Analysis and control of humidity in the industrial environment through rooftop and humidification systems

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Master Dissertation

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Abstract

Heating, Ventilation and Air Conditioning (HVAC) installations are an increasingly demanding area of activity. One of the reasons for this increase in demand is climate change, combined with increased innovation and increased attention to more energy-efficient solutions.

The present work focuses on the control of humidity in the industrial environment. A case study for air conditioning of a production hall is presented, in which an costs analysis is carried out focusing on two types of dehumidifiers (dehumidifying cooling coils and desiccant dehumidifiers) to evaluate which solution is the more economically viable. Both dehumidifiers had the ability to supply air to desired indoor conditions.

The author complemented the study with the use of a computer program called Hourly Analysis Program (HAP) that calculates thermal heat loads and makes energy consumption simulations. It was very resourceful to define the total load of the required dehumidifying cooling battery. For the desiccant dehumidifier, all calculations were hand made.

It was concluded that the desiccant dehumidifier, despite being 112100 € more expensive, per equipment, had a lower annual electrical cost of around 84522 €. To calculate the final cost, the investment cost of the equipment was added to the cost of electricity consumption in a year. The total cost of the desiccant is around 384522 € after a year, in comparison to the rooftop unit that is about 1777078 €. With the desiccant, on the electricity consumption alone it is possible to save around 1540956 € in the first year of operation. After 20 years of useful life, the difference is around 30819102 €. With these values, it was possible to conclude that the desiccant dehumidifier was economically more viable and, therefore, the right investment to make in this case study.

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Nomenclature

ACH	Air Changes per Hour	
BF	Bypass Factor	
CF	Contact Factor	
DB	Dry Bulb Temperature	[ºC]
EER	Energy Efficiency Ratio	
EST	Effective Surface Temperature	
h	Enthalpy	[kJ/kg·K]
HR	Humidity Ratio	[kg_v/kg_{da}]
$\dot{m}_{d.a}$	Mass flow of the dry air	[$kg_{d.a}/h$]
$\dot{m}_v$	Mass flow of the condensate	[kg_v/h]
MCDB	Mean Coincident Dry Bulb Temperature	[˚C]
P	pressure	[Pa]
P_s	saturation pressure of water vapor	[Pa]
P_v	partial pressure of water vapor	[Pa]

T	temperature	[˚C]
T1	Entry temperature for the desiccant dehumidifier	[˚C]
$T2$	Exit temperature for the desiccant dehumidifier	[˚C]
T_{db}	Dry-bulb temperature	[˚C]
T_{wb}	Wet-bulb temperature	[˚C]
T_{dp}	Dew point temperature	[˚C]
RH	Relative Humidity	[%]
U	Thermal transmittance	[W/(m^2·K)]
$\dot{V}$	Flow volume	$[\mathrm{m^3/h}]$
V	Volume	$[\mathrm{m^3}]$
$\bar{v}$	average specific volume	$[\mathrm{m^3/kg(d.a)}]$
v_1	Entry specific volume humidity for the desiccant dehumidifier	$[\mathrm{m^3/kg(d.a)}]$
v_2	Exit specific volume humidity for the desiccant dehumidifier	$[\mathrm{m^3/kg(d.a)}]$
W	Humidity Ratio	[kg_v/kg_{da}]

x_v mole fraction of the water vapor in a mixture

x_s mole fraction of the water vapor in a saturated mixture

Greek letters

ω	Humidity ratio	[kg_v/kg_{da}]
ϕ	Relative humidity	[%]
$\phi 1$	Entry relative humidity for the desiccant dehumidifier	[%]
$\phi 2$	Exit relative humidity for the desiccant dehumidifier	[%]
$w1$	Entry humidity ratio for the desiccant dehumidifier	[kg_v/kg_{da}]
w_2	Exit humidity ratio for the desiccant dehumidifier	[kg_v/kg_{da}]
Δw	difference between humidity ratios	[kg_v/kg_{da}]

Abbreviations

DX	Direct Expansion Cooling
CHW	Chilled Liquid Cooling System
HAP	Hourly Analyses Program
CAV	Constant Air Volume
HVAC	Heating, Ventilation and Air Conditioning

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1. Introduction

1.1. Project framework and motivation

Air conditioning systems are designed to satisfy the thermal comfort of the occupants and process conditions, in a specific space or a section of a building [1]. Providing a comfortable and healthy indoor air quality for any type of building is a major concern of HVAC engineers. Exposure to pollutants can lead to serious complications not only in terms of health deterioration but also to poor quality manufacturing results of several industrial processes [2].

Healthy indoor air and ventilation have become prominent requirements and in high demand in the building industry [20]. In that regard, one of the most prevalent issues in buildings that affect the human health is humidity [20]. Humidity control plays a crucial role in reducing the health risks of moisture and moisture related problems [20].

A high-water vapor content in air can also introduce many problems in the industry. In fact, humidity is a critical factor for quality of manufacturing processes and people's comfort. High humidity ratios not only encourage ill health but also contaminates material surfaces such as metal [4]. The present work focuses on the control of humidity in the industrial environment. It is written for anyone that operates a building or process which is influenced by atmospheric humidity [3].

