the ventilation system could be considered, when the return grills are on top of the room together with the supply grills. The ventilation efficiency is reduced when the air supply is directly withdrawn from the return grills without having been in the people's operative zone. So, the main **advantages** of this type of ventilation is found within the capacity to mix the fresh air with the air present in the room, as well as the ability to limit the pollutants. Also it's suitable for significant heating or cooling loads. The main **disadvantage** is connected to the high velocity of the air at the outlet of the diffusers, but with a correct design, we can assure the required velocity exactly at the chosen distance. In light of this, the latter ventilation system was chosen, as it was more compatible with the need for a high cooling load demand.

2.4 Experiment set-up

The experimental test was carried out in the air ventilation and air conditioning system laboratory in the engineering university of Debrecen - Hungary, with an average temperature of the room around 19.2°C. We took under consideration a small micro-climate and not the entire room, considering the sit position of an employee close to the diffuser. It's possible to delimit a measurements area of $4m^2$, insofar as, the maximum distance of the measurements under the diffuser, in x and y direction, was 2m, with a step of measurements of 0.5m.
The air diffuser is a DQJ-NW 600 ceiling swirl diffuser(figure 2.7 and figure 2.6), provided by the Shako manufacturer, which has been specially designed for heating and cooling purposes, supply and return air installations, particularly for infrastructures such as office spaces, shop-

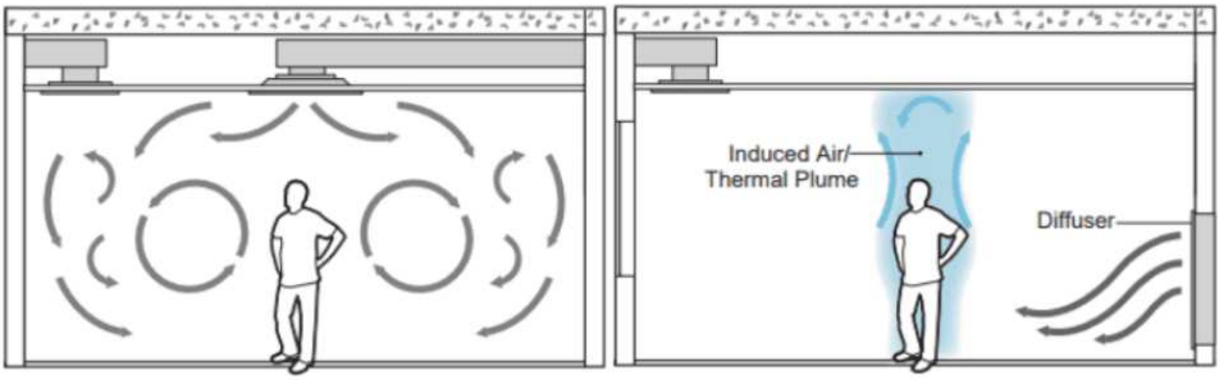


Figure 2.5: The mixing ventilation on the left, and the displacement ventilation on the right

ping centers, restaurants, or cinemas. The air jet injection it's has been ad-justed and it's has been settled in a medium configuration between summer and winter. Therefore, this entails that half of the blades, have been opened for the winter conditions, and the other half have been closed for the summer conditions, resulting in a mixed set up. In this way we're sure that the flow doesn't remain attached to the ceiling, but also doesn't point straight to the target. A AHU provided the necessary flow rate, with a constant inlet temperature of 22°C.

Figure 2.6: Diffuser DQJ-NW600 front

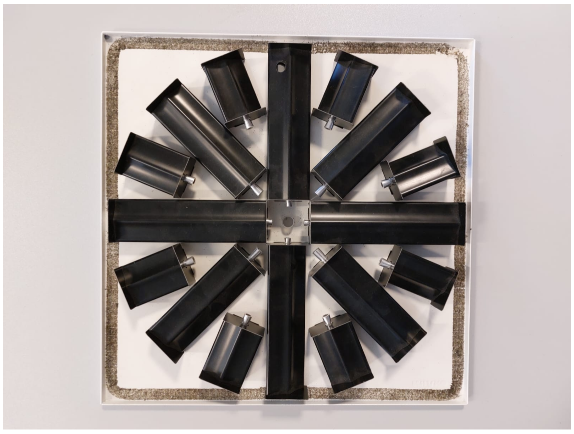


Figure 2.7: Diffuser DQJ-NW600 back

The first measurements are carried out without any desk, thus just by measuring the air flow and the parameters with the sensors that were installed at 0.1, 0.6, 1.1 and 1.7m, which correspond to the different part of the body in sit and stand position. The following heights refer to the sit and for the standing position:

- 0.1 m correspond to the ankles
- 0.6 m is more or less the height of the seat
- 1.1 m refers to the sit position, thus at the height of the neck,
- 1.7 m correspond to the standing position

The experimental set up is reported on the figure 2.8 and 2.9, where the measurements, as we can see, have been taken for the different heights, along the z-axis, and along the x-axis in steps of 0.5m from each other's until reaching 2 meters far from the diffuser.

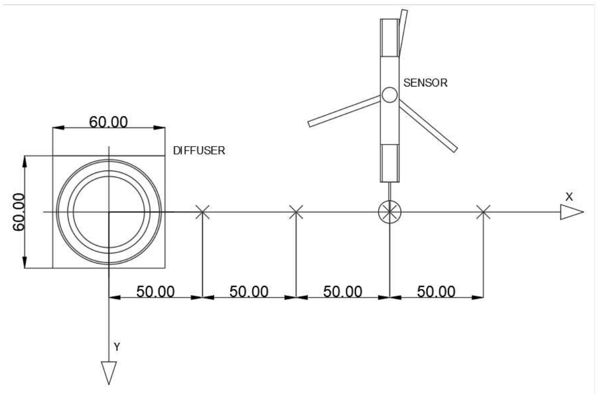


Figure 2.8: Height view

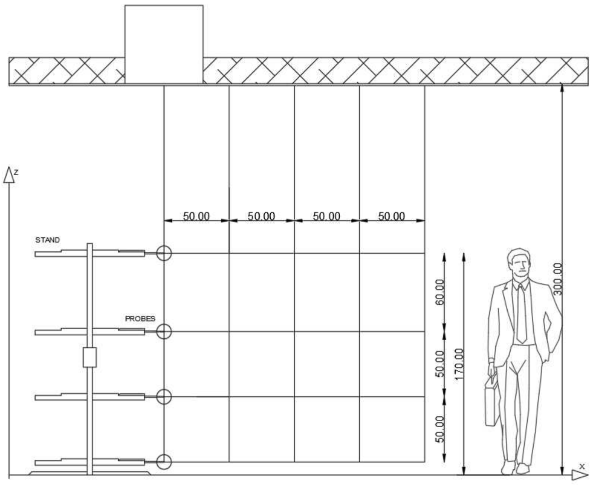


Figure 2.9: Side view

I have investigated the discomfort conditions, for five different air mass flow rates, 200, 300, 400, 500, and 600 m^3/h, by mainly observing four parameters: air velocity, draught risk, turbulence and temperature.

2.4.1 Results first measurements

- *Air velocity*

Air velocity, according to standards [5], should be kept under 0.2 m/s once the air reaches the target. The main thing that could be noticed, and without doubts the most important, is that the air flow, goes above the velocity limits just for the length of x=1m and x=1.5m, both of them with 0.21 m/s, which cannot be seen as a problem. That is good in terms of thermal comfort and draught risk. In all the other measurements at every distance, velocity remain below the limit value. Focusing on just one measurement, for example in x=0m; it can be noted that the data, hasn't the same path while the air flow descends towards the ground. However, there are two sets of flows (i.e., 400 and 600m^3, 500 and 200m^3) that have the same shape just with different velocities. The last one, 300m^3 instead, follows a totally different velocity variation. This is a behavior that could be seen in the measurements closest to the diffuser, like for x=0.5m but, slowly that we move away from it, (x=1, x=1.5 etc.) a totally random shape start to develope and the velocities decrease significantly, probably due to the major mixing of the flow within the air.

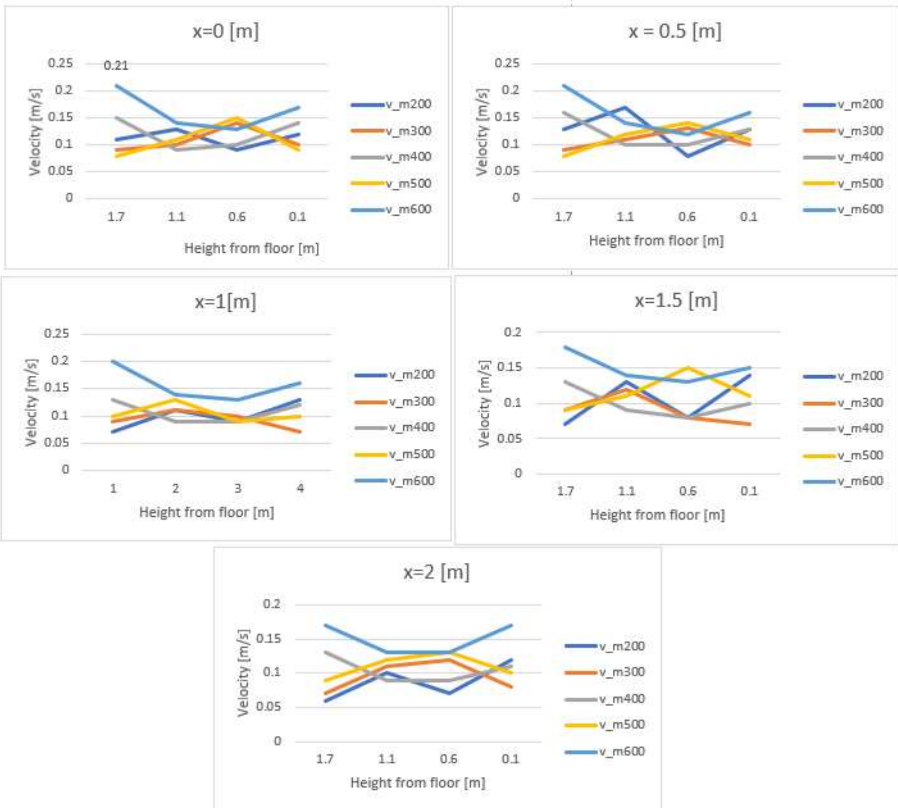


Figure 2.10: Velocity profile for different steps of measurements

- *Turbulence*

The turbulence is directly proportional to the draught risk, however it is also true that it has much more impact on the velocity. The turbulence equation, defines turbulence as a chaotic mutation in pressure and flow velocity. A higher rate of turbulence leads to high fluctuation of velocity through the air, which may lead into a non-perfect mixing of the air supply in the ambient. According to the present case, this is the main reason why we should pay attention to mantaining the between 30% and 60%, which are the limits imposed by the standards in space with mixing air ventilation, whereas for the displacement ventilation it may be set slightly lower.

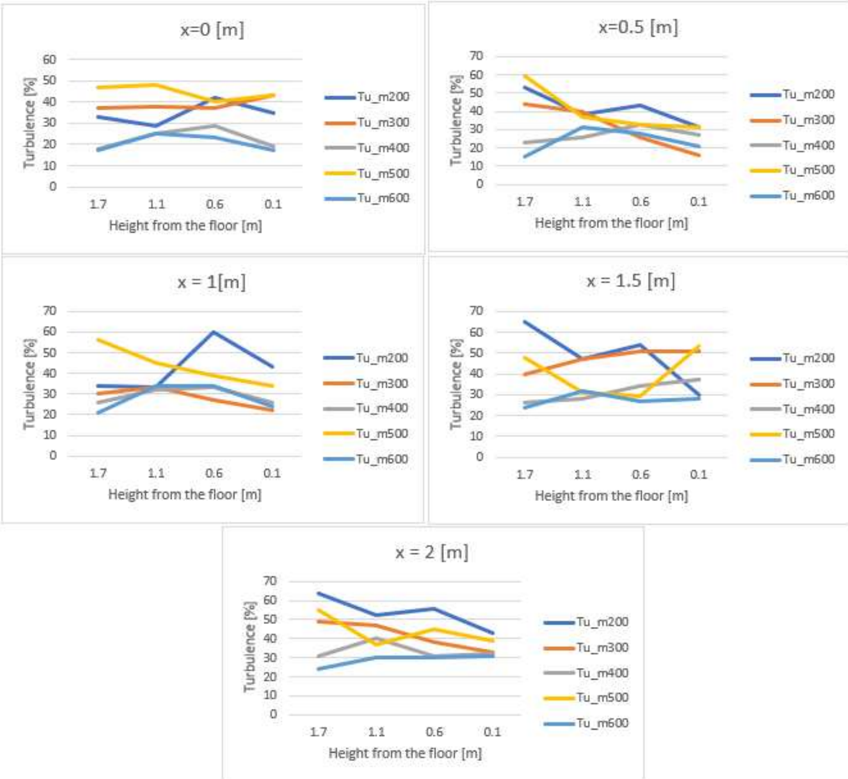


Figure 2.11: Turbulence profile for different steps of measurements

Contrary to the air velocity, in the several measurements done for the turbulence, from the data collect in figure 2.11, we can't individuate any coupling, and not even a geometry that is repeated in the various measures. It should be also noted that the higher value of the turbulence, results from the lowest value of the air flow rate, wich reaches a maximum of 65%. However, if we check for draught risk, the higher value doesn't result from the lowest air flow rate. This highlights that the velocity counts more than turbulence, and confirms that we don't face any problems keeping the turbulence in the safe range, between the 30% and 60%. For our micro-climate, the diffuser chosen seems to be working correctly.

- *Draught risk*

As already said in paragraph 1.6.4, draught risk represents the main reason of discomfort, thus is important to avoid surpassing the limits in order to prevent discomfort.
In the graph presented in the figure 2.12, a first glance it appears that there are no values. In every position and height that exceed 25%. This result, allow us to say that in every position, our micro-climate doesn't go beyond the class C of the thermal comfort. And it's also positive that there are few points that go above the 20%. As expected, initially, at a height of 1.7m, the biggest flow rate of 600 m^3, belongs to the category C in all the measured distances along the axis-x, and then slowly decrease reaching the lower value that belongs to the B category. Oppositely, the lowest mass flow rate, $200m^3$, denotes an abnormal trend in the second measure at x=0.5, is it reaches a value of 22% at the height of 1.1m; but except for this exploit, we can say that it belongs within the A category for all of the remaining measurements. For the $300m^3$

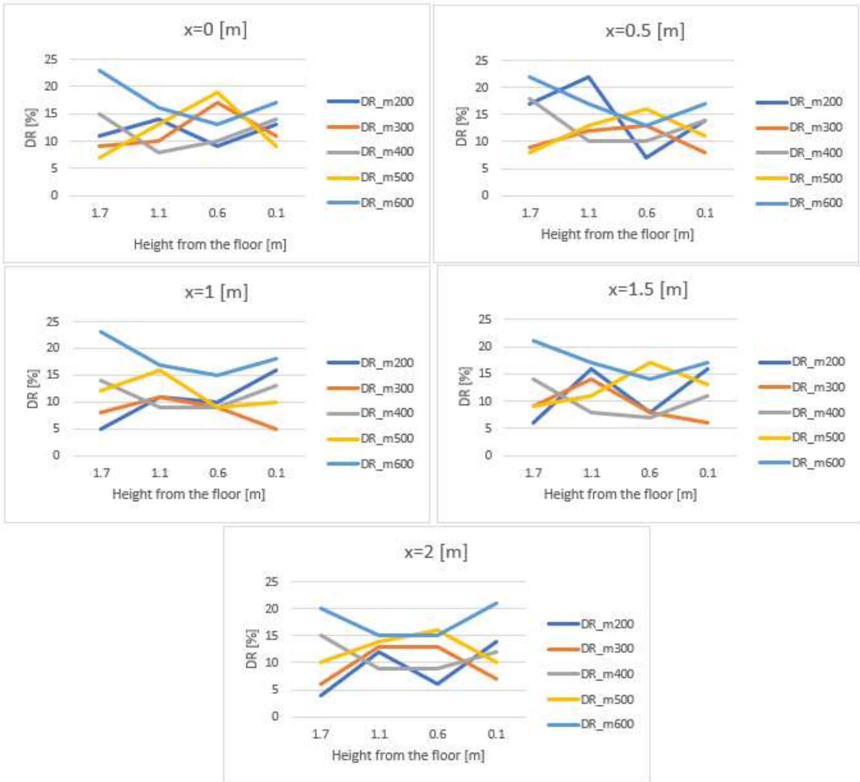


Figure 2.12: Draught risk profile for different steps of measurements

and $400m^3$, we can notice that except in one point, each value is over 15%, thus this may be labelled as a category A. However, the last mass flow rate, -$500m^3$-, shows an average behavior, fluctuating between 10% and 19%, reporting the higher values at a height of 0.6.

2.4.2 Results second measurements

For the second measurements, after having checked the parameters under the diffuser in the empty room, a standard office desk ($140x70cm, h110cm$) was added in order to investigate if it can make some difference in the thermal comfort parameters. Moreover, the measurements have been compared with and without a plexiglass panel on the desk, such as those that can be found in study rooms, bank counters or any public office post Covid19. The aim is to investigate, if the addiction of these two elements, could affect the thermal comfort, by studying the case with two different air flow rates and making a comparison between the empty room, and the room with the above-mentioned added furniture.
The average temperature during the measurement was: 18.6°C.

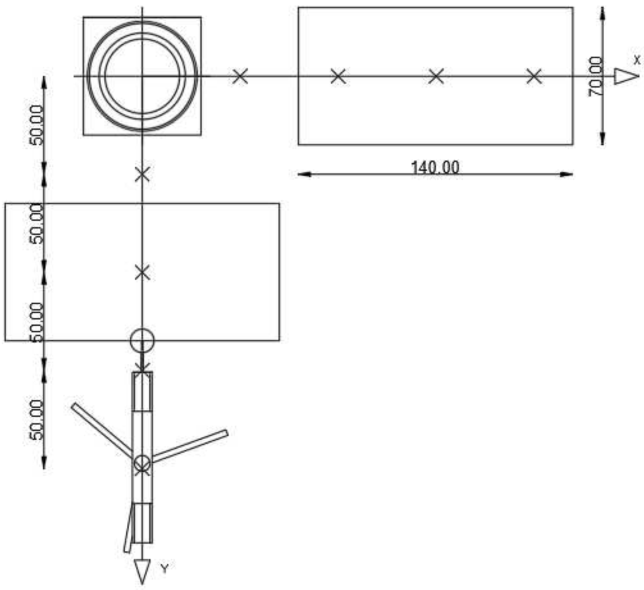


Figure 2.13: Height view of second measurements

After the measurements were carried out, along the x-axes, no significantly variation was noticed, and this is probably because the flow from the diffuser went to reach the plexiglass parallel to them, facing the thickness of the panel. For this reason, the results will not be reported; instead, I will report only the ones taken along the y-axis. Only two volumetric flow rates have been taken seening the time consuming of the measurements $300m^3$ and 600^3, and only at the height of 1.1m and 1.7m, due to the fact that the others two measures, are below the table, reason why they would not produce significant results. However, except the above outlined differences, the assessment of the room is the same used in figure 2.9.
Let's analyze the data for the 300 and $600\frac{m^3}{h}$

- *Velocity*

As we can see in figure 2.14, the measurement directly under the diffuser, plexiglass or not, doesn't present any differences, most probably because the air flow rate physically hasn't space to be stopped by the plexiglass, so the data are exactly the same, for the height of 1.7m as well as for 1.1m. But when me move a little bit away (i.e., up to the height of 1.7m), with and without plexiglass, the velocity continues to decrease until it reaches a minimum and start to increase again. At 1.5m and 2m, the velocity with plexiglass increases a lot compared to the other case, and this is due to higher turbulence as we can see in figure. That means that even at 0.6m over the desk, the panel has a visible effect. For the height of 1.1 from the soil, the measurements seem to not result in any significant differences, as, with and without plexiglass, velocity has the same value almost in every measurement. Anyway, the velocity as expected, remains below $0.2m/s$ which is the optimal value.

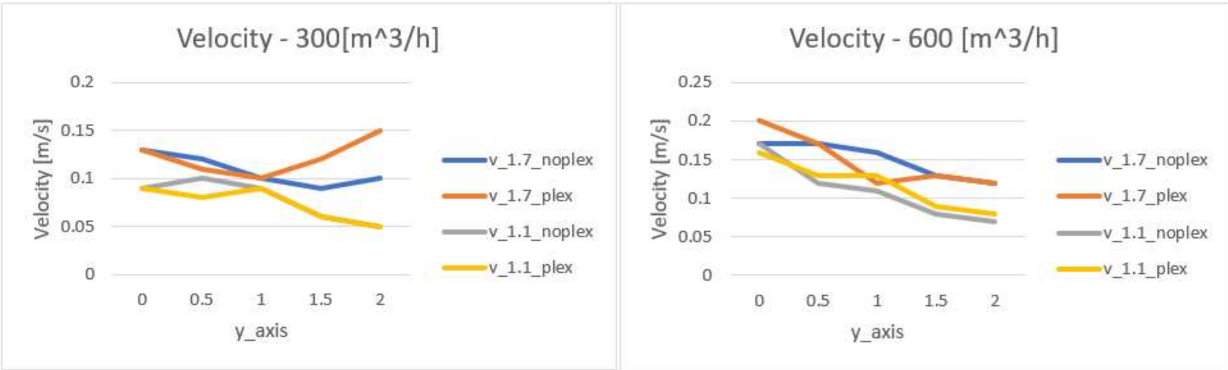


Figure 2.14: Velocity in the second measurements, 300 and $600m^3/h$

- *Turbulence*

For the turbulence, we should get far a little to see significant results, as we can see in the figure, the measurements of 0,0.5 and 1m are quite similar with the same path, but suddenly especially for the measurement close to the desk, 1.1m, the turbulence without plexiglass increase a lot reaching a maximum of 92%. This appear to be a quite strange value, probably due to the bounce of the air flow rate on the desk. With the panel, we don't reach so high values, however the turbulence reaches a value of 60%, which is still considered considerably high. Also, for the air flow rate of $600m^3/h$ the values remain high, around the limits of 60%, especially for measurements further away from the diffuser, however the values do not go below 30%. Nothing irregular can be noticed, due to the more normal distribution of the graphs.