In this thesis, the author explores dehumidification equipment for industrial environment. In specific two types of dehumidifiers: dehumidifying cooling coils and desiccant dehumidifiers. Those are the most prominent equipment's available on the market. It is presented a case study for climatizing a production hall to assess which one is the most viable solution. For the study of the dehumidifying cooling coil the author resorted to the assistance of a computerized tool. Such tool was resourceful to calculate heat loads and the cooling coil load capacity. Along this thesis the author explains what the program consists of, how does it work and interprets the results generated. For the calculations of the desiccant dehumidifier, they were all hand made.

A costs analysis is also an objective considered in this paper.

1.2. RACE

This dissertation was carried out at RACE company which is a reference in the sector of climatization for several different areas and purposes.

RACE is specialized in designing large-scale HVAC systems, responding with distinction to all associated technical and environmental standards and using the most advanced solutions for energy efficiency.

RACE has a high level of experience installing and maintaining HVAC systems for diverse locations: shopping centers, hotels, hospitals, office and service buildings, industry, housing, health clubs and spas. It is a company with more than 30 years of experience and with 5000 commercial or industrial installations. Not only in HVAC but also RACE gives their services to refrigeration projects. Whether it is for industrial refrigeration or commercial refrigeration.

1.3. Project objectives

The present work was elaborated to increase scientific knowledge on the practical implementation of dehumidifying equipment. To understand how the dimensioning and selection of such equipment is elaborated in the business world. The main objective was to perform a case study and test two different dehumidifiers and assess their operation. In order, to cover all the referred design specifications, this report presents the following objectives and tasks:

- Study and understand the operation of cooling coils and desiccant dehumidifiers;
- Research the main advantages and disadvantages of each dehumidifier;
- Preform a case study that allows to test each dehumidifier on the same operating conditions;
- Analise the results of the tested strategies and identify the best equipment to implement, in accordance with its characteristics, namely energy costs and equipment investment costs;

1.4. Dissertation structure

This dissertation is organized into 6 chapters, including Chapter 1 which serves as a contextualization and introduction to the others.

Chapter 2 presents a literature review of the areas relevant to the development of the present work. In this chapter it's addressed both dehumidification equipment's predominantly used, which are: dehumidifying cooling coils and desiccant dehumidifiers.

Chapter 3 presents a detailed demonstration of the computer tool used to calculated heat loads. Description of the program and how to use is demonstrated in this chapter.

Chapter 4 presents the solutions to the calculations.

Chapter 5 presents a costs analysis, to compare the investment and the electricity consumption costs of both equipment.

Chapter 6 summarizes the conclusions obtained with the project and perspectives of possible works to complement or improve this dissertation are presented.

2. Literature review

When industrial interiors get damp and stay damp, problems for the occupants and structure, materials, and machinery of the building recur. Very high indoor humidity has been associated with human health problems such as short useful life of equipment and reduced occupant satisfaction because of odours and stains. These and related problems can be costly and disruptive [2].

In this chapter, a review will be given about HVAC fundamentals and how to measure and calculate humidity in any kind of building. First, on should understand why humidity is harmful to human's health and its environment. Then, a review will be given on how to determine heat loads and how to interpret the psychometric diagram. Lastly, a review on the types of dehumidifiers available on the market and what they are constituted of.

2.1. Humidity risks

2.1.1. Physiological stress and discomfort

When a person is subjected to a hot climate, he or she controls his body temperature by sweating. It is a natural and biological cooling effect done by humans and other mammals [4]. This cooling effect becomes continuously more difficult to achieve with higher values of relative humidity [4]. The rate at which sweat can evaporate depends on how much moisture there is in the air and how much moisture the air can hold. As the air becomes drier, the human body starts to lose moisture. The physiological system begins to be under stressed as the heart rate increases to recirculate the blood faster [4]. This effect is presented in the Figure 1. For skin temperature it is illustrated in the graphic (a). For heart rate it is illustrated in graphic (b) and sweating is illustrated in graphic (c).

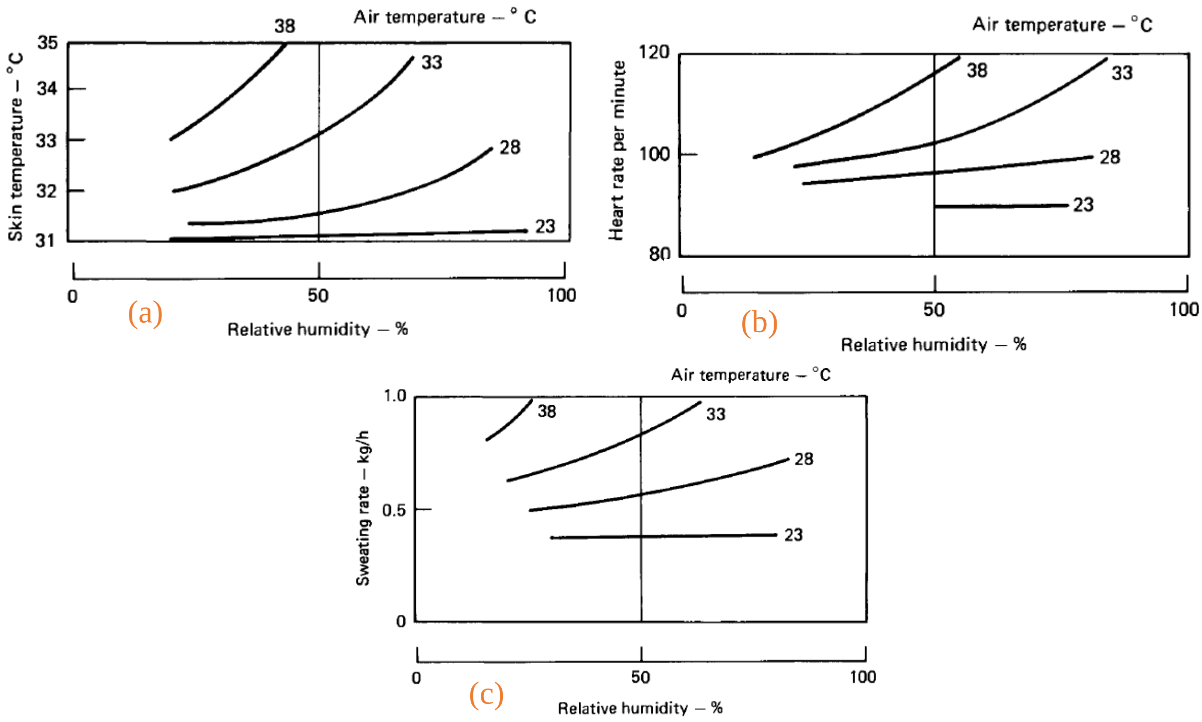

Figure 1 – Physiological behaviour of the human body to the increase of the air's relative humidity [4] for skin temperature (a), heart rate (b) and sweating (c).

Humidity is the critical factor for thermal stress, and in very hot conditions it should be as low as possible. Mild discomfort can also occur in less extreme conditions [4]. The human body is sensitive to climate as it is most often exposed to the surrounding atmosphere. The typical human perception of air humidity is shown in the Table 1.

Table 1 - Influence of Temperature and Humidity on Human perception of Air [5].

RH at 32°C (90°F)	Dew Point		Human Perception
≥73%	Over 26°C	Over 80°F	Severely high. Even deadly for asthma-related illnesses
62%–72%	24°C–26°C	75°F–80°F	Extremely uncomfortable, fairly oppressive
52%–61%	21°C–24°C	70°F–74°F	Very humid, quite uncomfortable
44%–51%	18°C–21°C	65°F–69°F	Somewhat uncomfortable for most people at upper edge
37%–43%	16°C–18°C	60°F–64°F	OK for most, but all perceive the humidity at upper edge
31%–36%	13°C–16°C	55°F–59°F	Comfortable
26%–30%	10°C–12°C	50°F–54°F	Very comfortable
≤25%	Under 10°C	Under 50°F	A bit dry for some

The rate at which transpiration can evaporate from the human body is related to the air's relative humidity. Therefore, the combination of high humidity and high temperatures is particularly dangerous. This combination reduces the body's capacity of cooling (air already full of moisture cannot absorb much more moisture expelled through the skin) and can cause considerable discomfort even lead to a heat stroke, exhaustion, or in more extreme cases, death [5].

It is worth mentioning that there is no specific temperature and humidity conditions in which everyone feels comfortable. People feel comfortable in a certain range of temperature and humidity. The comfort zones in which people feel comfortable are shown in Figure 2. Note that most people feel comfortable at higher temperatures if there is lower humidity. As the temperature drops, higher humidity levels are still within the comfort zone [5].

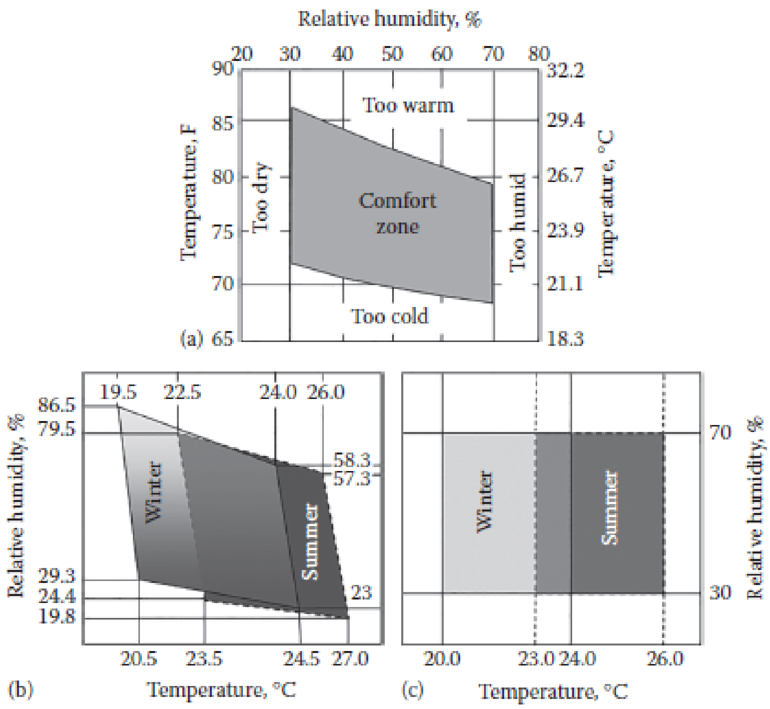

Figure 2 - Chart illustrating zone for comfort living [5].