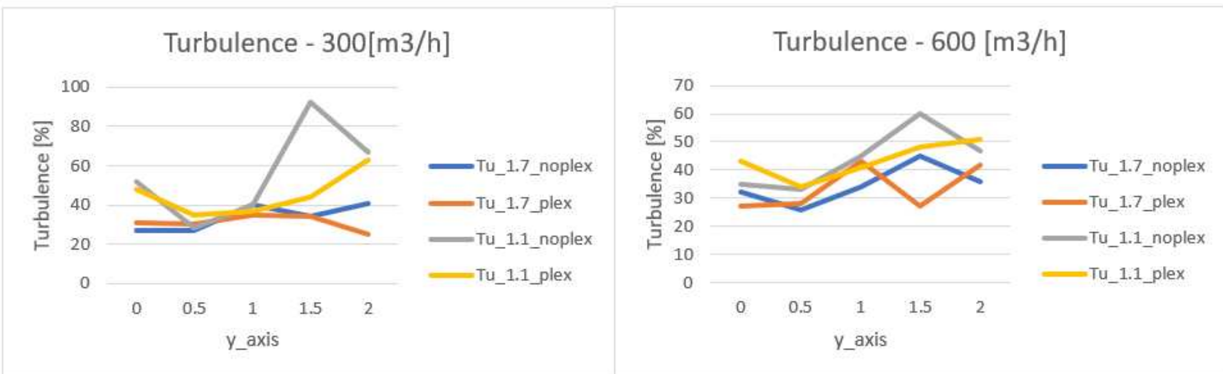


Figure 2.15: Turbulence in the second measurements, 300 and $600m^3/h$

- *Draught risk*

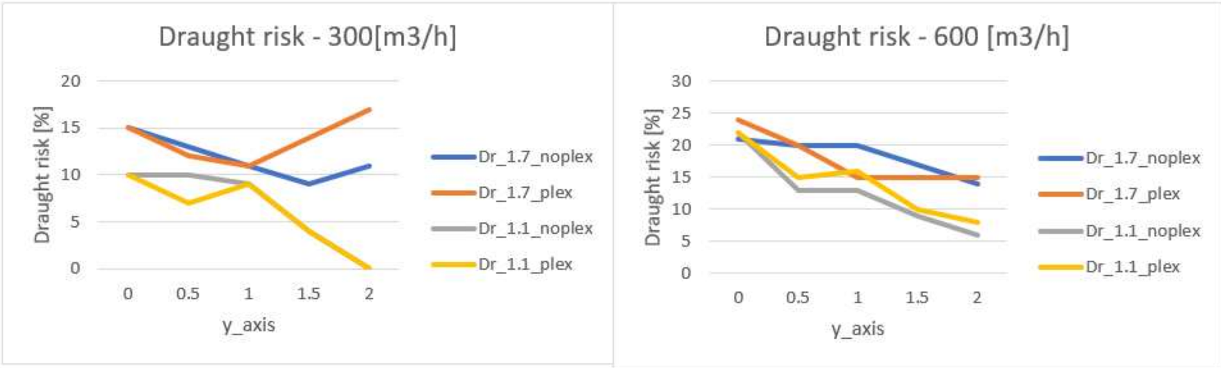


Figure 2.16: Draught risk in the second measurements, 300 and $600m^3/h$

It is important to notice that, the trend of the draught risk, with or without panel, at 300 and $600m^3/h$ is really close to the velocity showed in figure 2.14. Thus, as per the velocity; at $300m^3/h$ for the height of 1.7m, the flow starts from 15%, decreasing until 1-1.5m and then rising again. Opposite behavior is reported for the lower height, which decreases until it reaches the value of zero. At $600m3/h$ the trend becomes more regular, starting from a higher value, then, slowly decreasing at the furthest and distance, where both for 1.7 and 1.1 m reach value below the 15%, placing the diffuser in the A category. The C category it's reached just in the measurements under the diffuser, i.e., at 0 and 0.5m.
We can't properly compare the two measurements, because there are several different parameters, but what we can actually say, is that in both the cases, the diffuser perform really well, keeping all the parameters below the limit values, avoiding any kind of discomfort, and keeping the velocities in the correct range. Also, in light of the measurements, it's allowed to say that the plexiglass panel, doesn't cause an increase in the turbulence, or the draught risk. Morover it hasn't any effect on the parameters taken under study. Therefore, what is important is the choice of the right diffuser, starting from the correct sizing of the ducts, pressure drop and velocity in the final branches of an air conditioning system.

Chapter 3

Office building

3.1 General data of the building

The building taken by us under study, is located in Debrecen (HU), next to the city park and one of the university campus, an area regarded as the new business center of the city, surrounded by other commercial buildings and a residential area. Built in the 2018, the building is divided into three volumes, which are connected to each other by corridors. In our discussion, I will focus just on the ground floor, located in the bigger volume. Considering that I am discussing about a recently built building, the high efficiency of the structure should be clear, therefore great attention has been paid to details, including the ventilation system.
In order to simplify the calculation, the office area was considered as three large spaces for two reasons:

1. If we had considered individual offices, it would have led to a long and less accurate calculation.
2. The spaces within the office could change over the years making the calculation meaningless.

And at the end, room by room, the area and the volume extrapolated by the CAD files are:
In the figure 3.3, I reported the CAD blueprint of the entire office without furniture, to better understand the floor blueprint of the office.
The structure of the building is made of steel for the bearing structure, which has been also been insulated and covered. Therefore the steel is not visible anymore. Then as we can see in the pictures 3.1 and 3.2, it is almost entirely covered by windows.
In order to calculate the static peak power for heating and cooling, I used the following values of thermal transmittance respectively for the floor, the windows and the steel. For the ceiling it was not necessary because, since there is another office above the one that I'm considering, it could be considered at the same temperature. Therefore, it is like if the office had no losses through the ceiling.

For the pillars, it was not possible to find out exactly the layers which are installed, and I decided to use a U-value from the literature, which is $0.5[\frac{W}{m2K}]$ Instead, for the glaze of the building

Area [m2]	**Windows area [m2]**	**Walls area [m2]**	**Volume [m3]**
1894	622	156	6629

Table 3.1: Office values

Figure 3.1: Bigger block of the forest office

N°	**Layers description**	**S**	λ	ρ	**R**	**U**
	From internal to external	m	W/(mK)	kg/m3	(m2K)/W	W/(m2K)
1	**Ceramic tiles**	0.015	**1**	2300	0.015	66.66
2	**Sand and gravel concrete 1800**	0.04	1.01	1800	0.040	25.25
3	**Expanded cork panels with binders of 130kg/m3**	0.05	0.045	130	1.111	0.90
4	**Polyethylene**	0.005	0.35	950	0.014	70.00
5	**Expanded clay concrete 500**	0.06	0.2	500	0.300	3.33
6	**24cm mixed brick slab, ribs reinforced concrete; 1150**	0.24	0.68	1150	0.353	2.83
7	**Gypsum plaster**	0.015	0.7	1400	0.021	46.67
8	**TOT**	0.425			1.854	**0.53**

Table 3.2: Thermal transmittance for the floor

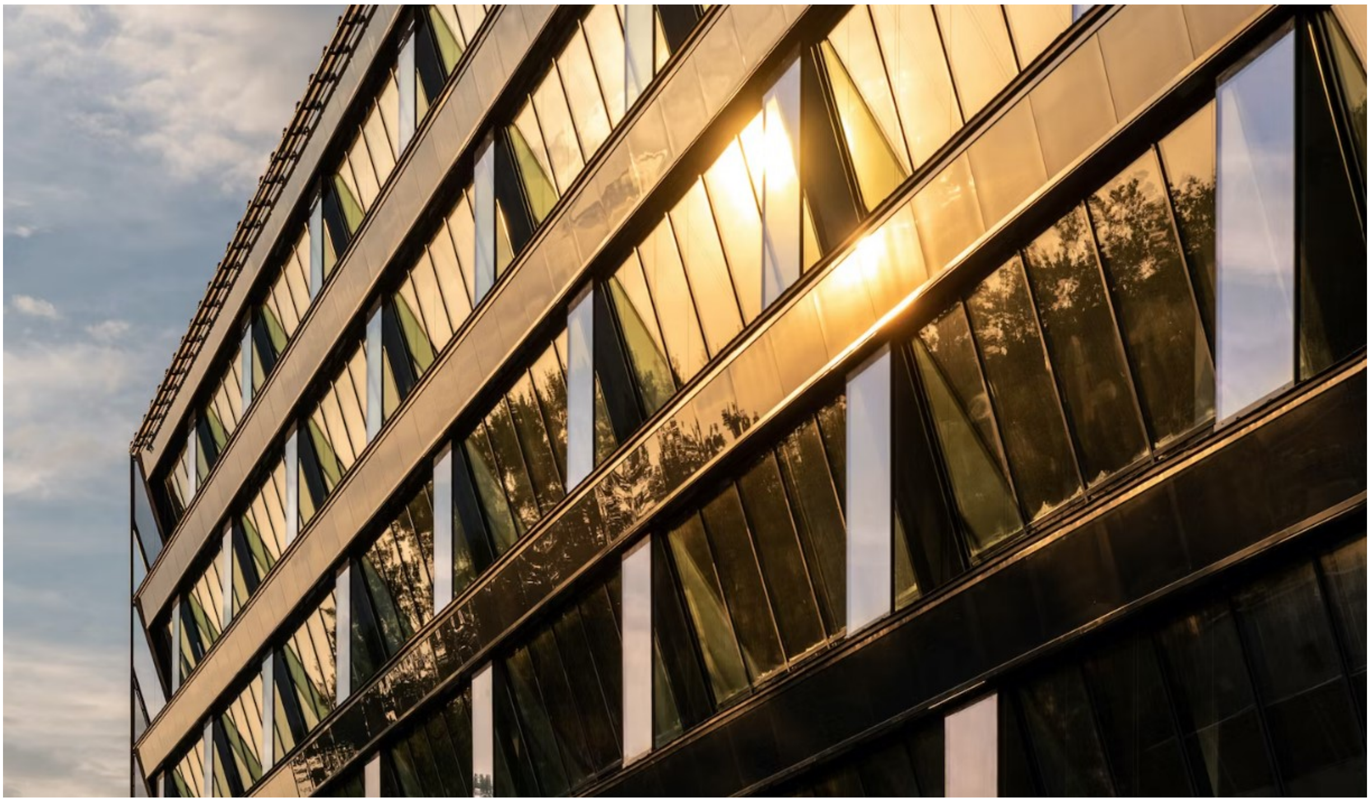


Figure 3.2: Windows details

($622m^2$ solely in reference to the floor that I'm considering), considering that they are covering almost the whole building, a triple glaze has been utilized full filled within the space by argon which significantly decreases the heat exchange between inside and outside, thus decreasing the transmittance value. These kind of windows are really high performance windows, and for the calculation, considering the entire window, i.e. frame and glass, I used a U-value of $0.5[\frac{W}{m2K}]$. An example of these two layers, has been reported in the figures 3.4 and 3.5
Also, the following assumption should be highlighted as it has been taken into account in the software calculation:

1. The building usually, it's occupied following the usual office work hours, i.e. from 08:00 h to 18:00 h during weekdays. The environmental design criteria should be met during the occupied hours of a design day in the summer. Furthermore, the 99% of the hours during the year will be less severe than the design day.
2. The spaces in the office building are used for ordinary office work inside the occupied zone: a distance larger than 0.6m form walls and heating and air terminal devices, and up to a height of 1.7 m above the floor.
3. The activity of the occupants is mainly sedentary office work, 1.2 [met] and the clothing insulation is 1.0[clo] during the winter season and 0.5[clo] during the summer season.
4. The air temperature is equal to the mean radiant temperature[1]
5. Low-polluting building materials and furnishing are systematically selected, providing a

[1] In spaces without heating panels, the mean radiant temperature is often close to the radiant temperature, i.e. the operative temperature is approximately equal to the air temperature, radiant asymmetries need not to be taken into account and may be discounted

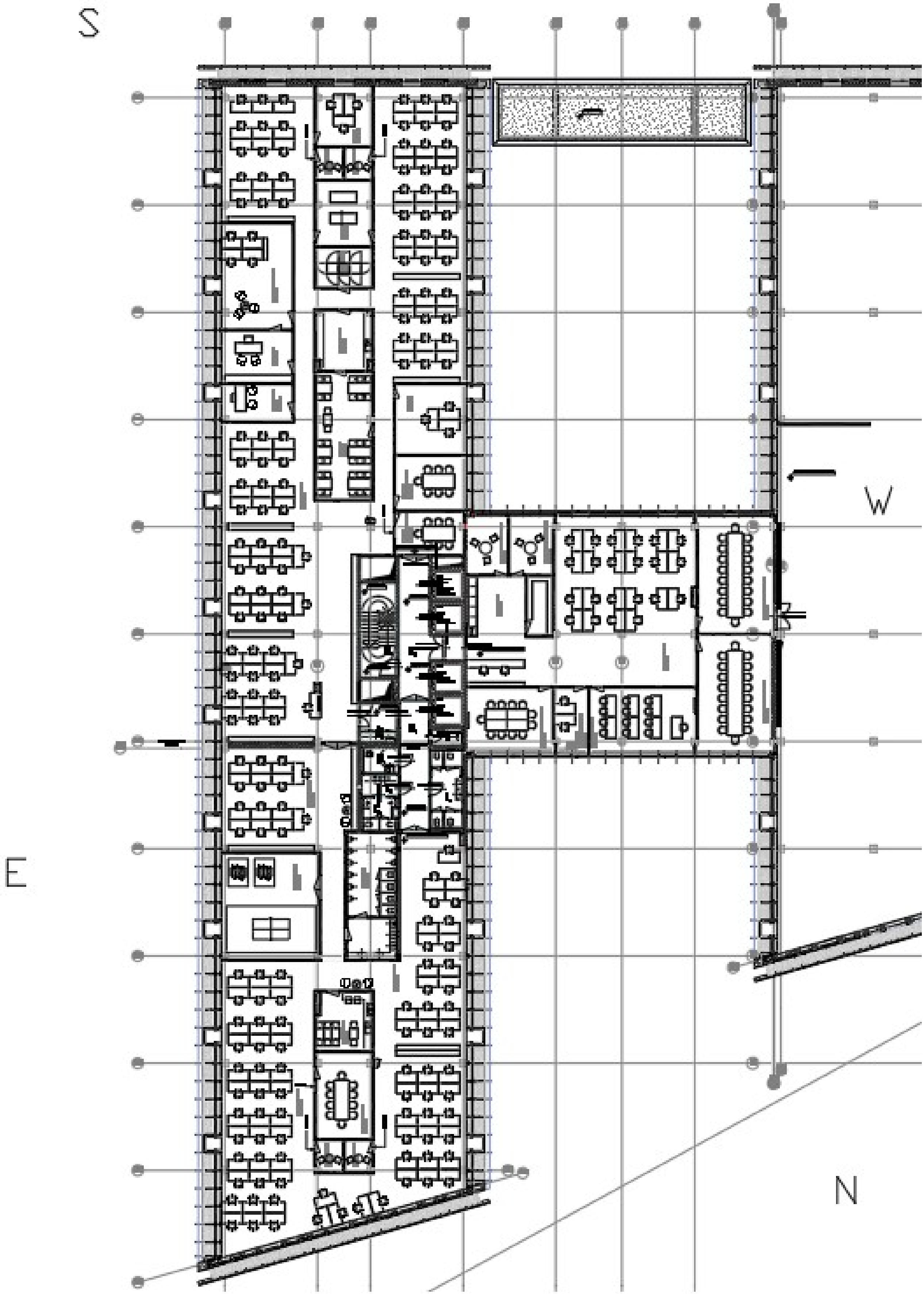


Figure 3.3: Office plan

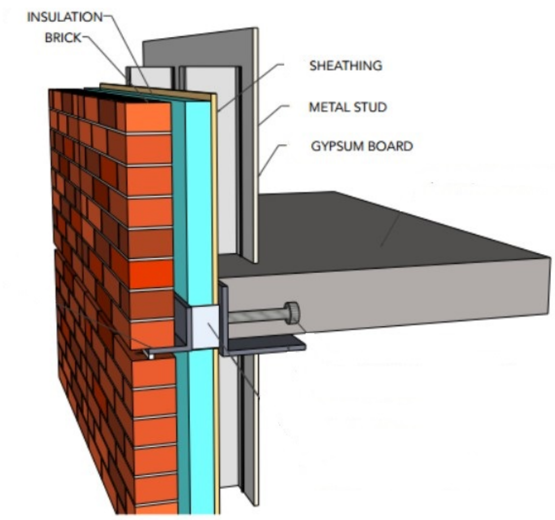


Figure 3.4: Pillar's section

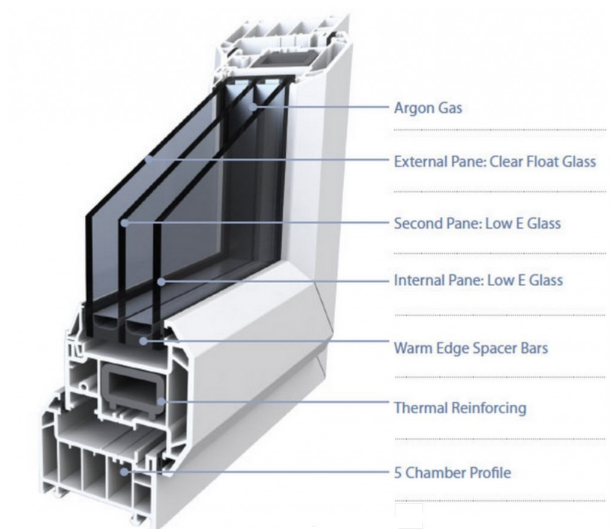


Figure 3.5: Window's section

pollution load of 0.1 [$olf/(m2floor)$], which corresponds to 1.0, 0.7 and 0.4 l/s, respectively for categories A,B and C.

At the end of the calculation I obtained the following values:

- Heating load = 22.8 kW
- Cooling load = 149 kW

These two values supply the project with a significantly wide range, that is probably due to the high percentage of the windows, which are reached by a huge quantity of solar heat gain, which help in winter, but becomes a big issue during summer.

Chapter 4

Total volumetric flow rate

The total volumetric flow rate, or total mass flow rate, represent the main parameter that it's required for the design of the air handling unit. The calculation of the total volumetric flow rate will determine the size of the plan, is immediately clear that a right sizing is fundamental to achieve the perfect matching with the power required from the building. The starting point of the sizing, are:

- Heating load = 22.8 kW
- Cooling load = 149.8 kW

It's immediately clear that the main condition that can cause the greater problems, is the cooling condition, due to the huge amount of the power required, and it's for this reason, that the discussion will begin with the latter point.
In the following figures 4.1,4.2.4.3, the main components and the schematic draw of our AHU for both seasons are shown. The following drawings, represent the heating and the cooling coils; the ones that aren't working in that specific condition, are colored in light gray.

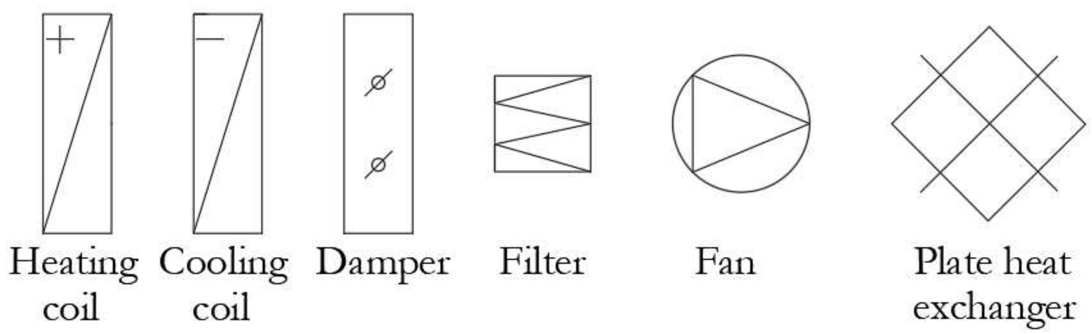


Figure 4.1: Main components of the AHU

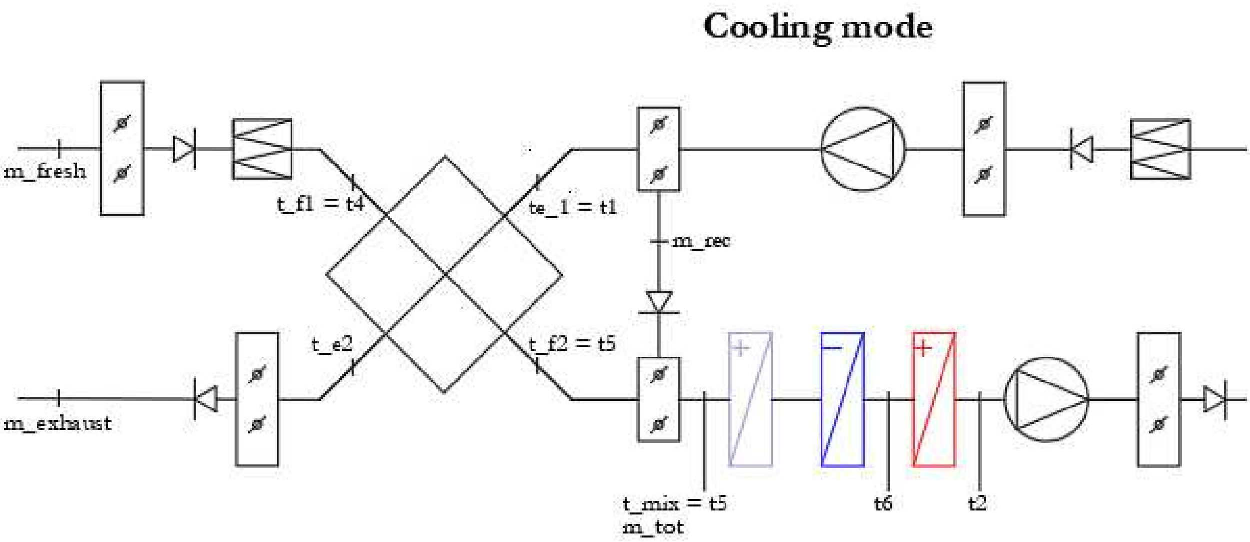


Figure 4.2: Schematic draw of the AHU for summer

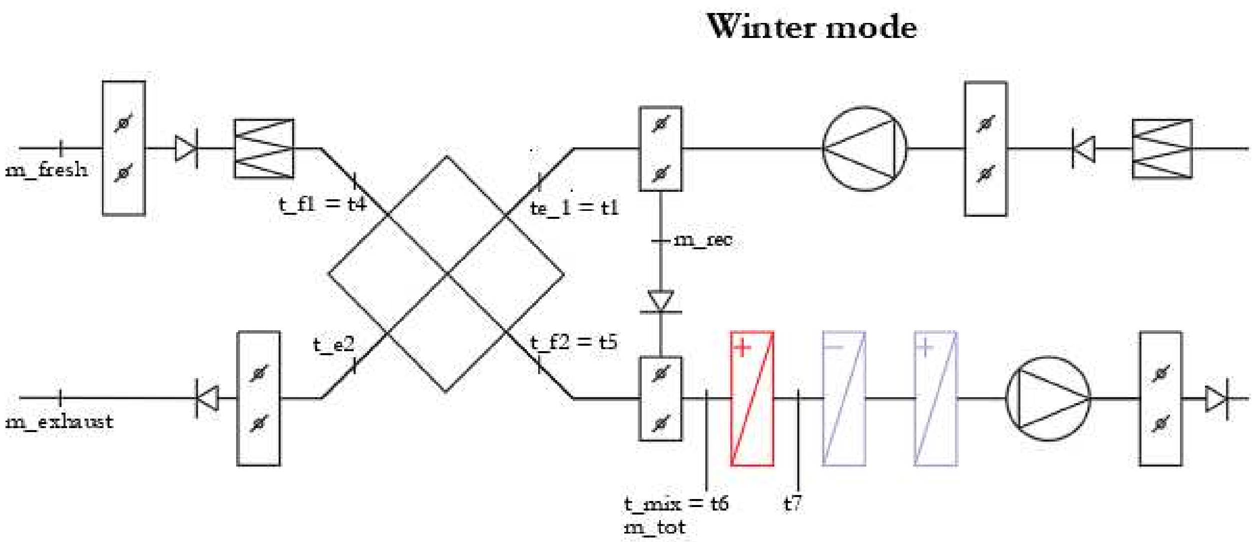


Figure 4.3: Schematic draw of the AHU for winter

4.1 SUMMER DESIGN

After understanding and assessing the cooling load, the next step is the calculation of the **humidity content**, which is defined per person by the Hungarian standards, which give us the value of $50\frac{g}{hour*person}$; since the maximum number of person estimated is 319, with the following equation, it's possible to calculate the humidity flux generated by the people inside the building:

$$m_v = \left(\frac{50}{1000 * 3600}\right) * 319 = 0.00443 \quad \left[\frac{kg}{s}\right] \tag{4.1}$$

Next step is the calculation of the ratio of enthalpy Δh, and relative humidity Δx, which is equal to the ratio of the cooling load divided by the humidity content:

$$\frac{\Delta h}{\Delta x} = \frac{Q_{cooling}}{m_v} = 33792 \quad \left[\frac{kJ}{kg}\right] \tag{4.2}$$