2.1.2. Health issues

At the beginning of the XIX century, urban housing problems were often linked to poor ventilation and damp environments. Many design problems were solved following the development of clear building legislations around the world. However, damp conditions and condensation problems are recurring due to our changing lifestyles and the house occupant's behavior to save energy by reducing ventilation [4].

Many studies on the health effects of moisture exposure conclude that there is strong evidence for a true association between moisture and health effects. Damage caused by dampness can be associated with several different health effects and symptoms [5]. Common health issues due to mold or moisture exposure are:

- Irritative and general symptoms such as rhinitis, sore throat, hoarseness, cough, phlegm, shortness of breath, eye irritation, eczema, tiredness, headache, nausea, difficulties in concentration, and fever [5].
- Infections such as common cold, otitis, maxillary sinusitis, and bronchitis [5].

2.1.3. Mold

Outdoor air contains a variety of mold spores in suspension, and these are small enough to be carried indoors. In general, spores will not germinate below 70% relative humidity. The actual temperature conditions for germination vary greatly between the different types of mold. Once germinated, the mold thrives, and the speed of growth is a function of temperature and humidity [4].

Mold always leads to undesirable odours and ruins the quality of food products as well as building materials. Once mold starts to grow, water is a by-product of its metabolism, and it can then succeed even when conditions become drier. This means that it is essential to prevent the germination of the mold spores by ensuring that high humidity is avoided [4].

2.1.4. Humidity influence on the material properties and industrial processes.

The amount of water vapor in the air can affect human comfort, as well as efficiency of many industrial processes. The humidity that exists in the air along with temperature has a long-term negative effect on machinery and building materials. Therefore, measuring and controlling humidity with aim of improving the product quality is vital for many industries from the textiles, food processing, paper, and others [5].

The damage, caused by excessive relative humidity (RH), can lead to several problems, such as the corrosion of steel and metals [5]. Figure 3 shows the rate of corrosion of metal surfaces to different relative humidity's [4].

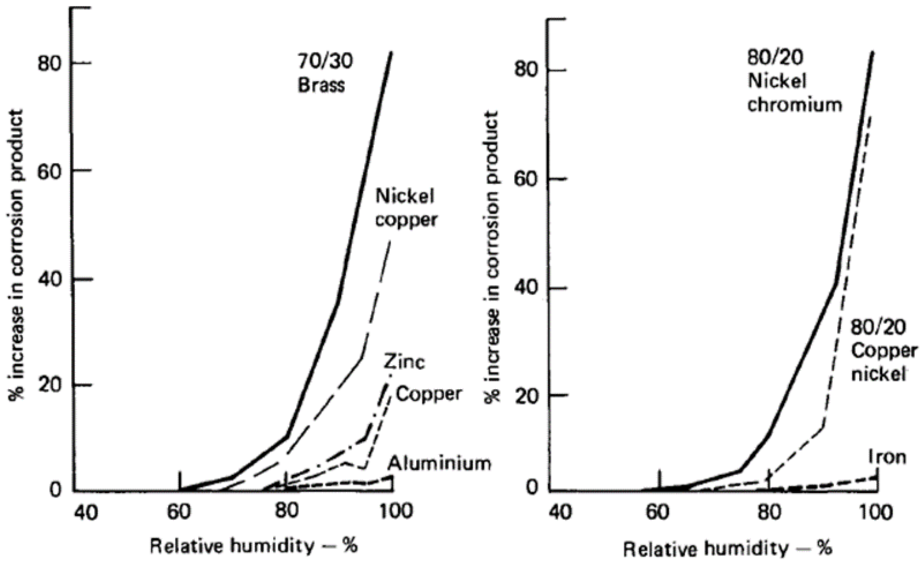

Figure 3 - Corrosion rates at room temperature for different humidities [4].

2.2. Methods of dehumidification

There are two known ways, among several, of dehumidifying air:

- Cooling the air to the point where the water condensates - lowering, therefore, the humidity ratio from the air in question.
- By passing the air through a desiccant material.

In this paper both of those methods of dehumidification are explored.

2.2.1. Cooling-based dehumidification technologies

This type of dehumidification system is characterized by converting the enthalpy of the air into sensible heat. An example of a simple dehumidification system is presented in Figure 4. The moist air goes through the surface of a cooling battery (or heat-exchanger), where the temperature of the air drops below its dewpoint. Afterward, the moisture condenses and drips to the nearest surface, giving up its latent heat of condensation. Essentially, the air goes through a process of cooling followed by condensation. The condensation of the water contained in the air reduces its humidity ratio. This is the process of dehumidification. The amount of water that is removed from the damped air depends on the temperature of the cooling battery – the lower the temperature, the drier the air [3].

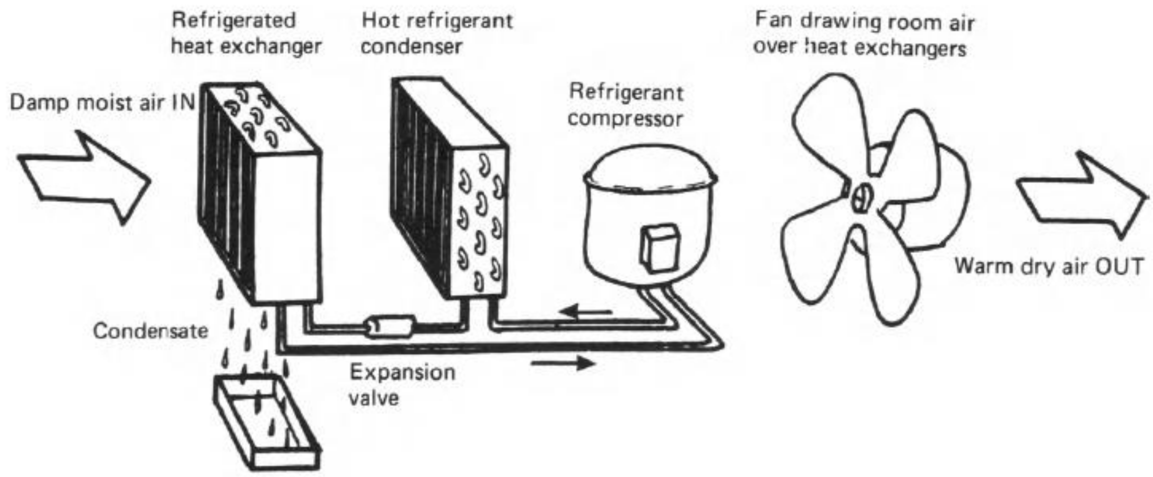

Figure 4 - Illustration of a dehumidification system using cooling coils [4].

For a better understanding of the cycle in question is presented Figure 5. The operating principle of this system is a refrigeration cycle that cools the air, condensing some of its moisture and then sending the resulting air back into space we want to dehumidify [3]. This cycle behaves like a heat pump, converting the latent heat that is given off in the condensation step, into sensible heat for a different air flow in a different location, using a refrigerant fluid to transport the heat [4].

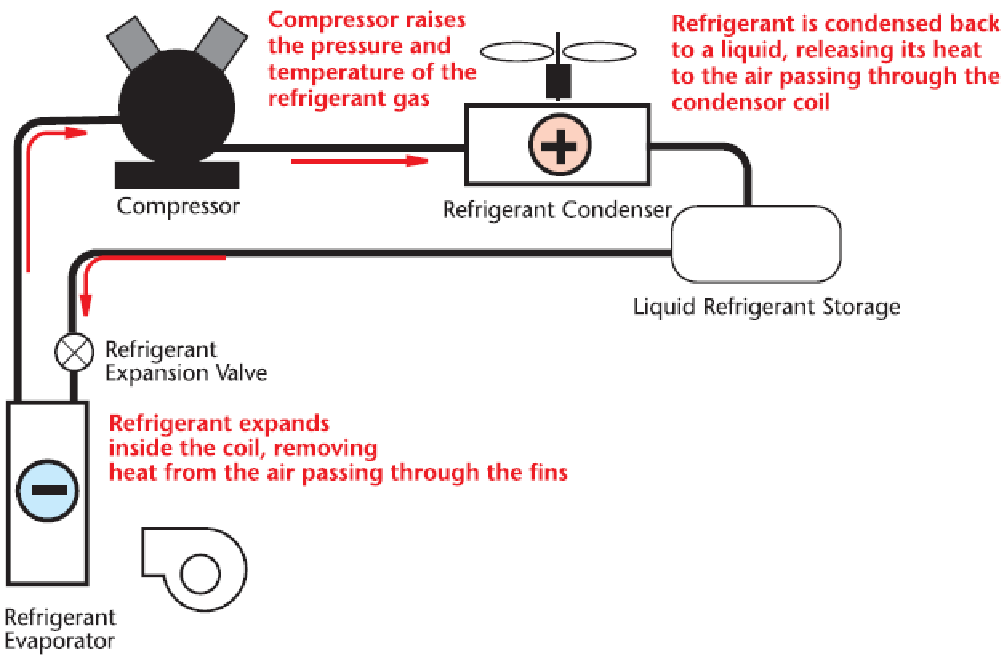

Figure 5 – Illustration of a cooling system using cooling batteries and refrigerant fluids to dehumidify de air [3].

The heat is removed from the air by transferring its thermal energy to a refrigerant fluid that is inside a battery. This battery is the evaporator of the refrigerant cycle, because inside the battery, the refrigerant is evaporating from liquid to gas. From the evaporator, the refrigerant fluid travels to a compressor where its pressure increases. The fluid now has a lower specific volume value, but the pressure increase has resulted in a temperature increase [3].

Figure 6 illustrates the processes the air goes when it is being dehumidified. As it flows through the cooling coil, it starts to release sensible heat. However, as soon as it reaches its dew point, and continues to be cooled down, all the excess water begins to condensate. That portion of heat that is released is called latent heat.

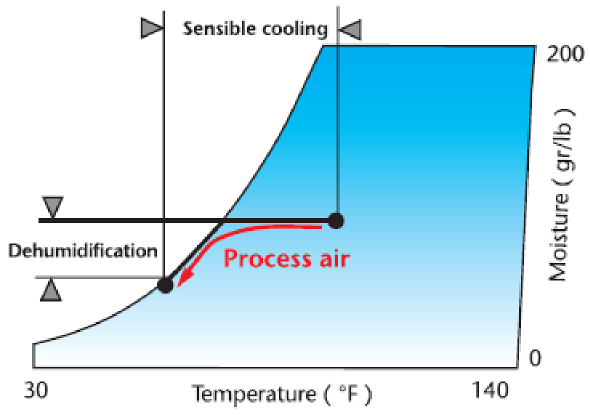

Figure 6 - Illustration of dehumidification air path [3].