This ratio, represents the initial point of the Mollier diagram h-x, that will help us find the temperatures, the enthalpy and consequently the power required from the different components of the system. Let's see how.
From the upper part of the diagram, we can find the positive and the higher value of the ratio $\frac{\Delta h}{\Delta x}$. Taking approximately the value of 33792; from there, I have drawn a straight line until the point of the 0°C on the left side of the diagram. This line has been translated until touching the point of the inner conditions of the building, 26°C and 50% of humidity, which is marked with the number one in the figure 4.6.
Directly on the same line, it's possible to find the conditions at which the air it's injected through the diffusers into the room, which are $t_{supply} = 18C$ and $\varphi = 75\%$ marked with 2. Precisely, the relative humidity has been uncovered, because the supply temperature has been defined once I decided to fix the ΔT equal to 8°C, leading to a supply temperature that is $18C$.
Once the two points have been identified, the enthalpies of them can be derived, via the gray lines in the graph, which allow the respective enthalpies to be read, and we have:

- $h_{internal} = 53\ \frac{kJ}{kg}$
- $h_{supply} = 42\ \frac{kJ}{kg}$

From here, using the inverse of the power equation, we can calculate the total mass flow rate:

$$P = \dot{m}\Delta h \tag{4.3}$$

$$\dot{m} = \frac{P}{\Delta h} = 13.6 \quad \left[\frac{kg}{s}\right] \tag{4.4}$$

And then the volumetric flow rate:

$$\dot{V} = \frac{\dot{m} * 3600}{1.2} = 40855\left[\frac{m^3}{h}\right] \tag{4.5}$$

And last the air change flow rate:

$$n = \frac{\dot{V}}{m^3} = \frac{40855}{6629} = 6.16 \quad [h^{-1}] \tag{4.6}$$

Acceptable value according with the hungarian standard HUNGARIA STANDARD
Where:

- P = cooling load [kW]
- $\dot{m}$ = mass flow rate $\left[\frac{kg}{s}\right]$
- Δh = enthalpy difference $\left[\frac{kJ}{kg}\right]$
- $\rho_{air} = 1.2 \quad \left[\frac{kg}{m^3}\right]$

The value of $\dot{V}$, is the main parameter that will go to define the size of our AHU. Once we have defined the value of the mass flow rate, knowing the fresh flow rate calculated in chapter 5, it is possible calculate the re-circulation flow rate, which is the difference between the total one and the fresh air needed to keep a certain value of the indoor air quality.

$$\dot{V}_{rec} = \dot{V}_{tot} - \dot{V}_{fresh} = 29260\left[\frac{m^3}{h}\right] \tag{4.7}$$

The re-circulation air represents the amount of exhaust air, that will be re-used and re-circulate in order to achieve a cooling effect during thesumme, or pre-heat the fresh air coming from outside during the winter, allowing to save energy for cooling and heating processes. In the figure 4.4, we find all the values so far calculated:

Cooling conditions			
Cooling_load	Q_h	149.8	kW
Humidity content	m_v	0.004433	kg/s
	Δ_h/Δ_x	33792	kJ/kg
Internal temperature	t_int	26	°C
Supply temperature	t_supply	18	°C
Temperature difference	Δ_t	8	°C
Enthalpy diagram	h_supply	42	kJ/kg
	h_internal	53	kJ/kg
mass flow rate	m1	13.6	kg/s
Volumetric flow rate	V1	40855	m3/h
Vol_flow_rate_fresh	V_fr	11594	m3/h
Mass_flow_rate_fresh	m_fr	3.86	kg/s
Vol_flow_rate_recirc	V_recirc	29260	m3/h
Mass_flow_rate_rec	m_rec	9.75	kg/s
Air change rate	n	6.16	h^(-1)

Figure 4.4: Cooling data

The next step of the design is the calculation of all the temperatures before and after the heat exchanger, and among the cooling and heating coils. This will allow to calculate the enthalpies of the different points and calculate the power required by each component.
In order to do that, the value of the efficiency of the heat exchanger is required; this value is impossible to know at the beginning, so, based on experience, I took a value from a machines similar to the one under examination in the present studye, and I used that as a comparison for the calculation, going after the design to substitute it. So, the starting data was:

- $\eta_{cooling} = 0.882$
- t_{f1} = temperature fresh air from outside = 35°C
- t_{e1} = temperature exhaust air from inside = 26°C

Instead, the unknown's temperature was:

- t_{f2} = temperature fresh air after the HE
- t_{e2} = temperature exhaust air after the HE
- t_{mix} = temperature of the air after mixing with the recirculating air

From the general equation of the efficiency of the heat exchanger 4.8, we can find t_{f2} and t_{e2} respectively with equations 4.17 and 4.10.

$$\eta_{summer} = \frac{t_{f1} - t_{f2}}{t_{f1} - t_{e1}} \tag{4.8}$$

$$t_{f2} = t_{f1} - \eta(t_{f1} - t_{e1}) = 28°C \tag{4.9}$$

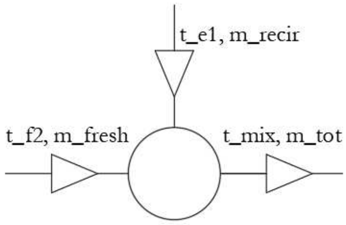


Figure 4.5: Mixing junction

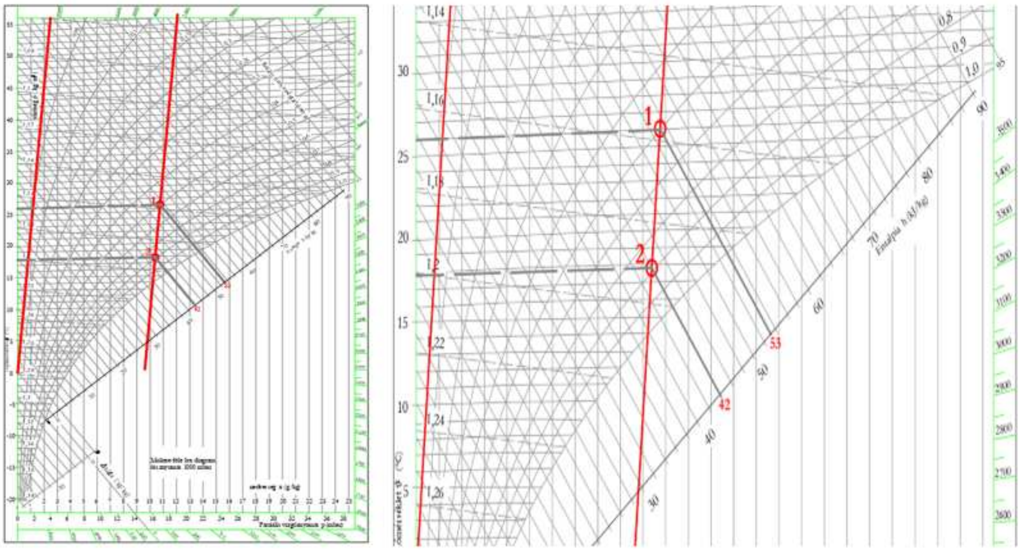


Figure 4.6: Diagram h-x

$$t_{e2} = \eta(t_{f1} - t_{e1}) + t_{e1} = 33°C \tag{4.10}$$

Having calculated all the temperature required, with a mass balance we can calculate the mixing temperature, before the coils, as shown in the figure 4.5
Mass balance:

$$\dot{m}_{tot} t_{mix} = \dot{m}_f resh t_{f2} + \dot{m}_{recirculation} t_{e1} \tag{4.11}$$

And from this equation, I calculated t_{mix}:

$$t_{mix} = \frac{\dot{m}_{fresh} t_{f2} + \dot{m}_{recirculation} t_{e1}}{\dot{m}_{tot}} = 26.59°C \tag{4.12}$$

Now let's come back, and looking on the graph, in order to build the path of the different steps and find the enthalpies. Starting from the point 3, which has the external characteristics of t_{ext}=35°C and $\varphi = 0.4\left[\frac{g}{kg}\right]$, since the heat exchange in the HE is isothermal, we can draw a vertical line until the temperature t_{f2}, reaching the 28°C and $\varphi = 0.6\left[\frac{g}{kg}\right]$, marked with the point 4; so the line 3-4, it's the work carried out from the heat exchanger.
Once at the point 4, we can draw the line that links the point 4 (condition of the air at the outlet of the heat exchanger) to the point 1, that as already said, represent the internal conditions. In the middle of this line, for the mass balance, must be present the point 5, which represent the mixing point. Its temperature should be found with the mass balance in the equation 4.19, which has a temperature of 26.59°C. The next process, is the cooling coil, which as we can see

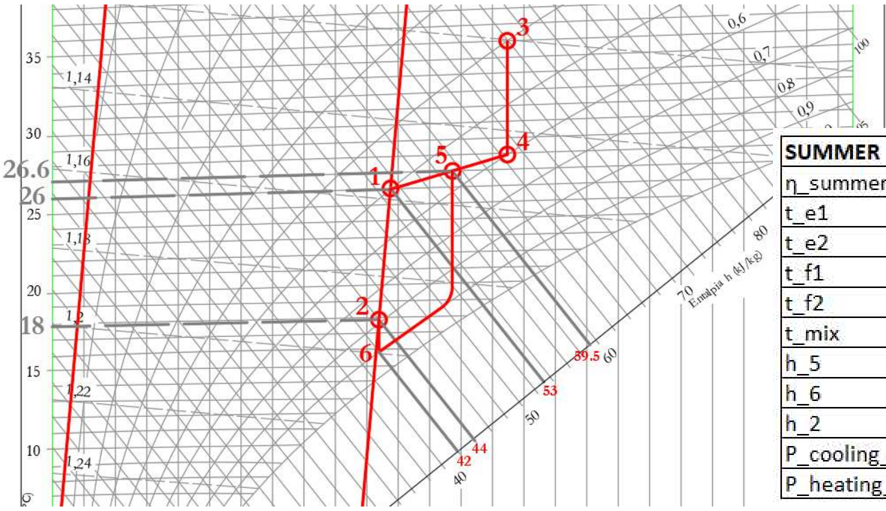


Figure 4.7: Mollier diagram h-x

SUMMER		
η_summer	0.744	-
t_e1	26	°C
t_e2	32.7	°C
t_f1	35	°C
t_f2	28	°C
t_mix	26.65	°C
h_5	59.5	kJ/kg
h_6	42	kJ/kg
h_2	44	kJ/kg
P_cooling_coil	238.3	kW
P_heating_coil	27.2	kW

Figure 4.8: Resuming data

in the graph 4.7 has a curve shape. If the heat exchanger carried on just a sensible cooling, the cooling coil also apply a latent cooling, bringing the air at temperature $t_{s1} = 16°C$ and a relative humidity equal to $\varphi = 0.9\left[\frac{g}{kg}\right]$, reaching the point 6. Finally, we have the heating coil, because we can't inject air at temperature of 16°C, it's too low, so a heating coil is needed, and it applied just a sensible heating, in order to arrive in the conditions of the injection point, marked with the point 2. Once we found all the points described, we can calculate the power required by the heating and the cooling coils with the equation 4.3, which make it possible to size the external system that is in charge of cool down and heat up the water flowing in the coils. In the figure 4.7 is it possible see all the points and all the transformations described. The results obtained are the following:

- Cooling coil: 238.3 kW
- Heating coil: 27.2 kW

4.2 WINTER DESIGN

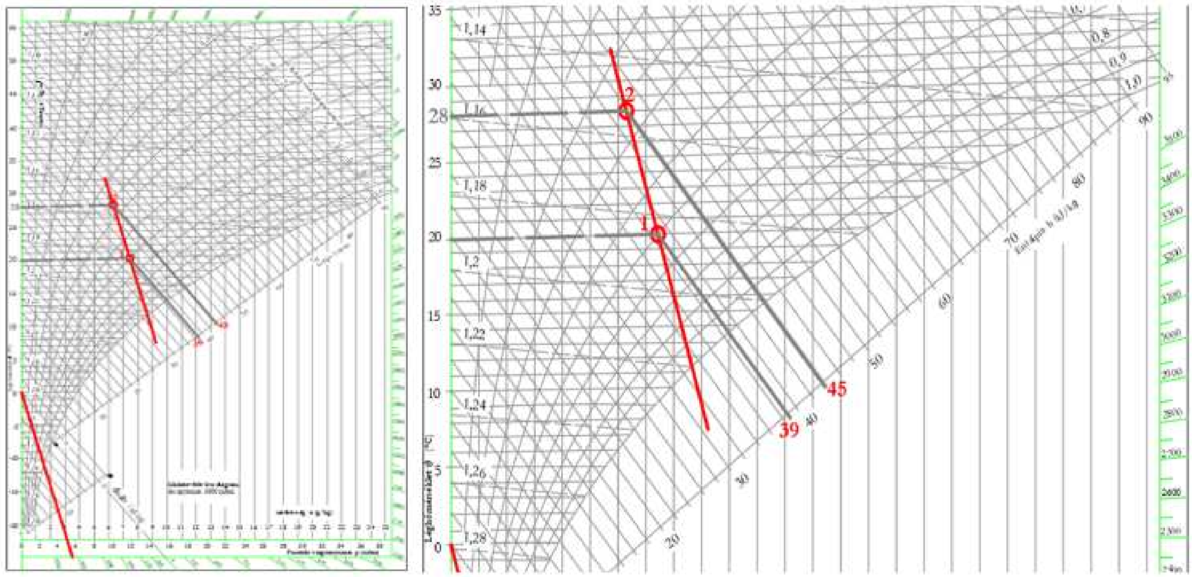


Figure 4.9: Mollier diagram h-x

The process for the winter design is exactly the same as for the cooling one, what change is the process on the diagram, but the calculations are exactly the same. Let's see briefly how to do that.
The starting point remain the heating load, which is 22.8 kW, same calculation for the humidity content, which will have the same value already described; what is different is of course the ratio, due to the lower power required, which is:

$$\frac{\Delta h}{\Delta x} = \frac{Q_{heating}}{m_v} = -5143 \quad \left[\frac{kJ}{kg}\right] \tag{4.13}$$

-DA CAPIRE PERCHè ESCE MENO- From the bottom part of the diagram, we can draw a line that goes from -5143 to zero, and then translate it on the inner point, which has $t_{int} = 20C$ and $\varphi = 0.5\left[\frac{g}{kg}\right]$.
Once again, I fixed the ΔT to 8°C, which fixes the supply temperature at $t_{supply} = 28°C$; so, on the same line, I found the point at 28°C, and consequently the enthalpies of the point 1 and 2. And with equation 4.4 we can calculate $\dot{m}$ and $\dot{V}$.
All the results are showed in the figure 4.10
The only problem so far, is that all the AHU, for both seasons, need to handle the same amount of volumetric flow rate,. However in our case, it happened that:

$$\dot{m}_{summer} > \dot{m}_{winter} \tag{4.14}$$

So, to fix the problem, we have to reduce the supply temperature, so that a higher volumetric flow rate is necessary.
By applying the equation 4.15 is it possible to calculate the enthalpy necessary, and then, from the graph, find the point 3 and find out which is the new supply temperature t* = t3.

$$h3 = h1 + \frac{\sum \dot{Q}_{heating}}{\dot{m}_{cooling}} = 40.67 \quad \left[\frac{kJ}{kg}\right] \tag{4.15}$$

m_summer > m_winter			
t*, m_winter new calculations			
h_supply_winter	h_supply	40.67	kJ/kg
	h_internal	39	kJ/kg
From the diagram t*	t*	23	°C
mass flow rate	m1	13.6	kg/s
Volumetric flow rate	V1	40855	m3/h
Vol_flow_rate_fresh	V_fr	11594	m3/h
Mass_flow_rate_fresh	m_fr	3.86	kg/s
Vol_flow_rate_recirc	V_recirc	29260	m3/h
Mass_flow_rate_rec	m_rec	9.75	kg/s
Internal temperature	t_int	20	°C
Supply temperature	t*_supply	23	°C
Temperature difference	Δ_t	3	°C
Air change rate	n	6.16	h^(-1)

Figure 4.10: Resuming data

With this value, from the graph, we found the temperature t* = 23°C.
Once the new supply temperature was found, the total mass and volumetric flow rate remained the same, and obviously also the recirculation and the fresh ones, including the air change rate, as we can see in the figure.

Still, the next step is to calculate all the temperatures at the end of every process, and then the power required by the heating coil and the humidifier, since we're using as it's improperly called, isothermal humidifier.
The starting data was:

- $\eta_{heating}$ = 0.882
- t_{f1} = temperature fresh air from outside = -15°C
- t_{e1} = temperature exhaust air from inside = 20°C

Instead, the unknown's temperature was:

- t_{f2} = temperature fresh air after the HE
- t_{e2} = temperature exhaust air after the HE
- t_{mix} = temperature of the air after mixing with the recirculating air

And again, from the equation of the efficiency of the heat exchanger:

$$\eta_{summer} = \frac{t_{e1} - t_{e2}}{t_{e1} - t_{f1}} = \frac{t_{f2} - t_{f1}}{t_{e1} - t_{f1}} = 0.843 \tag{4.16}$$

$$t_{f2} = t_{f1} + \eta(t_{e1} - t_{f1}) = 14.5°C \tag{4.17}$$

$$t_{e2} = t_{e1} - \eta(t_{e1} - t_{f1}) + = -9.5°C \tag{4.18}$$

Mass balance:

$$\dot{m}_{tot}t_{mix} = \dot{m}_{f}resht_{f2} + \dot{m}_{recirculation}t_{e1} \tag{4.19}$$

And from this equation, I calculated t_{mix}:

$$t_{mix} = \frac{\dot{m}_{fresh}t_{f2} + \dot{m}_{recirculation}t_{e1}}{\dot{m}_{tot}} = 18.44°C \tag{4.20}$$

Also, in this section, the efficiency has been taken from previous designs and then substituted. In the diagram, the process to find the mixing temperature it's the following.
Starting from point 4 which represent the external conditions of -15°C and $\varphi = 0.8\left[\frac{g}{kg}\right]$, I traced a vertical line, in order to reach the t5 = t_{f2}, which is at temperature of 14.505°C (point 5). From there, we can draw a line that link point 5 with point 1. On this line, somewhere, there will be the mixing temperature, as calculated, is at the temperature of the 18.44°C, so, we can identify the point 6. Then, after the heat exchanger and the mixing flow, air must pass through two other different process, heating, due to the heating coil, and humidification. Heating coil face just a sensible heat, and we have to heat up the flow, until reaching the inlet temperature previously defined. So drawing a vertical line, from the point 6, we reach the point 7 on the isothermal line of the 23°C. Checking the enthalpies of the two point, it is possible to calculate the design power of the heating coil (equation 4.3). The last step is to calculate the power required from the humidifier, which is forcing water vapour inside the mass of air, and in order to do that, a certain amount of energy is required, to produce the necessary steam. Also, here we can check the enthalpies and see the results below.
As we can see from the schemes of the AHU, figures 4.2 and 4.3, the humidifier it is not present in the unit, but it will be installed in the duct system; this lead to different advantages, because keeping the humidifier inside, means avoiding problem of freezing during the coldest days, that also mean less holes through the layers of the building structure, and also, the ducts which goes to the humidifier do not have to be isolated.

- Heating coil = 61.3 [kW]
- Humidifier = 69.5 [kW]

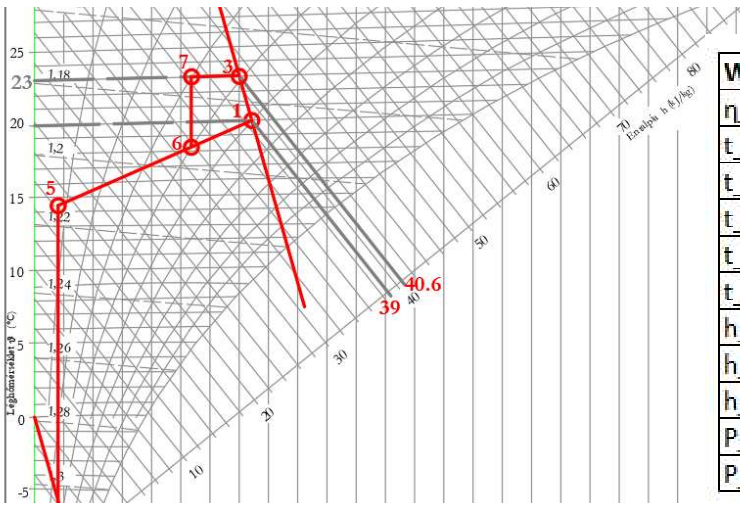


Figure 4.11: Mollier diagram h-x

WINTER		
η_winter	0.843	-
t_e1	20	°C
t_e2	-9.505	°C
t_f1	-15	°C
t_f2	14.505	°C
t_mix	18.44	°C
h_3	40.6	kJ/kg
h_6	31	kJ/kg
h_7	35.5	kJ/kg
P_heating_coil	61.3	kW
P_humidifier	69.5	kW

Figure 4.12: Resuming data

Chapter 5

Fresh air flow rate estimation

5.1 Based on the occupants

The ventilation required based on the number of occupants, is easily determined once it's clear, the number of maximum occupants that could be present at the same time in the building is clear. It's a rough estimation, based on the average number of occupants, because it's really difficult to know exactly which will be the real number.
In order to do that, I have divided the floor office that we are considering, into the micro-ambient, i.e. the smallest rooms into the floor, for examples, restrooms, office, kitchen, meeting room, classroom etc. At each of them, according to the standard [6], I assigned an average occupancy, dividing it into the following categories, table 5.1, which the relatives occupancy:

Type of space	**Acronym**	**Occupancy Person/m2**
Office	O	0.05
Cafeteria	CA	0.7
Conference room	C	0.5
Assembly	A	1.5
Classroom/school	**S**	**0.5**

Table 5.1: Rooms average occupancy

Multiplying the square meters of the room, for the value assigned, what we have obtained is the maximum[1] number of people expected at the same time in the floor, which is 319.
After having estimated the number of people, the method based on the occupancy, said to multiplies the number obtained times the liters per second per person that begin to the different categories A,B and C, which are respectively 10,7 and 4 [l/s person] showing the results in table 5.2

	[l/s]	[m3/h]
Cat A	3192	**11490**
Cat B	2234	**8043**
Cat C	1277	**4596**

Table 5.2: Ventilation rate based on the occupancy rate

[1]Based on the average

	Operative temperature [°C]	Mean air velocity [m/s]
Cat A	24.5 ± 1.0	0.18
Cat B	24.5 ± 1.5	0.22
Cat C	24.5 ± 2.5	0.25

Table 5.3: Initial thermal design criteria

5.2 Ventilation rate based on the square meters

This is another simple calculation, but that could be helpful for a rough estimation of the ventilation rate. It's based on the square meters of the office and the pollution of the building. Other two assumption should be added for this method:

- The occupancy is 0.07 $\frac{person}{m^2 floor}$
- The cooling load caused by occupants, machines, illumination, solar radiation etc. is 50 $\frac{W}{m^2 floor}$

It is due to make a premise in this regard, as a parameter for the objective assessment of pollutant sources must be introduced. From definition the Olf is a parameter used to quantify the pollutant emissions from sources, is defined as the rate of pollutants emitted by a standard person under thermal comfort and normal hygienic habits. Each pollutant source is then expressed through the power of the source equivalent, defined as the number of standard people (expressed in olf) required to cause the same dissatisfaction caused by the actual pollutant source. Should be defined also the operative temperature and mean radiant velocity during the summer for the three categories, reported in table 5.3:
The sensory pollution load is defined as:

Occupants:	1x0.1=	0.1 olf/(m2 floor)
Building:		0.1 olf/(m2 floor)
Total sensory pollution load:		0.2 olf/(m2 floor)

Once we have calculated the total sensory pollution level, we can calculate the required ventilation rate with the equation:

$$Q_c = 10 * \frac{G_c}{C_{c,i} - C_{c,o}} * \frac{1}{\varepsilon_v} \tag{5.1}$$

Where:

- Q_c = is the ventilation rate required for comfort, in liters per second [l/s]
- G_c = is the sensory pollution load, in olf [olf]
- $C_{c,i}$ = is the desired perceived indoor air quality, in decipol [decipol]
- $C_{c,o}$ = is the perceived indoor air quality at air intake, in decipol [decipol]
- ε_v = is the ventilation effectiveness

Which lead to the results showed in table

	l/s(m^2 floor)	l/s	m^3/h
Cat A	1.70	3221	**11594**
Cat B	1.21	2300	**8281**
Cat C	0.68	1288	**4638**

Table 5.4: Ventilation rate based on square meters and pollution

5.3 Ventilation rate based on CO_2

The starting point of this calculation method, is the following equation:

$$Q_h = \frac{G_h}{C_{h,i} - C_{h,o}} * \frac{1}{\varepsilon_v} \tag{5.2}$$

Where:

- Q_h = is the ventilation rate required for health, in litres per second [l/s]
- G_h = is the pollution load of a chemical, in micrograms per second [μg/s]
- $C_{h,i}$ = is the guideline value of a chemical, in micrograms per litre [μg/l]
- $C_{h,o}$ = is the outdoor concentration of a chemical at air intake, in micrograms per litre [μg/l]
- ε_v = is the ventilation effectiveness

The results are show in table 5.5

	l/s(m^2 floor)	l/s	m^3/h
Cat A	0.7	1326	**4774**
Cat B	0.5	947	**3410**
Cat C	0.3	568	**2046**

Table 5.5: Ventilation rate based on CO2 concentration

Briefly looking at results, the first thing that can be said is that the last method, has results, with all the probability too low to satisfy the thermal comfort condition, carried out in order to keep low the concentration of CO2 in the air, thus it is not enough to satisfy the levels of the indoor air quality. Instead, on the other hand, choosing one of the other two ventilation rate, means automatically matching the CO2 required standard. We can see, for instance taking into consideration the category A, the two values, respectively 11490 and 11594 $\frac{m^3}{h}$, using different methods, lead to very close results, ensuring to bring inside enough air to cover the excess of CO2. That also highlight that the two methods, based on the occupants, and on the square meters plus building pollution, are quite the same, without big differences. In order to ensure safety, we are always inclined to choose the bigger value between the options in order to match the thermal conditions even in the worst cases.