For the refrigerant cycle to work properly, the refrigerant fluid now needs to release all the heat that received. Therefore, the fluid passes through a second battery called condenser. This condenser is often located outside the conditioned spaces where heat can be rejected to the outside air. After that heat release, the refrigerant fluid condenses back to the liquid state and is then directed to an expansion valve. The cooled refrigerant liquid can now return to the cooling battery (condenser) and restart the hole cycle again [3].

The configuration in Figure 5 is referred to as direct cooling. The evaporator is in direct contact with the air in the cooling battery, removing heat from the airstream and dehumidifying it. It is widely used in residential air conditioners and commercial rooftops cooling packages. However other equipment configurations are used, like the one shown in Figure 7. It is denominated as chilled liquid cooling and is also a viable alternative. It is commonly used in large commercial buildings and offices. The difference between the two is that the direct expansion units (DX) cool air using refrigerant and chilled liquid cooling system units (CHW) cool air using chilled water [3]. In other words, CHW systems have a refrigerant – water heat exchanger.

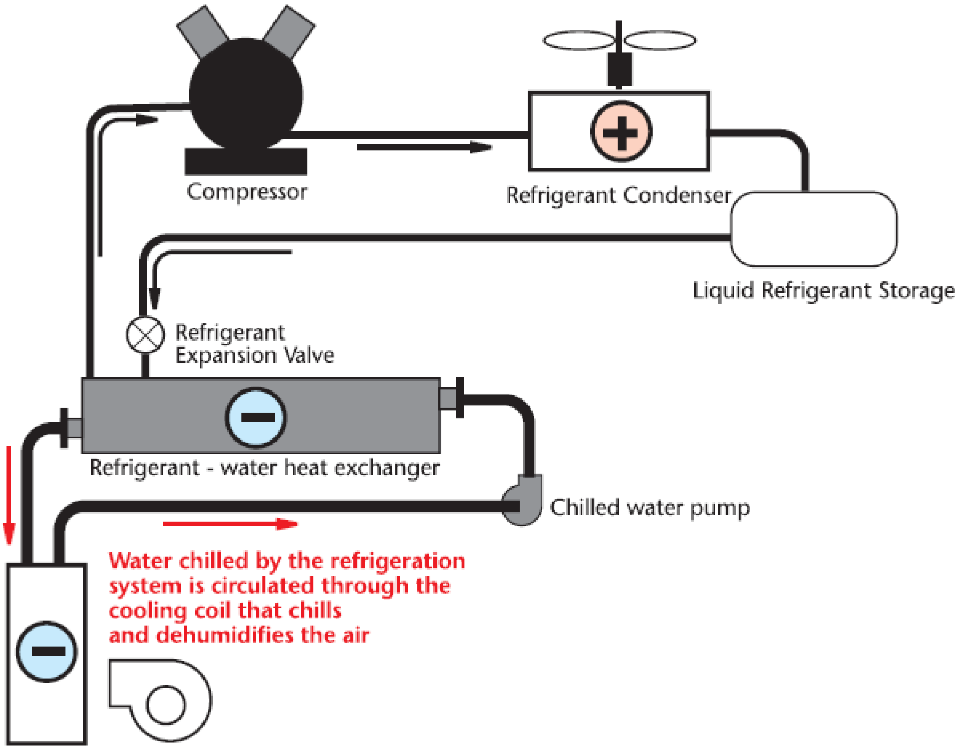

Figure 7 – Illustration of a Chilled liquid cooling system [3].

The direct expansion system, theoretically, has a higher efficiency, because the fewer heat transfer steps, the better the performance of the system itself. However, as it is illustrated in Figure 8, this performance really depends on the relative humidity of the income airstream. At low relative humidity, the air must be cooled down to its dew point before the condensation starts. Much of the refrigerant cooling capacity goes into lowering the air temperature, because of the heat that the refrigerant must extract and only a small part is left to condensate the water contained in the air [4]. This means that this cycle is not very effective at low relative humidity values.

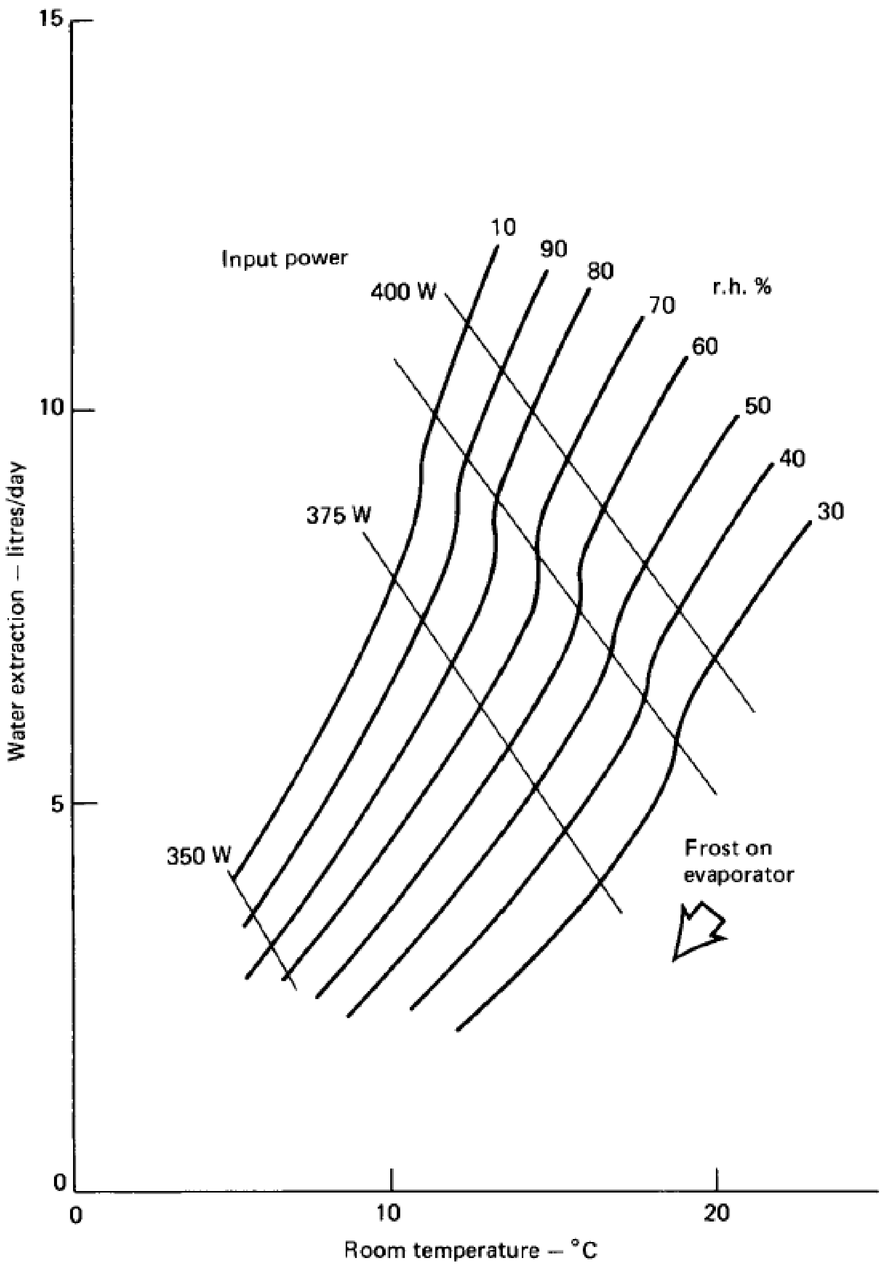

Figure 8 - Illustrative performance of a large DX domestic dehumidifier [4].

Some literatures, refer the CHW system as a better efficient arrangement than the DX system [3]. That is verifiable but only for configurations where it is necessary to have several chillers. When the intention is to have a considerable amount of small chillers systems, like the water coolers shown in the Figure 7, it becomes a very complex and expensive project if you are considering a DX configuration. The advantage of the chilled liquid cooling system is that you can have a steady load on the compressor and condenser, of the refrigeration cycle, when many different air streams must be cooled by a single refrigeration system [3].

2.2.2. Desiccant Dehumidifiers

This type of dehumidifiers are largely different from cooling-based dehumidifiers. It doesn’t remove the water from the moist air by means of a cooling battery. Instead, it mixes the moist air with a special material (adsorbents) or a fluid (absorbents) that captures the humidity [3]. Absorbents can be liquids or solids, like lithium chloride – much like an absorption refrigeration cycle, which collets water by exposing the moist air to salt – which becomes liquid as the absorb moisture [3]. Adsorbents, capture humidity by a special material, like silica gel, which the water molecules bond with without changing the nature of the material [4]. It consists of air being blown through a rotating drum of desiccant with a large surface area. So instead of cooling the air, the water vapor condenses inside the material releasing it latent heat of condensation. The thermal process that the air is submitted is illustrated in the Figure 9. The moisture is removed along a constant specific enthalpy.

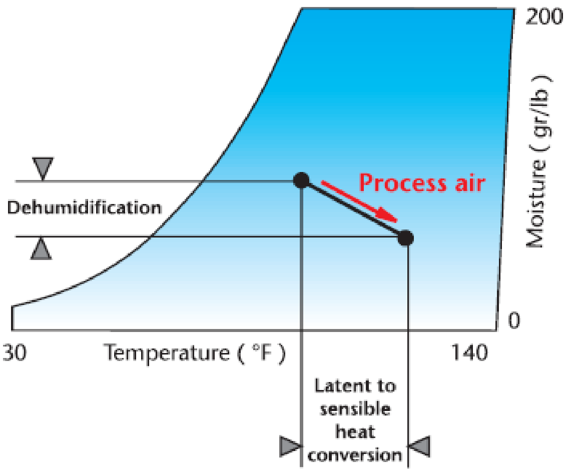

Figure 9 - Dehumidified air path of a desiccant dehumidifier [4].

The essential characteristic of a desiccant is the area of low vapor pressure created at its surface, as the moist air contacts with it. If the desiccant is cool and dry, its surface vapor pressure is low, and it can capture the moisture from the air which has a high value of vapor pressure when it is moist [4]. Figure 10 shows a typical configuration of a desiccant dehumidifier.