Chapter 6

AHU - Air handling unit

The main purpose of the thesis, is going to design an air handling unit system, also called AHU. The air handling unit system, is the heart of the central air conditioning, which is a huge indoor unit of commercial air conditioning equipment. It collects outside air and room air, removes dust and other particles from the collected air, adjusts the temperature and humidity and then supplies comfortable and refreshing air-conditioned air into the rooms through ducts. In the oldest installation, there was just one large AHU[1], usually located on the roof which was supposed to supply the entire building. They often didn't have a return duct, some older designs rely on the air just leaking out of the building. This design is not so common anymore in new buildings because it is very inefficient, now it is most common to have multiple smaller AHU's supplying different zones with supply and return ducts, and it is for this reason that, we're going to design an AHU for the ground floor of the building examinated. Further, the units will have a duct run to pull the used dirty air out of the rooms, back to the AHU, where a fan will discharge it back into the atmosphere. Some of this return air will be recirculated back into the fresh air supply in order to save energy.

6.1 Operation and components

After having calculated all the volumetric flow rates, the supply temperature and the temperature after the cooling coil; defined the type of heat exchanger, the position and the location of the machine, it has been possible send all the data reported in the figure 6.1, to the Rosemberg manufacturer, which, with the data given, was able to find out which was the best option in order to match our necessity. In the following paragraph will be introduce the components of the AHU that has been chosen. Of course, we're talking about a two line system, one for the supply side, and the other for the exhaust side, with recirculation of the exhaust air.

[1]The newest AHU are often installed in the different floors of the building or on the rooftop.

Summary table		
Volumetric flow rate_Total	40855	m3/h
Volumetric flow rate_Recirculation	29260	m3/h
Volumetric flow rate_Fresh_air	11594	m3/h
Pressure drops	450	Pa
Supply air temperature_winter	23	°C
Supply air temperature_summer	18	°C
Temperature fresh air_winter	-15	°C
Temperature fresh air_summer	35	°C
Temperature after the cooling coil	16	°C
AHU position	Horizontal	
Internal/external	External	
Heat exchanger	Plate heat exchanger	

Figure 6.1: Resuming data

Let's see the the components one by one.

- **Dumpers**

A dumper is composed by multiple sheets of metal which can rotate. It can be opened or closed to prevent air from entering or exiting, it can open to fully allow air in or out, and it can also vary their position somewhere in between to restrict the amount of air flow, thus the air entering and exiting. Both sides, the supply and the exhaust, are also equipped with a grill at the end of the duct in order to prevent to external agents to get into the principle line.

Material	Quantity	$H[m]$	$\Delta P[Pa]$
Steel	**2**	1.15	**2**

Table 6.1: Dampers

After the grill and the dumper, we find the:

- **Filters**

Located after the dumpers, filters have the functionality to filter all the dirt and dust present in the exhaust air; from entering the AHU and the building. Usually the most common filters present in the AHU, are the **bag-filters** which utilizes a broad spectrum activated carbon layer to ensure removal of a very wide range of airborne chemicals. The broad spectrum carbon operates with a rapid adsorption dynamics (RAD) mechanism that ensures high efficiency against the multiple chemicals typically present in city-centre buildings. A high media area ensures high efficiency, long life and low pressure drop. Across each bank of filters, there must be a pressure sensor. This will measure how dirty the filters are and warn the engineers when it is time to replace them. As the filters pick-up dirt, the amount of air that can flow through is restricted and this causes a pressure drop across the filters. Typically, we have some panel filters or pre-filters to catch the largest dust particles like the "bag filter short ISO ePM10 50% Sup (supply) Z1 C2" in our case, and then, there are other bag filters which catch the smaller dust particles, for instance the "bag filter HE ISO ePM1 60% - Sup (supply) Z5 C1" for our machine. Also, in our case, in the air supply we can find both of the filters, instead, in the exhaust side, we find just the Z5 C1, because we don't need to purify the air from the biggest particles that comes from outside, but just the pollution from within the office.

- **Heat exchanger**

During most of the year, the air temperature outside air deviates from the conditions required for the air supply and therefore heat treatment is necessary. To minimise the energy consumption, the mass flow rate is subject to a treatment in the heat recovery system. Since the entry of the Ecodesign Regulation (EU) No 1253/2014 [4] the installation of a heat recovery unit has become mandatory in the European Union. A recuperator transfers thermal energy from the warmer air flow to the colder one in both seansons. Specifically in our winter case, this means for example that the air outside would be heated from -15°C to 14.5°C only using the heat of the extract air, which would be cooled from 20°C to -9°C. Consequently, the energy demand of heating for the downstream heating coil is much lower. In summer periods, the opposite operation and the recuperator reduces the cooling demand. Normally, into the AHU there are two different types of recuperators. First, there are the regenerative systems, which in most cases are rotary heat exchangers. A wheel rotates through both air streams transferring thermal energy from one flow to the other. With rotary heat exchangers there is always a contamination between the outside air flow and the extract air flow. This must be taken into account in cases where there is the necessity to account for very high hygienic requirements. Rotary heat exchangers can transport moisture, which can then be recovered. Heat exchangers can also dehumidify, with a positive impact on the cooling power during the summer season.
Morevover a static recovery system, like in our case, in which the air outside and extract air pass through very small ducts. This system ensures that both air flows exchange thermal energy with the help of the surfaces over which they are passing. Normally plate heat exchangers are employed. With metal heat exchangers, no moisture is transferred and in winter condensation can occur and in the case of low temperatures, even frost. Some plate heat exchangers have permeable membranes that also allow the transfer of moisture. Static recovery systems have less cross-contamination between air flows compared to rotary heat exchangers.

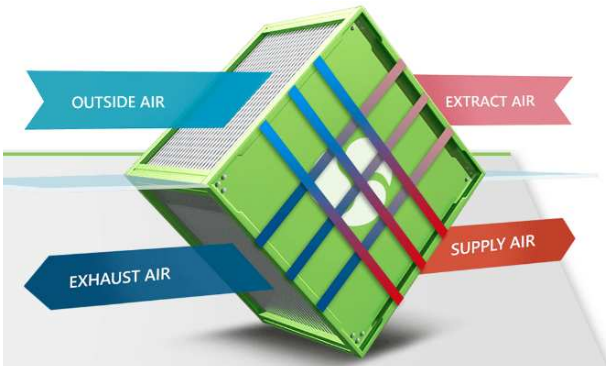


Figure 6.2: Plate heat exchager

Figure 6.3: Example of plate heat exchanger inserted into a AHU

It is important to notice that, the manufacturer provides the heat exchanger the winter and the summer efficiency, which has been used to calculate the temperature of the stream after the heat exchanger and calculate the power for the different components in the chapter 4:

- $\eta_{winter} = 0.843$
- $\eta_{summer} = 0.744$

All the technical data will be showed in the chapter DA CONCLUDER

Taking into account the winter season, we find the:

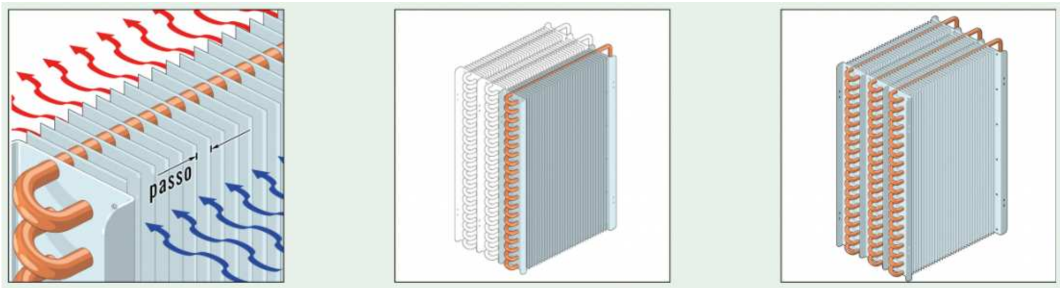


Figure 6.4: Coil's scheme

- **Pre-heating coil** - Winter season

As we can easily see in the figure 4.3, the pre-heating coil is placed after that the flow of air which is already passed through the heat exchanger, and after that the fresh air, has been mixed with the exhaust side. It's placed at the inlet of the fresh air intake. This is usually an electrical heater. The heater will switch on and heat the air up to bring the air at the right supply temperature, and at the same time, protect the components inside from frost, otherwise too low temperatures could freeze inside of the heating and cooling coils. During the winter, this is the only coil that works, the post heating coil, works only during the summer and it is not necessary during winter.
For our case, after the calculation I found that 61 kW were necessary for the supply temperature, the manufacturer design a heating coil with a heating capacity of 104 kW, supplying water at 60°C.

- **Cooling and post heating coils** - Summer season

Basically, the purpose of the two coils, is to heat or cool down the air. Contrary with the pre-heating coil, which could be electric, these two are heat exchangers, they carry a hot or a cold fluid, depending on season and on the coil, usually heated or chilled water, refrigerant or steam. Depending on the type of plant (full-air or air and water based solution) the coils have to heat, cool and dehumidify the air. In fact, the cooling battery can be used in two different ways. Firstly, it can be used dry. When the water content of supply air and outside air is similar, condensation does not occur. In addition to dry operation, it can be used for dehumidification: condensation water must be drained from the AHU via a condensate tray and piping. They are regulated by modulating the water flow rate with a 3-way valve in constant-flow systems, or 2-way in variable-flow systems. Modulating regulation is preferred because the heat exchange inside the coils has a very low thermal inertia. An ON/OFF type regulation would cause continuous switching on and off of the coil leading to excessive fluctuations in the supply air temperature.

In the figure 6.4 is showed how a heating or a cooling coil is composed. It is composed by the:

- Pitch = is the distance between the fins
- Row = is the set of finned tubes traversed by the heat transfer fluid

The set of rows forms the actual coil.

	Calculation [kW]	Manufacturer design [kW]	Supply temperature °C
Heating	27.2	104	60
Cooling	238.3	292	7

Table 6.2: Heating and cooling capacities

- **Fans**

At the end of the chain there is the fans, one for the supply side which is employed to bring inside the room fresh air, and one for the exhaust side, which instead, is responsible for the exhaust air, and both, are required to move the air that increases the pressure to overcome all the load losses. Motors for fans must always be equipped with speed controllers to ensure that only the required air flow rate is generated. This speed regulation can be achieved with frequency converters like in the case of AC motors or directly with control electronics included in the case of EC[2] fans. Usually the fans, represent the bigger electric consumption required from the system, and should be design carefully.

- **Silencers**

In order to avoid annoyance, one silencer could be installed after the supply fan and another one before the exhaust fan. There are many options for the silencers, they can be rectangular or circular. If the AHU is located in a quiet urban area and therefore may affect the neighbours a silencer has to be installed also after the intake air grill and before the exhaust grill to reduce the noise in the surroundings. But in our case, no silencers have been installed, because the machine is too big to be kept inside.

- **Humidifier**

The last component of the system is the humidifier. The humidifier is usually situated at the end of the heating and cooling coils, but it is not a standard rule. It's placed there in application with low power and mild climate. In application like the case that I'm studying, it's preferable to keep the humidifier inside the line of the pipe, for several advantages:

- First of all we avoid the problem of condensation, due to the contact of the vapour with the pipe wall which could be at a lower temperature below the dew point.
- Second,it is not necessary to insulate the pipes that carry the water that is going to evaporate and then injected in the air flow
- And third, we can avoid creating holes and lines that run from the external to internal side and vice versa.

There are mainly two different types of humidifier. **Adiabatic** humidifiers which produce direct evaporation of water in the air without energy administration from outside and thus without causing a rise in temperature. The heat needed for vaporization is provided by the humidified air, which therefore cools. These devices provide a large interface surface between air and water in the liquid state, on the surface of which a thin layer of saturated vapor forms at a partial

[2] They are motors that can work at maximum efficiency at all times and provide significant energy savings over conventional asynchronous (i.e., alternating current) motors

pressure equal to the saturation pressure at the temperature of the liquid itself. Where this pressure is greater than the partial pressure of the vapor in the air, a pressure gradient exists that fuels the progressive evaporation of the liquid at the expense of the sensible heat of the water and air. This kind of humidifier, shows several advantages, like:

- Low consumption of electric energy
- Significant capacity and good regulation, from few kg/h to thousands of kg/h
- They don't require too much maintenance

Instead, in the case of **isothermal** humidifiers, these force water vapour inside the mass of air, but contrary to the adiabatic ones, this process requires external energy to produce the necessary steam. Because the mass of the vapor is much less than the mass of the air in which it is absorbed, the transformation causes a modest increase in air temperature, which is why it is improperly called isothermal humidification. As we can see from the picture 4.11, from the point 7 to 1, in the humidification process, the temperature remains the same. These humidifiers ensure the maximum hygienic conditions, due to the high temperature of the steam. Still from the pictures 4.11 and 4.12, we can see that the humidifier must be properly design, because it requires a massive amount of power.

The following pages, are reporting the technical data of the AHU designed by Rosemberg manufacturer.

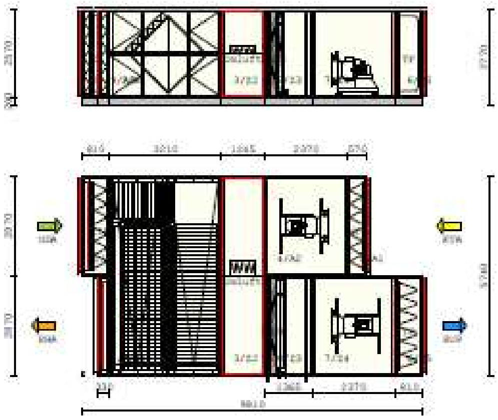
Rosenberg Hungária Kft.
2532 Tokodaltáró,
József Attila út 34.

Pers. of Cont.:	**horvath.peter**	**Date:**	**2023. 05. 16.**	**Order**	-
Project No:	**Unideb**	**Pos-No:**	**AHU-01**	**Customer:**	
Project:	**Unideb**			**RZ-No:**	

Supply:	Airbox S60-2825	Height:	2770mm	Width:	5740mm	Weight:	10550kg
Exhaust:	Airbox S60-2825	AHU-S602825WW		Length:	9810mm	S60 - 60mm Insulation	

Weather Proof Design

Reference City Eurovent: Hungary/DEBRECEN



Heat recovery - side by side

Supply: Vol=40855m³/h dPext=450 Pa V=1,7m/s (V2)

Sound Power Levels:

	63Hz	125Hz	250Hz	500Hz	1kHz	2kHz	4kHz	8kHz		Total		
At outer Panel:												
Lw	82	72	64	62	61	59	57	41	dB	LwA	67	dB(A)
At Inlet Side:												
Lw	65	70	65	64	61	54	42	31	dB	LwA	65	dB(A)
At Outlet Side:												
Lw	85	88	84	83	82	78	75	71	dB	LwA	86	dB(A)

Casing data according to DIN EN 1886:
Thermal transmittance class: T2 (M); Thermal bridging class: TB3 (M); Leakage class +/- pressure: L1/L2(M);
Mechanical strength: D1 (R)

Exhaust: Vol=40855m³/h dPext=450 Pa V=1,7m/s (V2)

Sound Power Levels:

	63Hz	125Hz	250Hz	500Hz	1kHz	2kHz	4kHz	8kHz		Total		
At outer Panel:												
Lw	82	72	64	62	61	59	57	41	dB	LwA	67	dB(A)
At Inlet Side:												
Lw	80	83	79	78	77	73	70	66	dB	LwA	81	dB(A)
At Outlet Side:												
Lw	82	84	80	77	76	72	68	65	dB	LwA	81	dB(A)

Casing data according to DIN EN 1886:
Thermal transmittance class: T2 (M); Thermal bridging class: TB3 (M); Leakage class +/- pressure: L1/L2(M);
Mechanical strength: D1 (R)

Frame parts, corner connectors and connecting lugs are in contact with air on the inside. Corner connectors and connecting lugs are made of plastic or AlMg3. Maximum over/under pressure [Pa]: +2500/-3000

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Figure 6.5: Technical data AHU

Rosenberg Hungária Kft.
2532 Tokodaltáró,
József Attila út 34.

Pers. of Cont.:	horvath.peter	Date: 2023. 05. 16.	Order	-
Project No:	Unideb	Pos-No: AHU-01	Customer:	
Project:	Unideb		RZ-No:	

Supply:	Airbox S60-2825	Height: 2770mm	Width:	5740mm	Weight:	10550kg
Exhaust:	Airbox S60-2825	AHU-S602825WW	Length:	9810mm	S60 - 60mm Insulation	
		Weather Proof Design				

Technical Data

Sup Z1 C1 air damper steel galv. (EN 1751 Kl.2) size 1 entire cross section; 2 piece; W=2548; H=1158; flange=30

dp Air:	2 Pa		

Sup Z1 C2 bag filter short ISO ePM10 50% (M5), frame AISI304 (V2A)

Filter length:	370 mm	Filter surface:	46,8 m²
Air Velocity:	1,81 m/s		
dP Calc:	68 Pa	Energy-Class:	E
dP Start:	34 Pa	dP Ende (EN13053:2019):	102 Pa
Measures:	16x592², 4x287x592,		

Sup W1 C1 PWT-H2A1500-2825-N

Exchanger Type:	H2A1500-4837-030-2EOO-2-3-6-5607		
Recovery Factor wet:	84,3 %		
Air Volume Supply:	40855 m³/h	Air Volume Exhaust:	40855 m³/h
Temp outside Air:	-15 °C	Temp. Exhaust:	21 °C
Humidity outside Air:	90 %r.F.	Humidity Exhaust:	45 %r.F.
Temp. Supply:	15,4 °C	Temp out Exhaust:	-1,05 °C
Humidity Supply:	10,4 %r.F.	Humidity out Exhaust:	97,7 %r.F.
dP Supply:	174 Pa	dP Exhaust:	180 Pa
Heating Power:	411 kW	Condensate Flow:	167,04 l/h
Kondensations Temp.:	9 °C		

Temperature outside Air = Design outdoor Temperature (Eurovent)
Leakage: <0,1% of rated airflow for favourable fan configuration
Data according EN 13053 (2012-02) at dry reference conditions and balanced air mass flow

Recovery Factor dry:	74 %	Efficiency:	71,2 %
dp Supply (Std. Density):	188 Pa	dp Exhaust (Std. Density):	187 Pa
Power Cons. Due to dp:	7,09 kW	Heat Recovery Class (2012/02):	H2
		Heat Recovery Class (2020/05):	H2

Operating Point 2:

Air Volume Supply:	40855 m³/h	Air Volume Exhaust:	40855 m³/h
Temp outside Air:	35 °C	Temp. Exhaust:	26 °C
Humidity outside Air:	40 %r.F.	Humidity Exhaust:	50 %r.F.
Temp. Supply:	28,3 °C	Temp out Exhaust:	32,7 °C
Humidity Supply:	58,6 %r.F.	Humidity out Exhaust:	34 %r.F.
dP Supply:	192 Pa	dP Exhaust:	190 Pa
Total Power:	91,6 kW	Recovery Factor wet:	74,4 %

Sup Z3 C1 Heater PWW Cu/Al Frame FeZn

Number of Rows:	1	Air Volume:	40855 m³/h
Temp. Air in:	15,4 °C	Temp Medium in:	60 °C
Temp. Air out:	23 °C	Temp Medium off:	40 °C
dp Air dry:	8 Pa	dp Medium:	20,3 kPa
Heating Capacity:	104 kW	Medium flow rate:	4,91 m³/h
Surface Reserve:	4,9 %	Percentage Glycol (Ethylene):	35 %
		Medium Volume:	27 l
Air Velocity:	1,89 m/s	No of Circuits:	15
		Pipe Connection:	DN40 - 1 1/2"
Code:	HW 12 3329V4.0 70T2572 1R 90		

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Figure 6.6: Technical data AHU

Rosenberg Hungária Kft.
2532 Tokodaltáró,
József Attila út 34.

rosenberg THE AIR MOVEMENT GROUP

Pers. of Cont.:	**horvath.peter**	**Date:**	**2023. 05. 16.**	**Order**	**-**
Project No:	**Unideb**	**Pos-No:**	**AHU-01**	**Customer:**	
Project:	**Unideb**			**RZ-No:**	

Supply:	Airbox S60-2825	Height:	2770mm	Width:	5740mm	Weight:	10550kg
Exhaust:	Airbox S60-2825	AHU-S602825WW	Length:	9810mm	S60 - 60mm Insulation		
		Weather Proof Design					

Sup Z3 C2 Empty Casing

dp Air:	0 Pa

Sup Z3 C3 Cooler PKW Cu/Al Frame AISI 304/1.4301

Number of Rows:	4	Air Volume:	40855 m³/h
Temp. Air in:	28,3 °C	Temp Medium in:	7 °C
Temp. Air out:	16 °C	Temp Medium off:	12 °C
Humidity Air in:	58,6 %r.F.	Medium flow rate:	56,5 m³/h
Humidity Exhaust:	93,3 %r.F.	dp Medium:	44,7 kPa
dp Air wet:	69 Pa	Percentage Glycol (Ethylene):	35 %
dp Air dry:	58 Pa	Medium Volume:	103 l
dP Drop Seperator:	23 Pa	Condensate Flow:	172 l/h
Air Velocity:	1,99 m/s	No of Circuits:	89
Cooling Capacity:	292 kW	Pipe Connection:	DN80 - 3"
Surface Reserve:	2,04 %		

Code: CW 12 3329V2.8 67T2532 4R 200
Fan designed for pressure drop of wet conditions

Sup Z3 C4 Heater PWW Cu/Al Frame AISI 304/1.4301

Number of Rows:	1	Air Volume:	40855 m³/h
Temp. Air in:	16 °C	Temp Medium in:	60 °C
Temp. Air out:	18 °C	Temp Medium off:	40 °C
dp Air dry:	8 Pa	dp Medium:	20,6 kPa
Heating Capacity:	27,7 kW	Medium flow rate:	1,32 m³/h
Surface Reserve:	275 %	Percentage Glycol (Ethylene):	35 %
		Medium Volume:	22 l
Air Velocity:	1,87 m/s	No of Circuits:	7
		Pipe Connection:	DN20 - 3/4"
Code:	HW 12 3329V4.0 70T2592 1R 90		

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Figure 6.7: Technical data AHU

Rosenberg Hungária Kft.
2532 Tokodaltáró,
József Attila út 34.

Pers. of Cont.:	**horvath.peter**	**Date:**	**2023. 05. 16.**	**Order**	**-**
Project No:	**Unideb**	**Pos-No:**	**AHU-01**	**Customer:**	
Project:	**Unideb**			**RZ-No:**	

Supply: Airbox S60-2825 Height: 2770mm Width: 5740mm Weight: 10550kg
Exhaust: Airbox S60-2825 AHU-S602825WW Length: 9810mm S60 - 60mm Insulation
Weather Proof Design

Sup Z4 C1 Fan free-run, stand. Motor RLM E6-9010-63-33-A

Fan Data

Air Volume:	40855 m³/h		
dP total:	995 Pa	dP dynamic:	99 Pa
dP stat:	916 Pa	dP Mounting:	10 Pa
dP external:	450 Pa	dP internal:	466 Pa
Shaft Power:	14,1 kW		
Efficiency (tot):	80 %	Sound Power Level LwA6:	91 dB(A)
Speed:	1086 U/min	max. Speed:	1165 U/min
VFD-Frequency:	55 Hz	VFD-Frequency (max.):	59 Hz
Speed Reserve:	79 U/min	Efficiency Fan (stat):	66 %
dP inlet Cone:	1589 Pa	Calibration Factor k10:	794 m²s/h
SFPv:	1,26 kW/(m³/s) (SFP3)		

Motor Nominal Data

Size:	200L / IE3-6	Nominal Power:	18,5 kW
Nominal Voltage:	3~400-50	Motor Protection:	PTC
Power Consumption (Pel):	15,7 kW	Nominal Current:	37 A
Nominal Speed:	975 U/min		
Nennfrequenz:	50 Hz		
Efficiency (Design):	- %	Efficiency-Class:	IE3

Efficiency-Class

Velocity Class:	V2		
Power Consumption Class:	P1	Pm_ref (DIN EN 13053):	19,3 kW

Fan operation data fulfil accuracy class 1 according DIN 24166

Sup Z5 C1 bag filter HE ISO ePM1 60% (F7)

Filter length:	635 mm	Filter surface:	136 m²
Air Velocity:	1,81 m/s		
dP Calc:	103 Pa	Energy-Class:	C
dP Start:	53 Pa	dP Ende (EN13053:2019):	153 Pa
Measures:	16x592², 4x287x592,		

Exh A1 C2 bag filter short ISO ePM10 50% (M5)

Filter length:	370 mm	Filter surface:	46,8 m²
Air Velocity:	1,81 m/s		
dP Calc:	68 Pa	Energy-Class:	E
dP Start:	34 Pa	dP Ende (EN13053:2019):	102 Pa
Measures:	16x592², 4x287x592,		

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Figure 6.8: Technical data AHU

Rosenberg Hungária Kft.
2532 Tokodaltáró,
József Attila út 34.

Pers. of Cont.:	horvath.peter	Date:	2023. 05. 16.	Order	-
Project No:	Unideb	Pos-No:	AHU-01	Customer:	
Project:	Unideb			RZ-No:	

Supply:	Airbox S60-2825	Height:	2770mm	Width:	5740mm	Weight:	10550kg
Exhaust:	Airbox S60-2825	AHU-S602825WW	Length:	9810mm	S60 - 60mm Insulation		
		Weather Proof Design					

Exh A2 C1 Fan free-run, stand. Motor RLM E6-9010-63-33-A

Fan Data			
Air Volume:	40855 m³/h		
dP total:	839 Pa	dP dynamic:	99 Pa
dP stat:	760 Pa	dP Mounting:	59 Pa
dP external:	450 Pa	dP internal:	310 Pa
Shaft Power:	12,2 kW		
Efficiency (tot):	78 %	Sound Power Level LwA6:	91 dB(A)
Speed:	1043 U/min	max. Speed:	1165 U/min
VFD-Frequency:	53 Hz	VFD-Frequency (max.):	59 Hz
Speed Reserve:	122 U/min	Efficiency Fan (stat):	64 %
dP inlet Cone:	1589 Pa	Calibration Factor k10:	794 m²s/h
SFPv:	1,15 kW/(m³/s) (SFP3)		
Motor Nominal Data			
Size:	200L / IE3-6	Nominal Power:	18,5 kW
Nominal Voltage:	3~400-50	Motor Protection:	PTC
Power Consumption (Pel):	13,6 kW	Nominal Current:	37 A
Nominal Speed:	975 U/min		
Nennfrequenz:	50 Hz		
Efficiency (Design):	- %	Efficiency-Class:	IE3
Efficiency-Class			
Velocity Class:	V2		
Power Consumption Class:	P2	Pm_ref (DIN EN 13053):	15,2 kW
Options:	Inlet protection		

Fan operation data fulfil accuracy class 1 according DIN 24166

Exh A3 C2 air damper steel galv. (EN 1751 Kl.2) size 1 entire cross section; 2 piece; W=2548; H=1158; flange=30

dp Air: 2 Pa

Air Density: 1,2 kg/m³, Barometric Pressure: 1013,25 hPa
SFP-Class (EnEV): under validation load conditions including bonus if heat recovery class H1 or H2

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Figure 6.9: Technical data AHU

Rosenberg Hungária Kft.
2532 Tokodaltáró,
József Attila út 34.

Pers. of Cont.:	**horvath.peter**	**Date:**	**2023. 05. 16.**	**Order**	**-**
Project No:	**Unideb**	**Pos-No:**	**AHU-01**	**Customer:**	
Project:	**Unideb**			**RZ-No:**	

Supply: Airbox S60-2825 Height: 2770mm Width: 5740mm Weight: 10550kg
Exhaust: Airbox S60-2825 AHU-S602825WW Length: 9810mm S60 - 60mm Insulation
Weather Proof Design

Type:	**NRVU** Non-Residential-Ventilation-Unit	Unittype:	**ZLA (Engl.: BVU)** Bidirectional-Ventilation-Unit
Type of drive:	**multi-speed drive**		

SFPint:	746 W/(m³/s)
SFPint Limit (2016):	1110 W/(m³/s)
SFPint Limit (2018):	830 W/(m³/s)

100% ErP ready 2018

Properties:		
Type of HRS:	**Plate Exchanger**	
Thermal efficiency (EN13053):	**74 %**	
External leakage rate (+400Pa / -400Pa):	**0 %**	**/ 0 %**
Internal leakage rate (250Pa):	**0,87 %**	

Supply		**Exhaust**	
Air volume:	**11,3 m³/s**	Air volume:	**11,3 m³/s**
Δpsext:	**450 Pa**	Δpsext:	**450 Pa**
Input Power:	**15,7 kW**	Input Power:	**13,6 kW**
Δpsint:	**241 Pa**	Δpsint:	**221 Pa**
Δpsadd:	**225 Pa**	Δpsadd:	**89 Pa**
Velocity:	**1,7 m/s**	Velocity:	**1,7 m/s**
ηstat(327/2011/EU): / Eff. Grade N	**71,5 % / 70,8**	ηstat(327/2011/EU): / Eff. Grade N	**71,5 % / 70,8**
Filter:	**F7**	Filter:	**M5**
Sound power level:	**67 dB(A)**	Sound power level:	**67 dB(A)**

General:	
Disassembly instruction	Can be found on www.rosenberg-gmbh.com
Installed fans fulfil Commission regulation (EU) no. 327/2011	
Filter warning:	To fulfil the regulation 1253/2014/EU the installed filters need to be equipped with a visual signal or an alarm in the control system which shall be activated as soon as the filter pressure drop exceeds the maximum final pressure drop.
Thermal bypass:	The thermal bypass is realized by a mechanical bypass.
Drive:	If the ventilation unit is ordered and delivered without multi-speed drive or variable speed drive, it has to be provided with it to fulfil regulation 1253/2014/EU.

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Figure 6.10: Technical data AHU

Rosenberg Hungária Kft.
2532 Tokodaltáró,
József Attila út 34.

Pers. of Cont.:	horvath.peter	Date:	2023. 05. 16.	Order	-
Project No:	Unideb	Pos-No:	AHU-01	Customer:	
Project:	Unideb			RZ-No:	

Supply:	Airbox S60-2825	Height:	2770mm	Width:	5740mm	Weight:	10550kg
Exhaust:	Airbox S60-2825	AHU-S602825WW	Length:	9810mm	S60 - 60mm Insulation		
		Weather Proof Design					

Location	Name	ArtNo	Qt.	Wt. [kg]
Sup Z1	Housing Airbox S60-2825 l=810mm		1	249
Sup Z1 C1	air damper steel galv. (EN 1751 Kl.2) size 1 entire cross section; 2 piece; W=2548; H=1158; flange=30	JK2IC2825N1	1	185
Sup Z1 C1	Information: Attention, 2 damper-motors are required	info-SMB	1	0
Sup Z1 C1	condensate tray bottom 480 mm size 28 AlMg3 Ø32 mm	KWB00X280480A	1	0
Sup Z1 C2	bag filter short ISO ePM10 50% (M5), frame AISI304 (V2A)	FTN2825-1050EK2	1	84
Sup Z1 C2	2 press. measurement nozzle incl.cover (PVC),D=6.5mm,ass.	MSS000-0001B	1	0
Sup Z1 C2	Lighting (Hygenic, IP65, LED 8W)	LEU220-0008B	1	1
Sup Z1 C2	Inspection window	SGLX060D160N	1	0
Sup Z1 C3	flex connector entire cross section B=2718; H=2418; Flansch=30	ELSC2825001N	1	5
Sup W1	Housing Airbox S60-2825 l=3210mm		1	1155
Sup W1C1	PWT-H2A1500-2825-N	PWT28251500N	1	2086
Sup W1C1	2 trays for condensate and cleaning works	KWA000-0000P	1	0
Sup W1C1	Bypass damper	mpr-JKB_H2_PWT	1	0
Sup W1C1	2 press. measurement nozzle incl.cover (PVC),D=6.5mm,ass.	MSS000-0001B	2	0
Sup Z2	Housing Airbox S60-2825 l=1245mm		1	566
Sup Z2 C1	weatherproof Circulating box, side by side	BOX28R-2WU7N	1	240
Sup Z2 C1	Air damper weather proof, 2550x1095	JKL28R-2003W	1	80
Sup Z2 C1	grating for floor openings, galvanized, 1100 mm x 1000 mm	SCHWPR-1100N	3	33
Sup Z3	Housing Airbox S60-2825 l=1365mm		1	369
Sup Z3 C1	Heater PWW Cu/Al Frame FeZn	PWW282501A0NTP	1	101
Sup Z3 C1	Frost protection thermostat	FST000-021LN	1	0
Sup Z3 C2	Empty Casing		1	0
Sup Z3 C3	Cooler PKW Cu/Al Frame AISI 304/1.4301	PKW282504A0ETP	1	296
Sup Z3 C3	cond. tray AISI304, 2.740 x 680, drainage sidew. D=40mm	KWA00X280680E	1	60
Sup Z3 C3	Drop separator PP/AlMg3 for elevated drain	TAS2825-101N	1	58
Sup Z3 C4	Heater PWW Cu/Al Frame AISI 304/1.4301	PWW282501A0ETP	1	93
Sup Z4	Housing Airbox S60-2825 l=2370mm		1	587
Sup Z4 C1	Fan free-run, stand. Motor RLM E6-9010-63-33-A	RLME6B20S3-100	1	587
Sup Z4 C1	IEC Motor 200L / IE3-6;18.5kW		1	0
Sup Z4 C1	repair switch BK/GY (GS13) 1 speed 30KW	H80-00043swgr	1	2
Sup Z4 C1	executing circular lead	mpr-Rmlaus10	1	0
Sup Z4 C1	Lighting (Hygenic, IP65, LED 8W)	LEU220-0008B	1	1
Sup Z4 C1	Inspection window	SGLX060D160N	1	0
Sup Z5	Housing Airbox S60-2825 l=810mm		1	249
Sup Z5 C1	bag filter HE ISO ePM1 60% (F7)	FTN2825-0160NHE1	1	49
Sup Z5 C1	2 press. measurement nozzle incl.cover (PVC),D=6.5mm,ass.	MSS000-0001B	1	0
Sup Z5 C1	Lighting (Hygenic, IP65, LED 8W)	LEU220-0008B	1	1
Sup Z5 C1	Inspection window	SGLX060D160N	1	0
Sup Z5 C2	flex connector entire cross section B=2718; H=2418; Flansch=30	ELSC2825001N	1	5
Exh A1	Housing Airbox S60-2825 l=570mm		1	197

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Figure 6.11: Technical data AHU

Rosenberg Hungária Kft.
2532 Tokodaltáró,
József Attila út 34.

rosenberg THE AIR MOVEMENT GROUP

Pers. of Cont.:	horvath.peter	Date:	2023. 05. 16.	Order	-
Project No:	Unideb	Pos-No:	AHU-01	Customer:	
Project:	Unideb			RZ-No:	

Supply:	Airbox S60-2825	Height:	2770mm	Width:	5740mm	Weight:	10550kg
Exhaust:	Airbox S60-2825	AHU-S602825WW	Length:	9810mm	S60 - 60mm Insulation		
		Weather Proof Design					

Location	Name	ArtNo	Qt.	Wt. [kg]
Exh A1 C1	flex connector entire cross section B=2718; H=2418; Flansch=30	ELSC2825001N	1	5
Exh A1 C2	bag filter short ISO ePM10 50% (M5)	FTN2825-1050NK2	1	83
Exh A1 C2	2 press. measurement nozzle incl.cover (PVC),D=6.5mm,ass.	MSS000-0001B	1	0
Exh A1 C2	Lighting (Hygenic, IP65, LED 8W)	LEU220-0008B	1	1
Exh A1 C2	Inspection window	SGLX060D160N	1	0
Exh A2	Housing Airbox S60-2825 l=2370mm		1	587
Exh A2 C1	Fan free-run, stand. Motor RLM E6-9010-63-33-A	RLME6B20S3-100	1	587
Exh A2 C1	IEC Motor 200L / IE3-6;18.5kW		1	0
Exh A2 C1	repair switch BK/GY (GS13) 1 speed 30KW	H80-00043swgr	1	2
Exh A2 C1	executing circular lead	mpr-Rmlaus10	1	0
Exh A2 C1	Lighting (Hygenic, IP65, LED 8W)	LEU220-0008B	1	1
Exh A2 C1	Inspection window	SGLX060D160N	1	0
Exh A3	Housing Airbox S60-2825 l=330mm		1	144
Exh A3 C1	flex connector entire cross section B=2718; H=2418; Flansch=30	ELSC2825001N	1	5
Exh A3 C2	air damper steel galv. (EN 1751 Kl.2) size 1 entire cross section; 2 piece; W=2548; H=1158; flange=30	JK2IC2825N1	1	185
Exh A3 C2	Information: Attention, 2 damper-motors are required	info-SMB	1	0
Exh A3 C2	condensate tray bottom 480 mm size 28 AlMg3 Ø32 mm	KWB00X280480A	1	0
AHU	frame profil powder-coated 60 µm spezial color (17535mm)	GVRC2825RP1XX	1	0
AHU	exterior panels coil coated RAL 7035 (17535mm)	GVAC2825NB110	1	0
AHU	Inside cover and side walls galvanized steel (17535mm)	GVIC2825GN110	1	0
AHU	Inside bottom galvanized steel (17535mm)	GVIC2825BN110	1	0
AHU	Floor sealing works per meter length (17535mm)	mpr-28XXHYGd	1	18
AHU	Rain cover trapezoid color coated RAL-7035 (17535mm)	WSDC28250000TR	1	596
AHU	Rain cover trapezoid mandatory from roof width 2770mm !	WSDX25QFLACH	1	0
AHU	Module joint inside cover, 4-sided, circumferential	VBMXXXX	9	18
AHU	Wire mesh unit exit RAL7035, stiffener RAL7035	SGAC282500BB	1	22
AHU	Base frame assembly to module; Add. cost per module	mpr-GRBan	10	0
AHU	door stay hinge	TFSD-000001N	9	0
AHU	Maintenance doors >435 mm with 3D-hinges & central levers	mpr-Tür60zen	1	0
AHU	Base frame 200mm, galvanized steel, loose (17535mm)	GRBXX280000DNN	1	947
AHU	4 Lifting eyes for welded base frame, 1 set per unit	KRAXB010000N	1	8
AHU	Syphon suction side / UV-resistant	SIS000-0003N	1	1
AHU	Syphon pressure side / drain trap / UV-resistant	SID000-0003N	1	1

Delivery Condition
Payment Conditions
Delivery Time

General Information
The weather protection roofs attached to the field devices are made of AlMg3.

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Figure 6.12: Technical data AHU

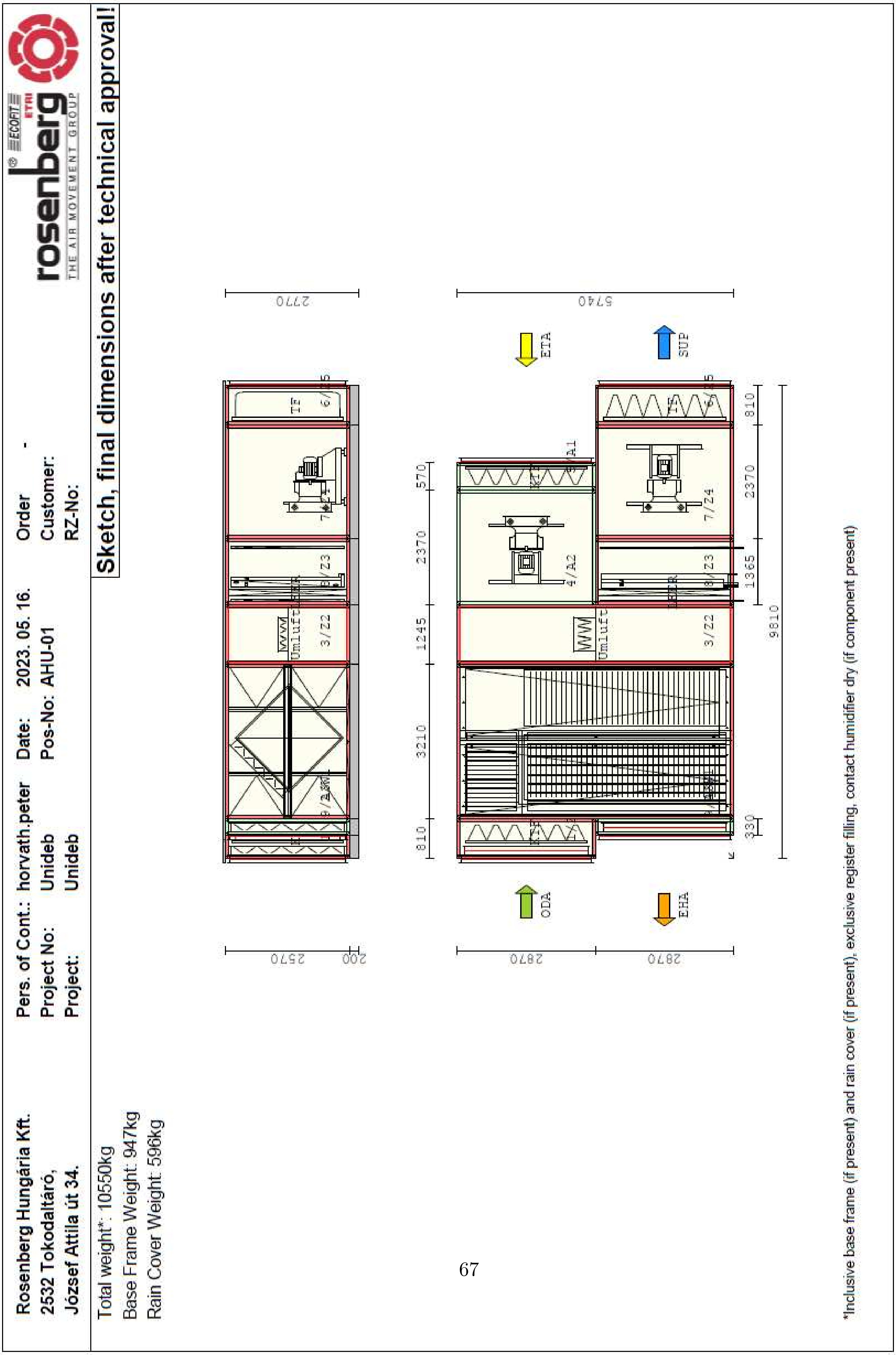

Figure 6.13: Technical data AHU

6.2 HEAT PUMPS

Firstly, it is important to highlight that this section could not be considered as part of the air handling unit, but on the other hand, it could not be inserted in any other chapter. This is explained by the fact that the heat pumps aren't properly part of the air handling unit that we are considering, but, however, they are the auxiliary system that feed the two coils, both for heating and cooling.
Therefore this small chapter gives a small overview in regard to the heat pumps and their functioning.
A heat pump uses technology similar to that found in a refrigerator or in an air conditioner. It extracts heat from a source energy(i.e., the surrounding air, geothermal energy stored in the ground, or nearby sources of water or waste heat from a factory) and then amplifies and transfers the heat to where it is needed.
Because most of the heat is transferred rather than generated, heat pumps are far more efficient than conventional heating technologies such as boilers or electric heaters and can be cheaper to run. The output of energy in the form of heat is normally several times greater than that required to power the heat pump, normally in the form of electricity. For example, the coefficient of performance (COP) for a typical household heat pump is around four, i.e. the energy output is four times greater than the electrical energy used to run it. This makes current models 3 to 5 times more energy efficient than gas boilers.
COP is defined as:

$$\mathrm{COP} = \frac{Q}{W} \tag{6.1}$$

Where:

- Q = thermal power realised from the condenser
- W = electric energy absorbed by the compressor

During the heating mode, (since the heat pumps works with a reversible cycle, they are able to work as a cooler too) the COP depends on two different factors, the water supply temperature and the external air. The newest heat pumps, are able to work in the worst scenario, i.e. supplying water at 55/60°C even with the external air at -15°C maintaining the COP higher than 1.5.
This further demonstrates the convenience of a heat pumps over the traditional methods.
During the summer period, the heat pumps can invert the cycle, and exactly like a refrigerator, are able to extract the heat contained into the cold source due to the functionality of the evaporator. As well as for the winter, there is a sort of efficiency also for the summer, which is the EER (energy efficiency ratio), which similar to COP is defined as:

$$\mathrm{EER} = \frac{Q}{W} \tag{6.2}$$

Where:

- Q = cooling power realised from the evaporator
- W = electric energy absorbed by the compressor

The figure 6.14 shows the cycle in the heating mode. Firstly, there is the **EVAPORATION**, the outside air is drawn in to the unit. It warms the refrigerant which turns to gas at very low temperature. Then we have the **COMPRESSION** where the gas is compressed, this operation raises the temperature and the pressure of the internal gas. The useful work (as a constant during the winter season) is given by the **CONDENSATION** process, where the gas

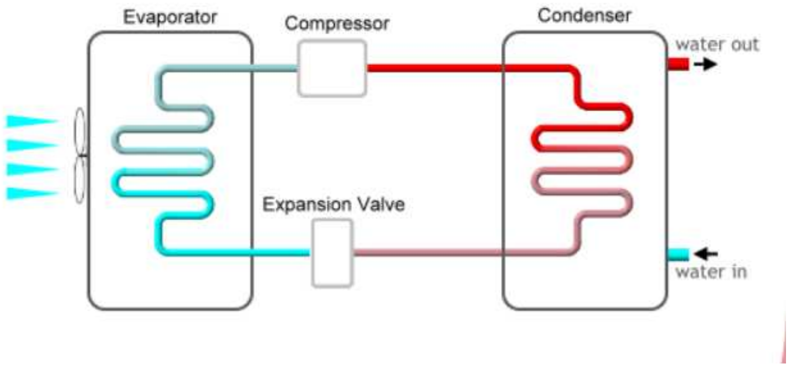

Figure 6.14: Heat pump winter working cycle

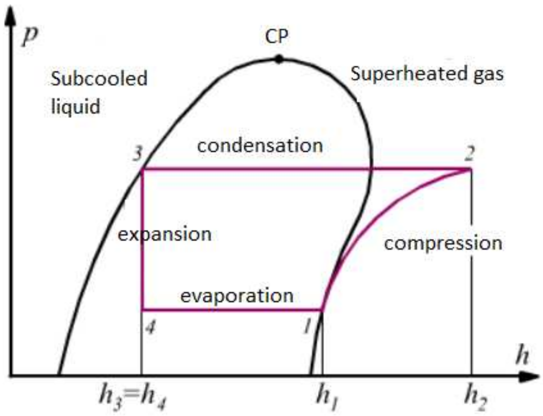

Figure 6.15: Heating process on p-h diagram

release heat to the system and condenses back to a liquid state. And finally the **EXPANSION**, the liquid passes through an expansion valve, reducing temperature and pressure. From here, the cycle start again 6.15.

- Sizing

In order to achieve a correct size of the heat pumps, a correct evaluation of the power required from the space that we are considering is necessary. That has been done with the dynamic simulation and presented in the chapter 6.
Therefore, what is remaining, is to chose the most fitting heat pump from the catalogue. I picked the JODO company, previously known as ATAG Italia, which is an established company with significant knowledge and experience in the heating, cooling and hybrid systems. I started from the maximum electrical power required, which is during the winter season and it is around 40 [kW], and I verified that the heat pump that I was considering, was able to supply the maximum heating and cooling energy.
JODO AIR 120HP MAX - a air to water heat pump has been chosen and it is represented in figure 6.16
In the table 6.3, I reported the most important data of the machine.

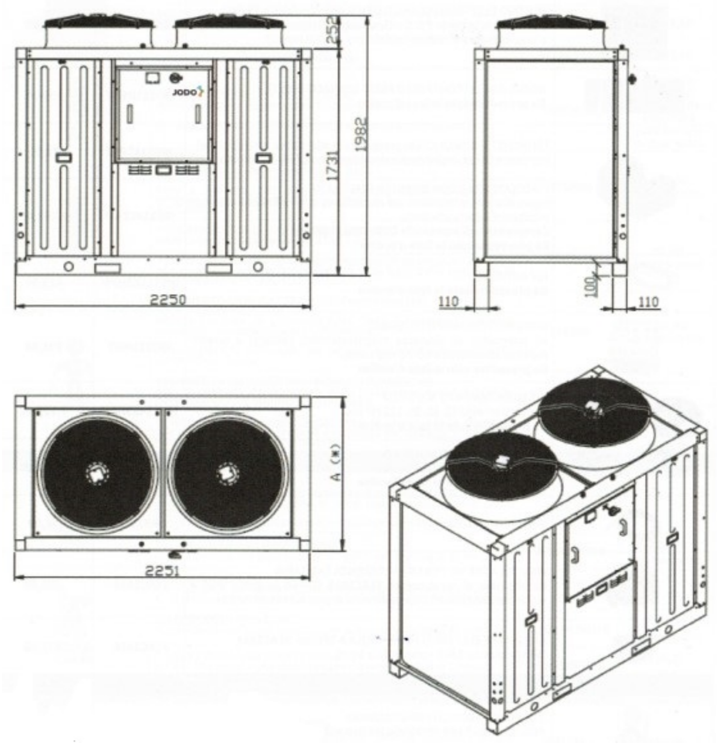

Figure 6.16: Heat pump drawing

External unit techical data		Unit	Model
Electrical data	Alimentation		**120 HP MAX**
	Max power absorbed	kW	55.8
	Max current absorbed	A	83.9
Cooling	Power delivered	kW	127.3
	Power absorbed	kW	34.9
	EER	W/W	3.65
Heating	Power delivered	kW	97.13
	Power absorbed	kW	30.35
	COP	W/W	3.2

Table 6.3: Technical data heat pump

CARATTERISTICHE TECNICHE UNITÀ ESTERNE		Unità di misura	Modello AIRP-HP MAX			Modello AIRP-HP MAX		
			75 HP MAX	85 HP MAX	95 HP MAX	115 HP MAX	120 HP MAX	135 HP MAX
Dati elettrici	Alimentazione		400V/3P+N+T/50Hz					
	Potenza massima assorbita	kW	39,9	42,3	46,7	52,3	55,8	63,0
	Corrente massima allo spunto	A	107,1	110,5	117,3	125,7	130,9	141,7
	Corrente massima assorbita	A	60,1	63,5	70,3	78,7	83,9	94,7
Raffreddamento	Potenza frigorifera (1)	kW	79,6	90,2	102,8	113,3	127,3	139,3
	Potenza assorbita (1)	kW	21,8	24,6	28,2	31,0	34,9	38,2
	EER (1)	W/W	3,65	3,66	3,65	3,65	3,65	3,65
	SEER (1)	W/W	4,73	4,72	5,31	4,55	4,72	4,74
	Potenza frigorifera (2)	kW	65,6	74,6	83,9	94,7	105,6	114,3
	Potenza assorbita (2)	kW	22,6	25,7	28,8	32,7	36,2	39,4
	EER. (2)	W/W	2,90	2,90	2,91	2,90	2,92	2,90
	SEER (2)	W/W	3,98	4,09	4,18	3,93	4,05	4,09
Riscaldamento	Potenza termica (3)	kW	68,40	74,7	85,6	93,34	102,47	111,47
	Potenza assorbita (3)	kW	16,85	18,44	21,14	23,87	25,3	28,58
	COP (3)	W/W	4,06	4,05	4,05	3,91	4,05	3,90
	Potenza termica (4)	kW	65,86	71,0	82,12	88,57	97,13	108,28
	Potenza assorbita (4)	kW	20,52	22,19	25,66	27,68	30,35	36,09
	COP (4)	W/W	3,21	3,20	3,20	3,20	3,20	3,00
	SCOP (5)	W/W	3,58	3,55	3,53	3,54	3,58	3,50
	classe energetica (5)		A+	A+	A+	A+	A+	A+
	SCOP (6)	W/W	2,85	2,82	2,83	2,83	2,85	2,82
	classe energetica (6)		A+	A+	A+	A+	A+	A+
Compressore	Tipo Quantità		Scroll					
			2 DC inverter + 2 on off			2 DC inverter + 4 on off		
	Numero circuiti frigoriferi		2					
	Olio (tipo, quantità per circuito)	mL	FVC68D, 8000			FVC68D, 11400		
Motore ventilatore	Tipo		EC					
	Numero		2					
	Potenza nominale (2)	kW	2,4	2,7	3,0	3,4	3,8	4,1
	Potenza massima assorbita	kW	3,6			4,2		
	Corrente massima assorbita	A	5,7			6,6		
	Portata d'aria nominale	m3/s	6,5 x2	7 x2	7,5 x2	8 x2	8,5 x2	9 x2
Refrigerante	Tipo		R410A					
	Quantità refrigerante per circuito (9)	kg	10,2	10,4	13,2	13,4	14,2	14,3
	Quantità di CO2 equivalente (9)	ton	21,30	21,72	27,56	27,98	29,44	29,86
	Pressione di progetto (alta/bassa)	MPa	4,15/2,7		4,15/2,7		4,15/2,7	
Circuito idraulico	Portata acqua (2)	L/s	3,13	3,57	4,01	4,52	5,05	5,47
	Perdita di carico interna (2)	kPa	32	36	37	34	33	38
	Attacchi idraulici	inch	2" ½ F		2" ½ F		2" ½ F	
	Minimo volume acqua	L	200			260		
Caratteristiche del circuito idraulico con accessorio Pompa AC integrata	Prevalenza utile (2)	kPa	83	79	78	81	82	77
	Potenza nominale pompa AC (2)	kW	1	1	1	1,2	1,2	1,2
	Potenza massima pompa AC	kW	1,10			1,32		
	Corrente massima assorbita pompa AC	A	1,96			2,35		
Rumorosità	Potenza sonora (7)	dB(A)	79	79,5	80	81	83	84
	Potenza sonora SL / SSL	dB(A)	77,5 / 76,7	78 / 77,2	78,5 / 77,7	79 / 78,7	81 / 80,2	82 / 81,2
	Pressione sonora 1 m/ 10m (8)	dB(A)	61,4 / 47,2	61,9 / 47,7	62 / 48,2	63,4 / 49,2	65 / 51,2	66,4 / 52,2
Dimensioni e pesi	Dimensioni (P×A×L)	mm				1450x2010x2250		
	Dimensioni (P×A×L) con kit SSL	mm	1170x2180x2250			1450x2270x2250		
	Dimensioni max imballo(P×A×L)	mm	1200x2150x2250			1480x2170x2250		
	Dimensioni imballo(P×A×L) kit SSL	mm	1200x2340x2250			1480x2430x2250		
	Peso in esercizio	kg	923	946	996	1011	1105	1120
	Peso netto/lordo	kg	903/943	915/55	971/1011	986/1026	1078/1128	1092/1142

Figure 6.17: Technical data

Chapter 7

AHU energy

Once understood how the air handling unit works, we have to calculate exactly the number of hours of functioning, and the energy that we are going to use in order to give an estimation and have an idea about which will be the final and consequently the primary energy consumption of the system.
Bearing in mind that the system that we're considering, has several aims, provide fresh air, extract the exhaust one, and provide heating and cooling, more or less, the machine is always running, except during the nights. In order to keep warm the office, the heating is switched on at 6 am until the 22 pm. this means that I have assumed to run the system even during some hours when no one is present, in order to exploit the thermal inertia of the building during the night hours and to have comfort conditions even during the very early hours.
The entire calculation, has been set up on an excel file, which will be attached with the thesis itself. But as first, I obtained from the UniDebrecen database, the weather data hour by hour of temperature and relative humidity of three different years, 2019-2020-2021. I choose the 2019, which was the one with the worst scenario, i.e. where I met the lowest temperature during winter and the higher during summer, which are respectively -12.65°C and 34.24°C. So this was the beginning of the core of the system, temperature and relative humidity hour by hour for a total amount of 8760 hours. In addition, I needed other parameters that are reported in the table 7.1.

Trasmission factor	**x**	1160	[W/K]
Max n of people	**n**	319	-
Winter heat ex. efficiency	η_{winter}	0.882	-
Summer heat ex. efficiency	η_{summer}	0.777	-
Volumetric flow rate	**Vtot**	40855	[m3/h]
Mass flow rate	**mtot**	13.6	[kg/s]
Minimal air fresh rate	**mmin,fresh**	2000	[m3/h]
Internal gain people	**People**	80	[W/person]
Internal gains computer	**Computers**	80	[W/each element]

Table 7.1: Initial data

I'll start explaining which are the parameters used at the beginning.

- **Transmission factor**

From the chapter 3, I obtained the thermal transmittance from the analysis in the WinWatt program, which as already explained has given a value of 0.175 $\left[\frac{W}{m^3K}\right]$ which has been multiplied

for the $6629m^3$ of the office. Obtaining a factor of $1160\left[\frac{W}{K}\right]$. This factor will be used to calculate the transmission losses through the envelope later.
Instead, items like **n° of people**, **volumetric and mass flow rate** has been already defined when I was looking for the flow rate estimation, and they represent the starting data to define the fresh air that we should put inside the office, and consequently the recirculation air flow rate which is constantly pushed inside.
As said, those are beginning data, because has been already calculated into the chapter 5. But in particular, for the number of people, I tried to imagine a real situation, thinking that having all of the 319 people inside the office at 8 or 9 a.m. it's unreal, so I tried to stagger the admissions, taking lunch breaks into account and never reaching the maximum number of admissions allowed, but I kept 80% for most of the working time. The entrances were restricted according to the following figure 7.1, considering no presence during the night hours and were spread out over all days of the year.

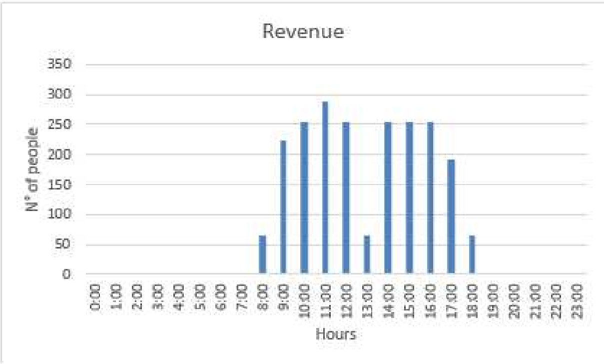

Figure 7.1: People present in the office all over the working day

It should be noted, that the entry and presence of people inside the office at various times is purely a hypothesis, which tries to reflect a real case history.

- **Internal gains**

Another important parameter, which contribute in heating and cooling, is the internal heat gain. Should seems low, but instead can help a lot during the winter period, and becomes a huge part of the cooling load during summers.
It's composed by the heat gain due to the people, and due to the electric appliances. For the ones due to the **people** [1], as already described in the chapter 1.3, for the internal load of the people, I pick the normal value during office activities, according to the standards [6] and from web site research like - `https://greenreport.it/news/economia-ecologica/quanta-energia-produce-un-corpo-umano/`
As electric appliances, I considered just the **computers**, because I think they should be the more predictable, and also they follow the curve of the people inside the office because are directly proportional to them.
So, I decided to consider a value of 80 [W/person or element] The value that I assigned, it's close to a value that I found in the Hungarian standards which report an average value around 8/15 W/m^2. Doing the multiplication, using an average value of 10 W/m^2. I obtained a value of 18940 W, instead using 80 W/person I obtained 17840W, quite close to the previous one.

[1]In the previous lines, I mentioned the number of persons, in this case, we talk about the heat load generated by them

- **Efficiency**

Last but not least, we also have, as starting data, the efficiency for heating and cooling of the heat exchanger, which has the aim to pre-heat or pre cool down the external temperature, in order to have a lower ΔT in the coils; η_{winter} and and η_{summer}. This two parameters, has been found, in the previous chapter 6, with the worst conditions and represent the starting point to find out which will be the temperature after the heat exchanger.
It is also fundamental, for a correct estimation of the net energy, the emission efficiency η_{em}, which represent the thermal losses that goes from the heat pumps direct to heater and the cooler and comes from the following equation:

$$\eta_{em} = \eta_{str} * \eta_{ctr} * \eta_{emb} = 0.97 * 0.95 * 1 = 0.92 \tag{7.1}$$

which stands respectively for:

- η_{em} = emission efficiency
- η_{str} = stratification
- η_{ctr} = control system
- η_{emb} = embedded losses

Stratification of the heat inside the room, since we're using an AHU should be quite low, due to a correct design of the air diffusers, for this reason, the stratification doesn't occur and consequently the efficiency is really high. Same logic for the control system, and the embedded losses which could be considered negligible.

7.1 Dynamic simulation

7.2 Base case - Constant air volume CAV - 55°C Water heater coil

After having introduced all the beginning data, let's see, point by point which is the procedure which lead to figure it out the energy consumption along the year.

- **Common parameters, both for heating and cooling**

First of all, from the meteorological station situated in Debrecen, I reported the meteorological data hour by hour, day by day for the entire year, obtaining a database of 8760 cells in excel, which was standing for **data, hour, average temperature** and **average relative humidity**.
After that, following the Hungarian standard, I set the internal temperature for heating and for cooling, which are respectively 21°C and 26°C.
Focus on the heating mode, I switched off the heating from the 15th of May, to the 15th of September, in order to avoid unnecessary heating and also for those hours whenever the external temperature was higher than 20°C; I left the cooling free, because, as I'll show later, the simulation report some hours, even during winter, where the sum of the internal load, is higher than the losses through the envelope, so I used the free-cooling where it was possible, but the normal cooling also.
After that, I allocated the number of people and the internal gains as explained, which are a huge part of heat that has to be managed, and bearing in mind the air class explained in the chapter 5, considering to inject 10 kg/s for each person present in the room, I found the minimal flow rate hour by hour in kg/s and in m^3/h with the simple equations:

$$\text{minimal volumetric flow rate } [m^3/s] = \text{n° of people} * 10\text{l/s} * \frac{3600s}{1000} \tag{7.2}$$

finding the minimal flow rate in m^3/h. With the following equation I found out kg/s:

$$\text{minimal flow rate}[kg/s] = \frac{\text{m.f.r.}[m3/h] * \rho}{3600} \tag{7.3}$$

Consequently, by a simple subtraction, knowing the minimal flow rate and the total one, I obtained the recirculation value, in [kg/s] and in [m^3/h] either.
Those, were parameter useful for the calculation of the thermal energy that should be injected or extracted from the building. Based on the season, this load could be positive or negative. It's easy to understand looking for example to the load generated by the people. Bearing in mind that a person in every activity (even sleeping), release a certain amount of heat, it's clear that during winter, it become a resource because it's going to help the heating of the building, and in some cases it could be an issue if the heat released becomes higher than the heat losses, in other hand, during summer season, the internal load generated by people, it's clearly a problem, just heat that has to be removed from the room.
With this in mind, the next step is to define the energy required to compensate the losses, i.e. the energy our building requires to maintain a constant temperature in winter and summer.
Let's see first the loads which contribute in both seasons.

$$q_{\text{heating/cooling}} = q_{people} + q_{\text{internal gains}} + q_{transmission} + q_{\text{solar heat gains}} \tag{7.4}$$

Let's see the items one by one.

GR	Athens	38' N	23' 43" E	2.00	2.52	3.67	5.21	6.38	7.52	7.61	6.91	5.57	3.50	2.16	1.63	4.56
HU	Budapest	47' 30" N	19' 3" E	1.00	1.71	2.76	3.90	5.03	5.30	5.62	4.84	3.57	2.24	1.17	0.88	3.17
IE	Dublin	53' 20" N	6' 15" W	0.56	1.07	1.97	3.32	4.40	4.30	4.30	3.40	2.69	1.43	0.77	0.43	2.39

Figure 7.2: Average solar radiation in Budapest

- **People load and electric appliances**

Once defined the number of people along the working hours, it's sufficient multiplying them for the heat generated from a person in sit position doing desk work, which is $80[W/eachperson]$. The highest peak of power reached by the people load is around 23kW, a considerable number which surely help the heating, but at the same time, produce a high value of exhaust air that must be extracted.
It's almost the same for the electric appliances. From a lecture of the university of Federico II in Naples - `https://www.docenti.unina.it/webdocenti-be/allegati/materiale-didattico/667663` I found out an average value for the computer between 8 and 15 W/m^2, instead, from Hungarian standards. has been possible to find out a value of $80W$/each element. Doing the calculation, we're more or less on the same values, so I used, also for the pc, a value of $80W$/each element, which multiplied for the numbers of computer, has given the thermal load generated by the computers.
I emphasise once again that these two parameters will play in favour in the winter season, and against in the summer season.

- **Transmission load**

This value, represent the constant losses through the envelope, which must be compensate with heating energy. It's a very important parameter that came out with the preliminary analysis of the building done with WinWatt, which goes to consider all the technical data of the building, like stratigraphy of the walls, ceiling and floors; the quality of the windows and the frame. External environment, like sun, wind and temperatures. The exposition of the external wall and the possibilities or the presence of shading.
So the peak power, required hour by hour in order to compensate the losses, is calculated as follow:

$$\text{transmission load} = \frac{x * (t_{internal} - t_{external})}{1000} = [kW] \tag{7.5}$$

The temperature difference was previously calculated when I set the internal temperature for heating and cooling at the beginning of the calculations. It's clear that in this way, the transmission losses, will change the sign based on the season, which I then had to pay attention to when calculating the net energy in heating and cooling.

- **Solar heat gain**

This is a particular item, because it's kind difficult to give an accurate values close to reality. There are several way to calculate it, but I'm gonna show the method that I built with the Hungarian professor Kostiàk Attila.
First of all, as shown in the figure 7.2, we started from the average solar radiation in Budapest of the last 10 years.
We then found a corrective value by normalising the average values like show in the table 7.2, dividing every average value of each month for the highest one which is in July.
Then, from WInWatt software, we found out which is the average solar heat gain for our Forest office for each hours of the day during the hottest month, July, like showed in the figure 7.3.

	Months	Solar correction
January	1	0.178
February	1.71	0.304
March	2.76	0.491
April	3.9	0.694
May	5.03	0.895
June	5.3	0.943
July	5.62	1
August	4.84	0.861
September	3.57	0.635
October	2.24	0.399
November	1.17	0.208
December	0.88	0.157
Annual average	3.17	0.564

Table 7.2: Normalized solar radiation

Average solar heat gain	
Hours	**kW**
0:00	0
01:00	0
02:00	0
03:00	0
04:00	0
05:00	12.9
06:00	30.9
07:00	45.4
08:00	59
09:00	66.5
10:00	70.7
11:00	70.1
12:00	72.1
13:00	78.5
14:00	86.9
15:00	92.4
16:00	95.5
17:00	97.5
18:00	90.7
19:00	76.8
20:00	51.7
21:00	33.5
22:00	26.9
23:00	0

Table 7.3: Internal heat gain hourly

Jan.	**Feb.**	**March**	**April**	**May**	**June**	**July**	**Aug.**	**Sept.**	**Oct.**	**Nov.**	**Dec.**
0	0	0	0	0	0	0	0	0	0	0	0
0	0	0	0	0	0	0	0	0	0	0	0
0	0	0	0	0	0	0	0	0	0	0	0
0	0	0	0	0	0	0	0	0	0	0	0
0	0	0	0	0	0	0	0	0	0	0	0
0	0	**6.34**	**8.95**	**11.55**	**12.17**	**12.9**	**11.11**	0	0	0	0
5.50	**9.40**	**15.18**	**21.44**	**27.66**	**29.14**	**30.90**	**26.61**	**19.63**	**12.32**	0	0
8.08	**13.81**	**22.30**	**31.51**	**40.63**	**42.81**	**45.4**	**39.10**	**28.84**	**18.10**	**9.45**	0
10.50	**17.95**	**28.98**	**40.94**	**50.81**	**55.64**	**59**	**50.81**	**37.48**	**23.52**	**12.28**	**9.24**
11.83	**20.23**	**32.66**	**46.15**	**59.52**	**62.71**	**66.5**	**57.27**	**42.24**	**26.51**	**13.84**	**10.41**
12.58	**21.51**	**34.72**	**49.06**	**63.28**	**66.67**	**70.7**	**60.89**	**44.91**	**28.18**	**14.72**	**11.07**
12.47	**21.33**	**34.43**	**48.65**	**62.74**	**66.11**	**70.10**	**60.37**	**44.53**	**27.94**	**14.59**	**10.98**
12.83	**21.94**	**35.41**	**50.03**	**64.53**	**67.99**	**72.1**	**62.09**	**45.80**	**28.74**	**15.01**	**11.29**
13.97	**23.89**	**38.55**	**54.48**	**70.26**	**74.03**	**78.5**	**67.60**	**49.87**	**31.29**	**16.34**	**12.29**
15.46	**26.44**	**42.68**	**60.30**	**77.78**	**81.95**	**86.90**	**74.84**	**55.20**	**34.64**	**18.09**	**13.61**
16.44	**28.11**	**45.38**	**64.12**	**82.70**	**87.14**	**92.40**	**79.58**	**58.70**	**36.83**	**19.24**	**14.47**
16.99	**29.06**	**46.90**	**66.27**	**85.47**	**90.06**	**95.5**	**82.25**	**60.66**	**38.06**	**19.88**	**14.95**
0	**29.67**	**47.88**	**67.66**	**87.26**	**91.95**	**97.5**	**83.97**	**61.94**	**38.86**	**20.30**	0
0	0	**44.54**	**62.94**	**81.18**	**85.54**	**90.7**	**78.11**	**57.62**	**36.15**	**18.88**	0
0	0	0	**53.30**	**68.74**	**72.43**	**76.8**	**66.14**	**48.79**	**30.61**	0	0
0	0	0	0	**46.27**	**48.76**	**51.7**	**44.52**	**32.84**	0	0	0
0	0	0	0	**29.98**	**31.59**	**33.5**	**28.85**	0	0	0	0
0	0	0	0	0	**25.37**	**26.9**	0	0	0	0	0
0	0	0	0	0	0	0	0	0	0	0	0

Table 7.4: Monthly solar heat gain

Therefore, taking it as a reference, I multiplied the scale factor of each month found previously, for the July's values, obtaining the average values for the whole year month by month.
And to go and use this data on the spreadsheet, I entered the average values of the days repeating them over the whole month, so for all 12 months of the year.
Since the data collected was based on completely clear weather, on the spreadsheet, I then reduced it by 50% to avoid overestimates.
At this point, in order to make the explanation clearer, it's convenient to separate the discussion into winter and summer design.
Let's start with winter.

7.2.1 WINTER

At this point, in order to proceed with the calculation of the net energy for heating, I calculated two different temperatures:

- t_{f2} = it's the temperature of the external air after have been passed through the heat exchanger, so it's the temperature of the fresh air that has been pre-heated.
  It fluctuates between a minimum value of 17°C and a maximum value of 21°C.

It's calculated with the following equation:

$$t_{f2} = t_{external} + \eta_{winter} * (t_{internal} - t_{external}) \tag{7.6}$$

which comes from the efficiency equation of the heat exchanger:

$$\eta_{winter} = \frac{t_{f2} - t_{f1}}{t_{e1} - t_{f1}} \tag{7.7}$$

The second temperature comes from the balancing equation:

$$\dot{m}_{tot} * t_{mix} = \dot{m}m_{fresh} * t_{f2} + \dot{m}_{ric} * t_{ex1} \tag{7.8}$$

From which we derive:

- t_{mix} = this is the temperature of the air flow resulting from the sum of the fresh air flow and the recirculation air flow, and is the temperature at which the air flow enters the preheating coil.

$$t_{mix} = \frac{\dot{m}_{fresh} * t_{f2} + \dot{m}_{ric} * t_{ex1}}{\dot{m}_{tot}} \tag{7.9}$$

From here, it is possible to calculate first, the energy for heat up the amount of fresh air from the temperature reached after the heat exchanger to the final one, and then sum up all the items to find out which is the net energy required.
T_{mix} is also, the maximum temperature that the system can reach without external energy. So:

$$Heating_{coil} = \dot{m}_{fresh} * cp * (t_{internal} - t_{f2}) \tag{7.10}$$

The above calculation has been done just whenever it's required a fresh air flow rate, hence just during the working hours, between the 15th of September to 15th of May and when the external temperature it's below 21°C.
And finally, to figure it out which is the net energy required, summing up all the terms in the equation 7.4, adding the energy calculated in the equation 7.10.
We can calculate the final heating energy which is:

$$\text{Net energy demand} = \text{Transm. load} + \text{Heating coil - people load - heat gains - solar gain} \tag{7.11}$$

Also, in the final heating energy calculation, there is an additional item, which could be considered external to the equation 7.11. Is the post heating energy, that is used to heat up the mass flow rate when the freecooling is used and whenever the supply temperature result below 18°C. But this is rarely required, therefore I preferred to leave out this item from the equation, and analyze after if it required or not.
It's interesting to notice that during the working hours, I obtained a negative value, which means that during that hours, contrary to logic, I have to cool down (even if we're in winter), and to remove the excess heat given by the extra heating loads. This phenomenon is due to different parameters, first of all, we are dealing with a new building, which has even the golden Leed energy certification, which ensure very high performance and consequently very low losses through the envelope. Also, during the most intense working hours it happen that the heat generated it's already enough, and even becomes too much due to the presence of people, the solar and the heat gain. This issue is solved using the freecoling.

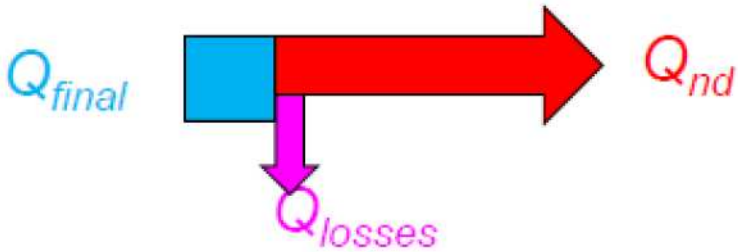

Figure 7.3: Path of final energy

The net energy demand Q_{nd} in the figure 7.3 ,is the amount of energy that the air flux should bring inside the ambient that we're considering. The next step consist to find out which is the final energy required from the matched system, which are the heat pumps. As shown in the figure 7.3 the final energy is bigger then the net energy demand, because take into account the efficiency of the heat exchange through the system and all the losses. The parameter which consider that, is the η_{em} explained in the equation 7.1. In the figure 7.4 it's represented the heater and its regulation. In the case that we're considering, the inlet and outlet temperatures of the heater has been fixed in the design of the machine, which are respectively 55°C and 40°C giving a ΔT of 15°C. For this reason what can be changed on the heat pumps side in order to have variable power is the flow rate inside the coil. It's consequently that a two way valve has been used.
So, the last step, has been to calculate the heating energy consumption that has to be delivered from the heat pumps, and through the COP to determine the electric energy required.
From the datasheet I retrieved the COP curve of the machine through interpolation in the figure 7.5, with a supply water temperature of 55°C, and I assigned the COP determined for every temperature at every hour of the year.
From the COP, with the equation 7.12, I found out which is the electric consumption for the heating.

$$P_{e,heating} = \frac{Q_{heating}}{\eta_{em} * COP} \tag{7.12}$$

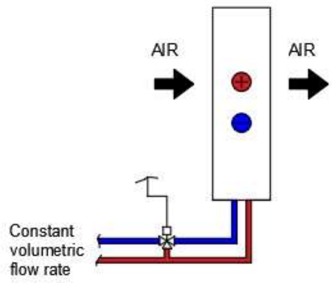

Figure 7.4: Heating battery

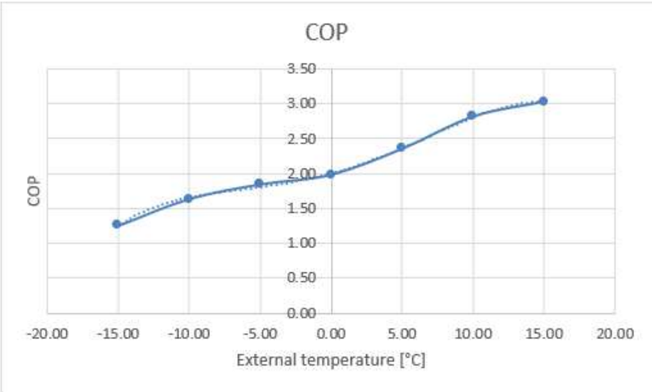

Figure 7.5: COP heating battery 55°C

Last step of the heating mode, and for the base case, was the calculation of the auxiliary systems, the fans.
Still on the datasheet, on the fan item, I took the value of $990[Pa]$.
I calculated the electric energy for the fans with the following equation:

$$P_{el} = \frac{\text{Volumetric flow rate} * \Delta P}{\eta 1 * \eta 2 * \rho} \tag{7.13}$$

Where:

- ΔP = pressure drop with the total volumetric flow rate
- $\eta 1$ = efficiency of the fan
- $\eta 2$ = efficiency of the electric motor
- ρ = Air density
- Volumetric flow rate = has to be in [kg/s]

And finally divided per 1000 in order to have kW and times 2 in order to consider both side, supply and exhaust.

.

$\Delta P[Pa]$	$\eta 1$	$\eta 2$	$\rho[kg/m3]$
990	0.8	0.89	1.005

Table 7.5: Data of the fans

7.2.2 SUMMER

For summer, the logic it's the same, so in order to arrive and find out the net energy cooling demand and the final energy, I found the two temperatures already described, which are t_{f2} and t_{mix}. The equation to find out the t_{mix} remain the same, while for the t_{f2} the differences are that, I used η_{summer}, and there is a difference also into the formulation of the equation, which comes from the equation of the efficiency of a heat exchanger:

$$\eta_{summer} = \frac{t_{f1} - t_{f2}}{t_{f1} - t_{ex1}} \tag{7.14}$$

and the equation that result is:

$$t_{f2} = t_{f1} - \eta_{summer} * (t_{f1} - t_{e1}) \tag{7.15}$$

All the terms present in these two equation 7.14 and 7.7, are easily seen in the figure 4.3 or 4.2. As for the winter, I calculated the cooling energy, with the same equation 7.10 only with the parameters in the right place. The cooling, it has been set in order to switch on during every period of the year when the external temperature goes over 25°C, for a total of 669h.
Moreover, it is important to highlights, that I don't take into account the dehumidification process, but I consider the energy required, going to multiplied the net energy required for cooling per 1.3, as a general rule.
Also in the summer case of course, we have the final energy for cooling provided by the heat pumps, and consequently the electric energy. The cooling energy of the heat pumps, has been found using the EER of the system as showed in the figure 7.6

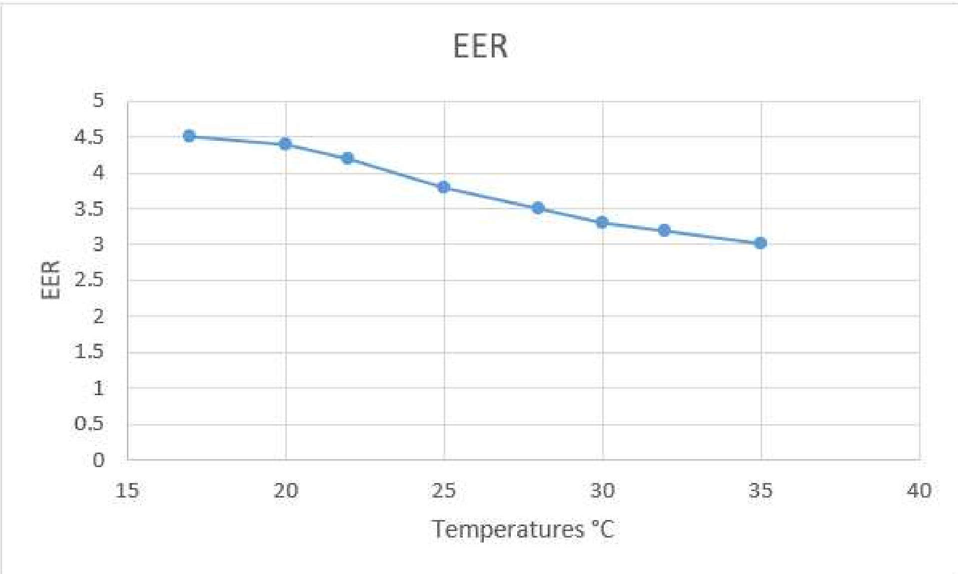

Figure 7.6: Energy efficiency ratio

As per calculating the electricity consumption of fans, the equations remains unvaried, therefore, I used the same data for the winter case.

7.3 Results base case

Regardless of the machine configuration that I'm going choose, what remains unchanged is the net energy required by the office building, in light on that, I'm going show the results of the energy required which are going to be the base for every case study.
Unless otherwise specified, all results refer to the whole year
Once again I'm going start with the **winter season**

	$[kWh/year]$	$[kWh/m^2year]$	$[kWh/m^3year]$
Net energy demand - Heating	70581.9	**37.27**	10.65
Electric consumption heat pumps	35098	**18.53**	5.30

Table 7.6: Heating energy consumption

As we can see from the table 7.6, the net energy demand for heating reach more than 70 thousand of kWh, but the most interesting value is the 37.27 $[kWh/m^2year]$ which allowed to make an easy comparison with the consumption of different buildings.
As we can see in the figure 7.7, I obtained the classical shape of the heating that goes to decrease as much as we get to the warmer months and goes to zero between 15th of May and 15th of September.
The peak power is reached during the coldest day in January and reach a value of 42.1kWh, almost double than the one find out with WinWatt. The difference could probably stay into a more accurate model for the solar heat gains, which has lower values than the ones considered from the software[2], increasing the heat energy required, due to the infiltration, which were not considered in the primary analisys. Moreover, in the excel file, I take into account just the 80% of the people previously calculated. The reason for that choice, is that having the maximum number of people allowed inside every day, it is almost impossible, and I think it is not going to represent the reality, so as already explained, I followed a different curve for people, which also reach the maximum number during the noon and the 01:00 p.m., i.e. during the less cold hours.

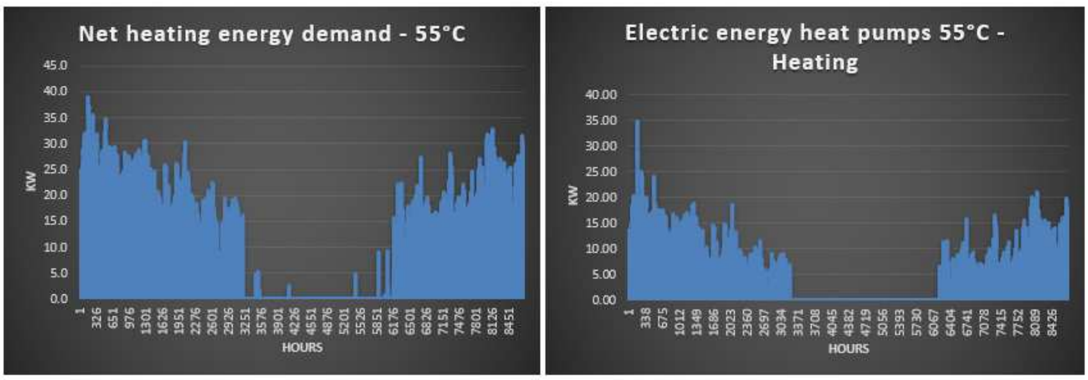

Figure 7.7: Net energy demand and final electric consumption

[2]Probably because the software was considering clear sky all over the whole year

	$[kWh/m^2year]$
Coventional buildings without energy reduction strategies	150-200
Conventional buildings with energy reduction strategies	80-100
Low energy buildings	30-50
Passive buildings	<15
Zero energy buildings	0

Table 7.7: Heating energy consumption in different buildings

As expected, looking to the table 7.7, which report the average net energy consumption for different buildings construction, we can see that our building, belongs to the category of "low energy buildings" reaching low values of energy consumption. Result expected considering that we're talking about a building build up in the 2018 where all energy-saving strategies were taken into account. In fact, usually in this type of constructions, the higher item of consumption, is traditionally given by the electric consumption of the fans whenever they are using an AHU. The base case, present an electric consumption as showed in the table 7.8

	[kWh/year]	[kWh/m2year]	[kWh/m3year]
Fans - Electric energy consumption	157257	83.03	23.73

Table 7.8: Fans electric energy consumption

In this case, I obtained huge numbers, which represent the 70% of the total electric consumption, and it's something completely out of the normal functioning, which would lead to totally incorrect sizing, and a totally waste of energy. For this reason, a case like this cannot be taken as a possible solution.
The reason for so high values is due to the fact that in for every hours of functioning, the system is using the 100% of the volumetric flow rate, with the maximum possible recirculation volume. This lead to a ΔT low, with a supply air temperature close to the internal one. In order to reduce the electric fan consumption, one of the possible strategy, is the one used in the second scenario.
Instead, regarding the inlet temperatures, I obtained the following values:

- T_{min} = 21°C
- T_{max} = 23.85°C

The 21°C are reached during that hours where we're recirculating the total amount of air, and it's the lowest value possible. Instead during the colder hours, I obtained, 23.85°C, which is a good value, because it means that the system is not consuming too much energy heating the air towards high temperatures, but on the other hand the total amount of air is being utilized, thus consuming a significant amount of electric energy. So, the inlet temperature, and the air mass flow rate either, will be adjusted in the second scenario. The results obtained by the **summer case** are the following:

	$[kWh/year]$	$[kWh/m^2year]$	$[kWh/m^3year]$
Net energy demand for cooling	66952.5	35.35	10.10
Electric consumption heat pumps	20268.19	10.70	3.06

Table 7.9: Cooling energy demand - Base case

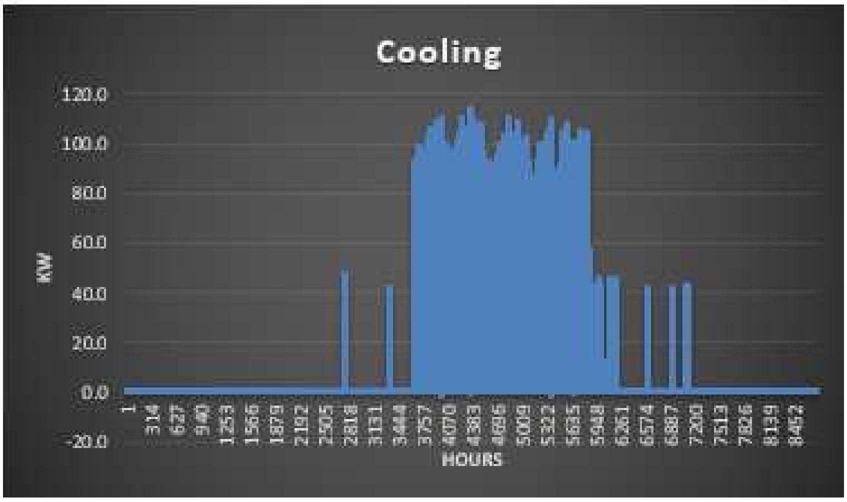

Figure 7.8: Net energy demand for cooling

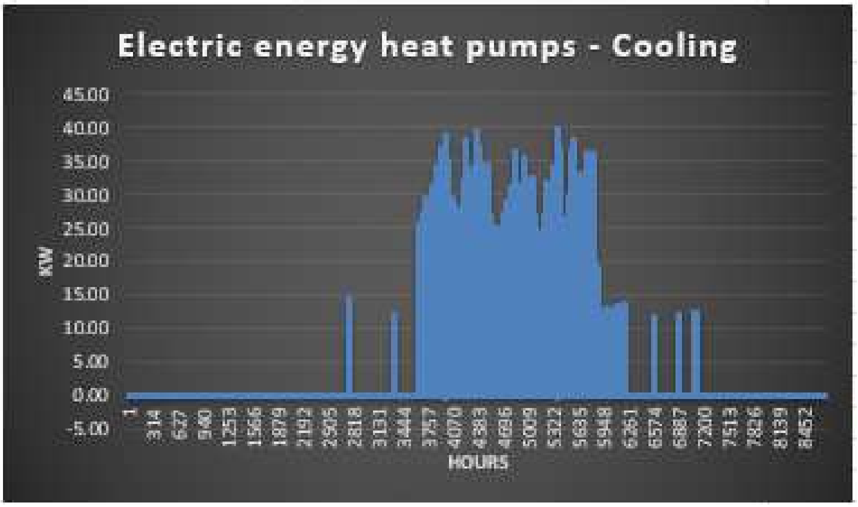

Figure 7.9: Electric consumption of the heat pumps in cooling mode

As expected, the most though point, is the cooling load, because even if the total amount of the whole year for heating is higher, the maximum peak load of the cooling, is 114[kWh], which is more or less three times higher that the heating one.
Also, through the value of EER, I calculated the electric energy required from the heat pumps in order to feed the AHU, which as we can see in the graph 7.9 follows the shape of the energy required for cooling, figure 7.8, but it's of course lower than the thermal energy required. If we consider the EER value in the graph 7.6, we can see that even in the hottest days, when the external temperature it's close to 35°C, the EER doesn't go below 2.9. This is a pretty good result due to the great performance of the heat pumps, and consequently, that's lead to a low electric energy consumption.
As for the winter case, another important aspect is the supplying temperature after the cooling coils. Of course, as for the other parameters, the supplying temperature has been calculated, using the power equation, and I found out:

- $T_{max} = 25.1$°C
- $T_{min} = 16.1$°C

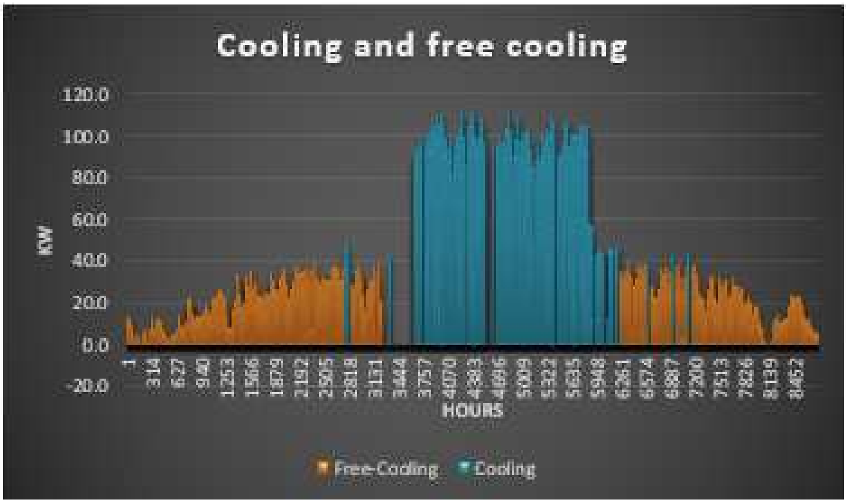

Figure 7.10: Net energy demand for freecooling

The two temperatures obtained, are in the range expected. The lower one, 16.1°C, is too low to be inject into the room, because could cause discomfort because is too cold. For this reason, it is necessary heat up the temperature with the post heating coil at least at 18°C. This lead to an additional heating energy which is 1038.9 kWh.

7.4 Freecooling

7.4.1 Definition and technical aspects

When a building is cooled naturally, i.e. without a refrigeration system, it is referred to as free cooling. A source of the cold air can be retrieved from the outside air flow, as well as from the water in rivers or lakes or directly from the ground.
There are two different types of freecooling, however, I'm going to explain just the direct one, which is the technique that has been adopted.

- **Direct freecooling**

The most common case is free cooling with outside air, which, in cool night hours, can make a valuable contribution to room cooling. Direct free cooling occurs when cold air is introduced into the building at night and the outside temperature is below 18°C (in summer, this is normally the case between 24:00 and 6:00). Free cooling can be achieved by simply keeping the windows open or with the employement of a ventilation system, therefore, simple and energy-efficient solutions. With direct free cooling, in the short time available in the summer nights, however, it is often not possible to remove all of the heat accumulated during the day. The inertia of the masses and the small temperature difference between inside and outside slow down the process considerably. Very substantial air flows are needed, thus a significant amount of 'cold air', in order to lower the temperature significantly. Moreover, 'tropical' nights, with outside temperatures above 20 °C, are increasingly frequent. If these last days or weeks, direct free cooling can become just about inefficient.

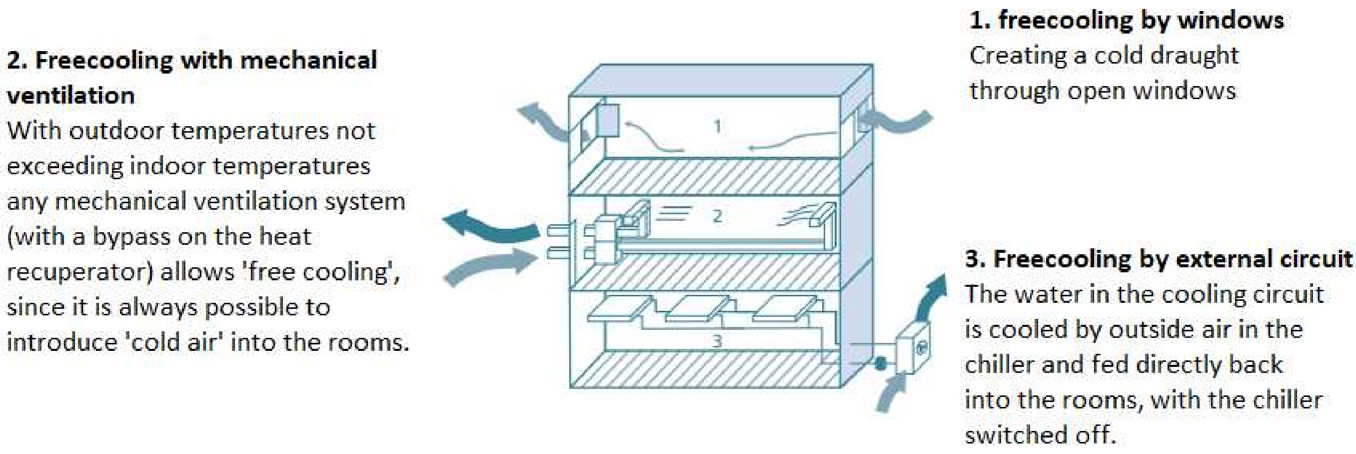

Figure 7.11: Direct and indirect freecooling

However, the use of free cooling is especially interesting during spring, autumn and winter, when, thanks to the low outside temperatures, it is possible to cool efficiently. Therefore, by taking advantage of the winter and mid-season, in addition to the large volume flow of air that the AHU can handle, it was possible to cover a large part of the excess heat load during the winter and spring period. However, freecooling is not exempt from expenses of course, but I had to consider the electric consumption of the fans, which counts for the sole source of energy required from the fans.
Thus in order to evaluate the possibility of the freecoling during the working hours, period during which the office needed to remove the heat load in excess, I made an evaluation based on the comparison with the external temperature.
Once again with the usual power equation I calculated the ΔT required:

$$\Delta T = \frac{P}{\dot{m} * cp} \tag{7.16}$$

From there, I calculated the mixing temperature or supply temperature[3]. This temperature, fluctuates between:

- $t_{mix,min} = 18.1$°C
- $t_{mix,max} = 21$°C

I based the possibility of the freecooling on the temperature mentioned above, therefore I compared it with the external temperature, which leads to two different possibilities as exhibited in the table:

	Feasibility	**Working hours**	**kWh covered**
Text < Tmix	Feasible	997	28710
Text > Tmix	Not feasible	87	3041

Table 7.10: Freecooling feasibility

[3]It is equal to writing mixing or supply temperature, is the mixing temperature because it is just after the mixing point, and it is also the supply temperature because it doesn't need to pass through any heating or cooling coil

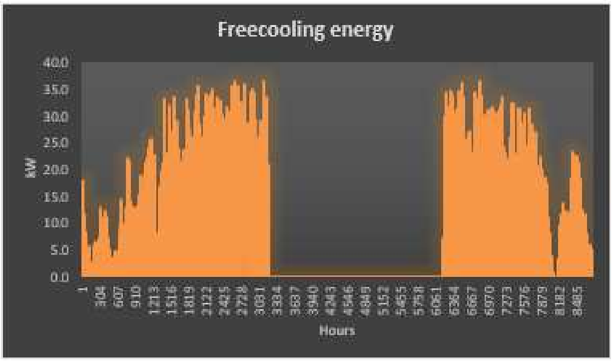

Figure 7.12: Freecooling energy

The freecooling results are feasible because thanks to the considerable volumetric flow rate, I have been able to cover the entire extra load even with a small temperature difference. That is because if the external temperature is below the mixing temperature, there is always a combination between the fresh mass flow rate and the recirculation one which leads to the right temperature that is required.
The result demonstrate that it is possible to work with the freecooling for 997 hours covering 28710 [kWh]. This is a considerably good result, as these represent almost half of the cooling load.
On the other hand, this technique is not feasible along the remaining 87 hours, where the office still needs 3041 [kWh]. But considering that these hours occur in staggered periods along the year, and no more than one or two hours by days, I think that integrate another chiller to supply to this load, could be too much, and a waste of money and resources.
The consequence of these 87 hours, could be a temperature inside the building slightly higher than the set point 21°C for few hours.
Also looking to the minimal and maximum temperature obtained, we can say that the minimal one is suitable for cooling purposes, because it is higher that 18°C, thus doesn't need post heating. Instead, in the case of the maximum temperature, which is 21°C, the employment of the freecooling is not feasible as it is impossible to cool down a room by injecting air at the same temperature which is at 21°C .
It could be seen in the figure 7.12, that the freecooling has been interrupted all of sudden, that is because along those hours, the external temperature started to increase too much to allow for the cooling of the office. So from then on, during the summer season, it is the cooling coil responsibility to supply the necessary power.
Regarding the electricity consumption, indirectly, it has been already calculated because in order to get every mixing temperature the elasticity, due to the total mass flow rate, is fundamental. That allows the possibility to be free to choose the correct air fresh flow rate and, consequently, the recirculation one. So considering that we are using the total amount of the mass flow rate, the calculation of the equation of the electric consumption is very straightforward as I used the equation 7.13, using 13.62 [kg/s] as volumetric flow rate for each hours when the fans are running, which multiplied for the working hours, which in turn results in the total electric energy which is demonstrated in the table 7.11.

Working hours	**Mass flow rate [kg/s]**	**Electric energy [kWh/year]**
997	13.62	15734

Table 7.11: Freecooling electric energy

7.5 Second scenario - Variable air volume VAV - 55°C Water heater coil

The aim of the second scenario, is to try to go forward for a smarter use of the machine, a smarter use of the system, in order to achieve lower energy consumption, which is always the final goal.
The necessity to study another case, comes from a massive electric consumption due to the functioning of the two fans, which are working almost the whole year dealing with the total possible amount of the volumetric flow rate, which is almost 41 thousand of cubic meter per hour. Using the total amount of the volumetric flow rate, as explained into the base case results; leads to a significantly low temperature gradient during every season. So in order to achieve other results, a larger temperature gradient has been set up in order to reduce the total volumetric flow rate.

7.5.1 WINTER

As already said, the aim is to reduce the volumetric flow rate. In order to do that, I increased the supply air temperature. I didn't fix a precise temperature, but I calculated it at the end of the cycle, and instead, I fixed a ΔT.
So the starting point of this new calculation is the:

- ΔT = 5°C

Bearing in mind that the energy required by the office is not going to vary, I used the usual equation of power 7.5.1,already known, using the inverse equation, in order to find out the mass flow rate that must be used for that power.

$$P = \dot{m} * cp * \Delta T$$

After that, the mass flow rate value must be compared with the minimal flow rate. That is because, of course, going below the minimal flow rate is not allowed for well being reason. That means that if the new mass flow rate goes below the minimal one, the latter[4] must be used for the calculation.
Once the mass flow rate required has been defined, it is then possible to calculate the recirculation subtracting the anew flow rate from the minimal one. Looking at the figure 4.5, and using the equation of the mass balance 4.19 I calculate the mixing temperature using the inverse of the same equation. Once defined the mixing temperature, I found out the supply temperature, because if the ΔT is equal to 5°C, the supply temperature will be the mixing temperature plus five degrees Celsius.

7.5.2 SUMMER

For the summer case, the process to find out the mixing temperatures, and the supply temperatures, are the same, because the logic is the same as well as is the equation.
I fixed the following values:

- ΔT = 8°C

[4]Referring to the minimal flow rate

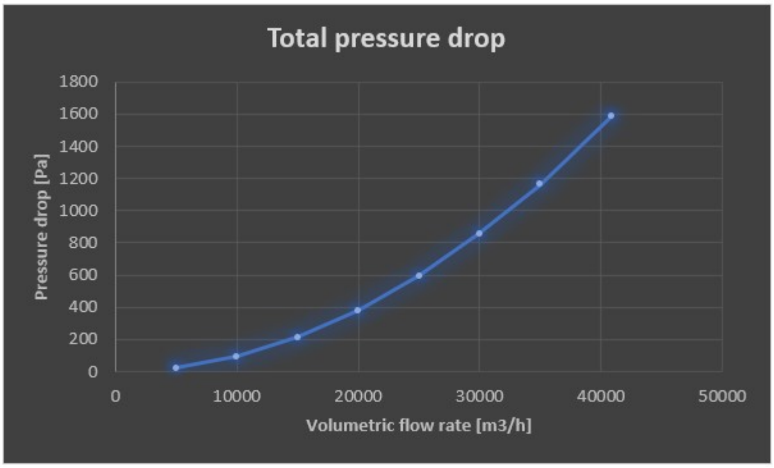

Figure 7.13: Pressure drop

Therefore, also for the summer season I followed the same process, I calculated the new mass flow rate, made the comparison with the fresh one, calculated the recirculation mass flow and then both temperatures, before and after the cooling coil.
What is also changing, is the total pressure drop, which could not be considered constant, but is changing with the different volumetric flow rate. Therefore, the pressure drop for the variable flow rate, should be calculated for each hours where the mass flow rate vary from the nominal one.
By the Rosemberg equation 7.17, it is easy to see that the total pressure drop are proportional to the velocity at power of two, and that create a exponential proportion between the pressure drop and the volumetric flow rate as we can see in the graph.
Shortly, the Rosemberg equation is:

$$Q = k * \sqrt{\frac{2 * \Delta P}{\rho}} \tag{7.17}$$

- Q = volumetric flow rate $[\frac{m^3}{h}]$
- k = calibration factor $\left[\frac{m^2 s}{h}\right]$
- ΔP = total pressue drop [Pa]
- ρ = air density $[\frac{kg}{m^3}]$

And from there, using the inverse formula, I can determine the ΔP:

$$\Delta P = \frac{Q^2 * \rho}{2 * k^2} \tag{7.18}$$

7.6 Results second case - VAV - 55°C Water heater coil

In this result I am going to show the differences among heating and cooling between the first and the second scenario, the freecooling, has been left as it was, because in the base case, I deem it is the best performance that I can achieve.

	Max mass flow rate [kg/s]	Tmin [°C]	Tmax [°C]
WINTER	8.38	24.68	26.88
SUMMER	12.14	15.07	23.10
TOT. ELECTRIC FANS CONSUMPTION [kWh/year]			60083

Table 7.12: Result second scenario

7.6.1 WINTER and SUMMER

Following the procedure showed in the previous chapter, I have been able to vary the mass flow rate fixing a certain ΔT.
It has been noticed that, even if I reduced the volumetric flow rate or the mass flow rate, the minimal fresh flow rate has been maintained, in order to ensure the well being of the office. I obtained the following results:

Interpreting the values obtained, with regard to the mass flow rate, it can be seen that although the temperature gradient has a significant value, the mass flow rate acquires important values in the peak demands, demonstrating the need for the building in question to handle large quantities of air. Even in the summer season, a flow rate almost equal to the nominal flow rate is required, while the flow rate for the winter season remains at lower values. On the other hand, as far as temperatures are concerned, for winter, we have significantly higher maximum and minimum temperatures, and a wider range for air conditioning. Also, as it happened in the case with the constant volumetric flow rate, the minimum inlet temperature for summer goes below the threshold which is 18°C touching the 15.07°C. This last temperature is not suitable to be injected in the office, therefore, also in this case, it is necessary to use the post heating coil which lead to an additional heating energy of 1211.4 kWh.

7.7 Third scenario - Variable air volume VAV - 35°C Water heater coil

In order to reach even lower consume, there is a third possibility, which consists in designing the heater coil for a supply temperature of 35°C instead of 55°C.
Increasing the number of the rows inside the heat exchanger, therefore increasing the contact time between the pipes where the water is flowing and the air, it is possible to heat up the air at the same temperature. This is also possible because I'm using an air system to provide the heat necessary for the office[5], which does not necessary require water at high temperature.
The main advantage of using water at lower temperature, is that we earn in terms of COP which increases significantly, as it is represented in the figure 7.14

[5]For instance, if I had a radiator as a vector of the heat, it should not be possible, because this system requires water at least at 65/70°C.

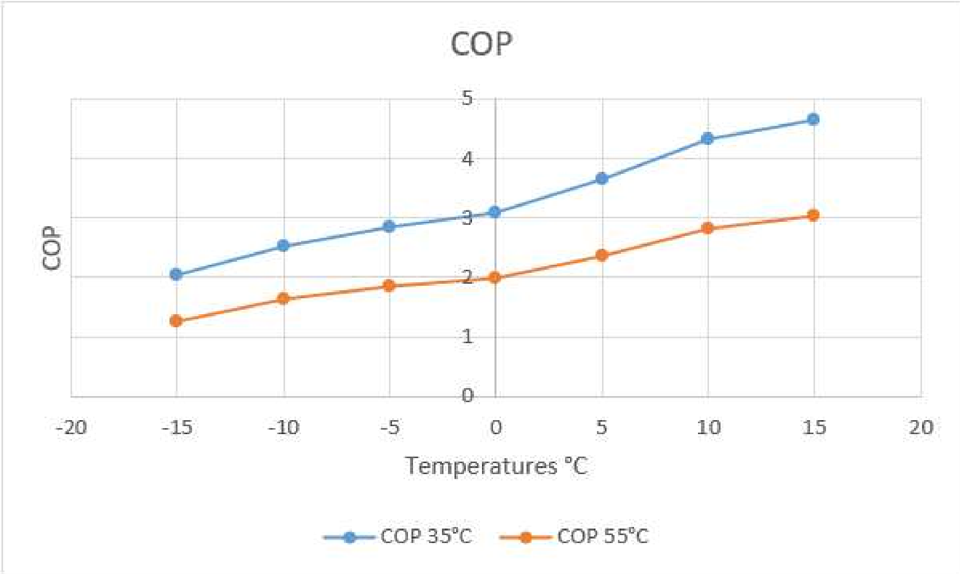

Figure 7.14: COP for two different supply water temperatures

In this way, it has been possible to reduce the electric energy for the heating from the initial one that was around 35000kWh/year to 22300 kWh/year, cutting the energy required of the 37%.

7.8 Comparison

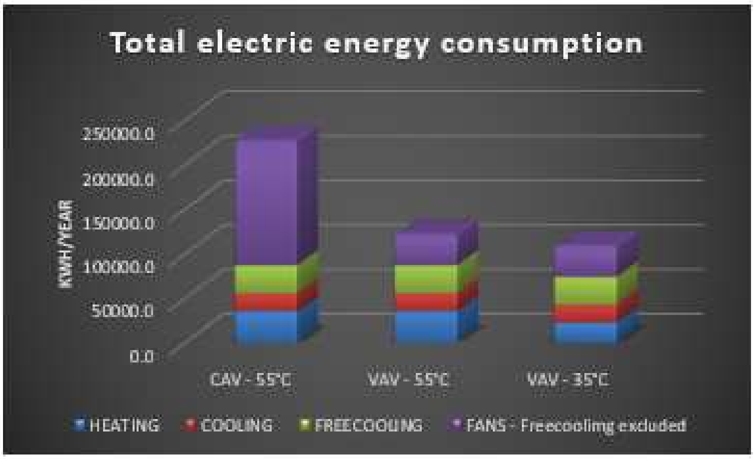

Figure 7.15: Total comparative electric energy

Therefore, up to here, I have extensively explained everything about the three different cases, I highlighted the temperatures both for winter and summer, and every thresholds was respected therefore no issues arose. For the mass flow rate, I have discussed the two different cases and how the mass flow rate varies. Therefore, what I am now going to compare is the final electric energy required by the three cases.
Looking to the graph 7.15[6] and bearing in mind that all the other values are not changing, it

[6]Into the graph, the energy of the freecooling and the ones for the fans, these have been separated even if the freecooling consumes electricity just from the fans. The separation is intentional as in order to better highlight

	CAV - 55°C [kWh]	**VAV - 55°C [kWh]**	**VAV - 35°C [kWh]**
Heating	35098	35098	22290.2
Cooling	20268.19	20268.19	20268.19
Freecooling	31464.7	31464.7	31464.7
Fans Freecooling excluded	125792	28618.4	28618.4

Table 7.13: Electric final energy for each case

is easy to see the variation of the electric consumption due to a variable air volume, which has been strongly reduced, and the convenience of a heating battery designed at 35°C; therefore is absolutely clear which could be the better choice in a possible application.

the difference energy consumed by the fans in heating and cooling mode

7.9 Placement of air diffusers

As shown in the chapter 1, for the well being of the employees inside the office, any kind of thermal discomfort must be avoided, and in order to achieve that, certain parameters must remain within a certain range. Consequently, the point of the air injection becomes fundamental, in order to support these requirements. Therefore, a correct design of the positioning of the air diffusers must be created for each room, this in turn leads to an equal diffusion of the injected air.
Of course, the air diffusers have to be designed in order to assist to the worst conditions,thus it is necessary to have the possibility to inject the maximum volumetric flow rate, which is 40855 [m3/h]. Moreover, beyond the aspect regarding the supply particularities, because we are talking about an air forced ventilation system, it is also important to take into account the characteristics of the exhaust. So the challenge was to displace both the exhaust and the supply diffusers, as the huge volumetric flow rate results in 194 pieces, 97 for the supply and the remaining for the exhaust. This ensures that the volumetric flow rate of the fresh air will be the same of the one that will be extracted.
In order to place the diffusers, the following data has been necessary:

- Office floor plan
- Square meters of each single room
- Schako program

The office plan, has been already presented in the chapter 3.
The Schacko program, is really simple but also fundamental for a design of this type, because, it is able to define the correct distance between the two diffusers. The following variables are necessary in order to set the required program:

1. Maximum volumetric flow rate.
   In this case, considering the huge amount of volumetric flow rate, two types of diffusers have been chosen. The models are the same, but they are different in regards to the volumetric flow rate that they can handle, the first one is able to inject $500[m^3/h]$, for the bigger rooms, and the second, for smaller offices, can inject up to $350[m^3/h]$.
2. The y-coordinate.
   I considered an average height of the room around 3m, and in the y-coordinate,the distance between the ceiling and an average height of a person should be taken into account. Considering an average of 1.7m, 1.3m was deemed appropriated.
3. Maximum velocity.
   Set at 0.2 $[m/s]$.
4. Set up of the diffusers.
   As said in the previous chapters,the setting of the fins of the diffusers have been chosen, which is a combination between winter and summer setting.

Once all of these parameters have been set, the programme automatically calculates the minimum distance the diffusers must stand between each other and the minimum distance they must keep between finishing and load-bearing elements such as walls, pillars etc. to prevent the flow from being altered. Therefore, this is the starting point. Regarding the minimum distance I'm talking just about the supply diffusers, because the line of the exhaust doesn't present any problems even if they are close to the walls. Once that the space available has been individuated, as a general rule, a rule used by the designers, I consider that one diffuser was able to cover more or less 10 [m^2]. Then using this value and the ones calculated, I tried to adjust, trying also to keep everything as in line as possible in order to optimize space.

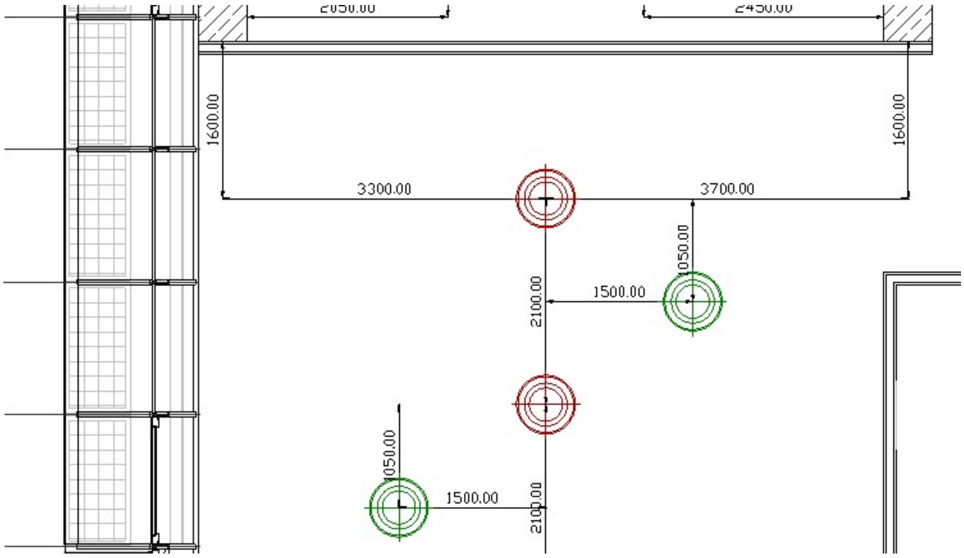

Figure 7.16: Particular drawing in autocad about the diffusers displacement

As we can see in the figure 7.16,they have been appropriately measured. The measures have been carried out taking into account the fixed elements of the office, for example the pillars, the lift, the stairs etc. That is because, using internal walls of the plasterboard, the shape of the internal office could change, but of course the positioning of the diffusers must remain the same.
There is also a third diffuser which has been used, and it is a diffuser which supplies the fresh air and removes it at the same time. This one in particular has been used in the smallest offices, which have a maximum volumetric flow rate of about 200 [m^3/h].
And also, on the base of the diffusers, I calculated the air change rate for each rooms. Considering to use the total volumetric flow rate, the air change rate demonstrated values around 7-8[h^-1] which is a significantly high number. Therefore, by using a variable air change rate these values (i.e. 7-8) have been reduced reaching the values of 1-2[h^-1].

Conclusions

Interpreting the electricity consumption data, at best[7], we see that the total consumption of the entire system equals:

	$kWh_e/year$	$kWh_e/year * m^2$
Total electric consumption	102645.9	54.16

Table 7.14: Total electric consumption

Thinking of an increasingly decarbonised future, one can see how convenient and possible all-electric operation is. There is a slight discrepancy with the peak powers due to some variations such as the reduction of solar heat gain to try to get closer to a real situation probably due to the reduction of the solar heat gain that I have done trying to resemble as much as possible the reality.
What I would like to emphasise most is the AHU itself, which is a large machine. Generally, the needs of a blueprint of this size, having to handle such a large air flow, are met by a machine that can handle the minimum air flow and an auxiliary system that can handle the rest of the load demand. The choice of a single machine was mostly a choice of the Hungarian supervisor who wanted to insist on this route. Personally, I would have preferred a smaller machine with an auxiliary system such as active beams or fan coils. In any case, it was a choice that had advantages and disadvantages, which are reported below.

Advantages:

1. I've been able to handle the 95% of the cooling load during the heating season with the freecooling avoiding the installation of an additional chiller.
2. The maintenance is not going to be in the office area, avoiding to disturb the normal office work
3. Lower cost of automatic system
4. Easier energy consumption measurements

On the other hand, some serious **disadvantages** can arise:

1. The size of the air distribution system
2. It is impossible to regulate the system room by room as it is only possible to regulate the system globally
3. To handle the heating cooling and the fresh air with just a single system might be risky, because in case of malfunctioning all the system will stop working, with no possibility backup.

[7] I mean with the VAV and the heating coil at 35°C

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Pictures references

Fig. 1.1 - Iain Campbell. Body temperature and its regulation. Anaesthesia & Intensive Care Medicine, 9(6):259–263, 2008.

Fig. 1.2 - https://www.researchgate.net/

Fig. 1.4 - ISO7730 ISO. 7730: Ergonomics of the thermal environment analytical determination and interpretation of thermal comfort using calculation of the pmv and ppd indices and local thermal comfort criteria. Management, 3(605):e615, 2005

Fig. 1.5 - ISO7730 ISO. 7730: Ergonomics of the thermal environment analytical determination and interpretation of thermal comfort using calculation of the pmv and ppd indices and local thermal comfort criteria. Management, 3(605):e615, 2005

Fig. 1.6 - ISO7730 ISO. 7730: Ergonomics of the thermal environment analytical determination and interpretation of thermal comfort using calculation of the pmv and ppd indices and local thermal comfort criteria. Management, 3(605):e615, 2005

Fig. 1.7 - ISO7730 ISO. 7730: Ergonomics of the thermal environment analytical determination and interpretation of thermal comfort using calculation of the pmv and ppd indices and local thermal comfort criteria. Management, 3(605):e615, 2005

Fig. 1.8 - ISO7730 ISO. 7730: Ergonomics of the thermal environment analytical determination and interpretation of thermal comfort using calculation of the pmv and ppd indices and local thermal comfort criteria. Management, 3(605):e615, 2005

Fig. 2.1 - https://www.testo.com/en-UK/testo-400/p/0560-0400

Fig. 2.2 - https://www.testo.com/en-UK/comfort-probe-for-degree-of-turbulence-measurement-according/p/0628-0143

Fig. 2.3 - This picture hasn't been taken from a website, but comes from my own telephone and has been taken in the laboratory of HVAC in the UniDeb.

Fig. 2.4 - This picture hasn't been taken from a website, but comes from my own telephone and has been taken in the laboratory of HVAC in the UniDeb.

Fig. 2.5 - https://www.simscale.com/blog/displacement-ventilation-cfd/

Fig. 2.6 - This picture hasn't been taken from a website, but comes from my own telephone and has been taken in the laboratory of HVAC in the UniDeb

Fig. 2.7 - This picture hasn't been taken from a website, but comes from my own telephone and has been taken in the laboratory of HVAC in the UniDeb

Fig. 2.8 - Autocad draw

Fig. 2.9 - Autocad draw