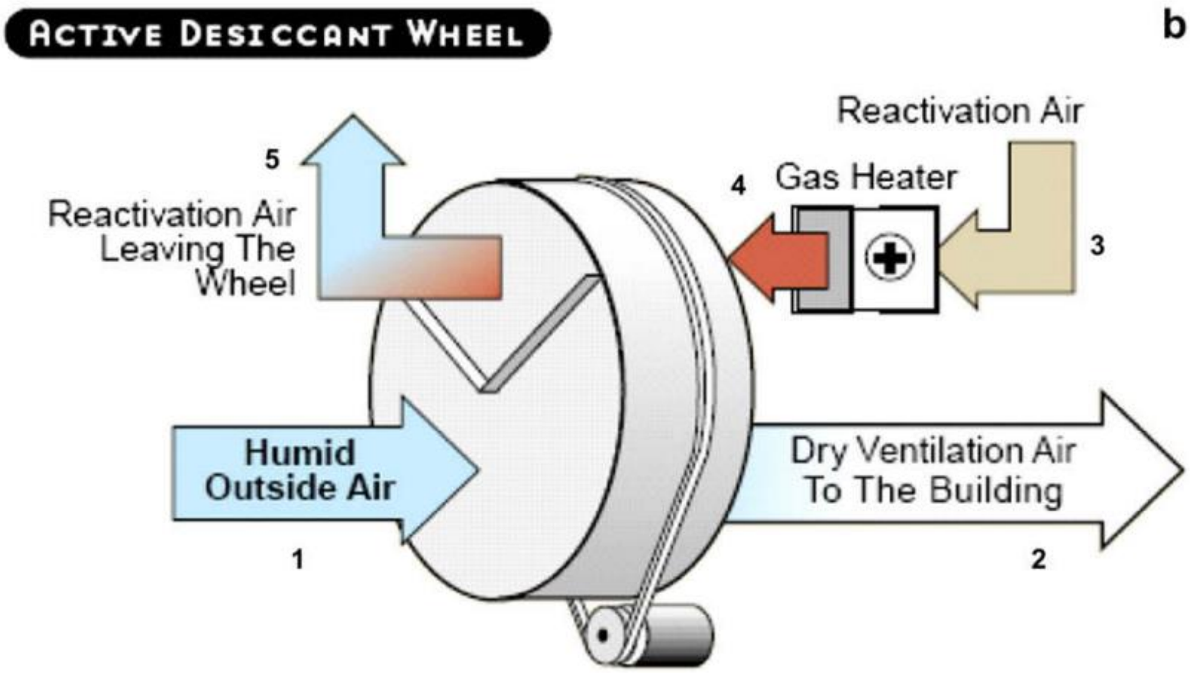

Figure 10 - Basic design of a solid sorbent rotary dehumidifier [15].

After the air passes through the desiccant, its temperature is raised by 10 to 20˚C and re-enters the conditioned space [3]. This temperature rise is directly proportional to the amount of moisture removed from the air — the drier the air leaves the dehumidifier, the warmer it will be. Also, it can happen that the enthalpy of the air being dehumidified increases slightly. This is because in several dehumidifiers, a residual amount of heat, can be left over from the desiccant reactivation, which then is carried into the dry air stream [4].

Typically, the desiccant is disposed in a round drum and while the drum rotates, by means of an electrical motor, is reactivated by blowing hot dry air through the matrix, as it can be seen in Figure 10. This process dries out the desiccant and the product of that is blown to waste. The cycle is repeated as the drum rotates [3].

Desiccant dehumidifiers are often used to provide air at low dew points. Applications requiring large volumes of air with low humidity ratios can be easily achieved by rotary desiccant units in a single pass [13]. Where extremely low dew points must be achieved (below 0 ºC), in some cases, the cooling battery has a risk of surface freeze [13]. However, the desiccant can fairly dehumidify at those temperatures.

3. Heat loads calculation

RACE presented a case study, as base exercise to compare the different dehumidification systems. The focus of this chapter consists in presenting the reader the object of the case study.

Thermal load calculations are the primary design basis for most heating and air-conditioning systems and components. These calculations affect the capacity of each system component that condition the indoor environments [7]. Cooling and heating load calculations can significantly affect building construction cost, comfort and productivity of occupants, and operating cost and energy consumption [7].

Heating and cooling loads are the rates of energy input (heating) or removal (cooling) required to maintain an indoor environment at a desired temperature and humidity condition [7]. Heating and air conditioning systems are designed, sized, and controlled to accomplish that energy transfer [7]. The amount of heating or cooling required at any time varies widely, depending on external and internal factors [7]. Heat loads result from many conduction, convection, and radiation heat transfer processes through the building materials and from internal sources [7]. Building components or contents that may affect heat loads include the following:

- External: Walls, roofs, windows, skylights, doors, partitions, ceilings,
- and floors.
- Internal: Lights, people, appliances, and equipment.
- Infiltration: Air leakage and moisture migration.

This chapter’s focus, seek to determine the peak heat loads for the Production Hall of this case study.

This chapter covers 3 topics. First, the representation of an example of a production hall in this case study. Mainly, the dimensions of the production hall, it’s location and the requested comfort conditions for the indoor space. Second, the heat load calculations. The heat loads where calculated using a computer tool. That computer tool is called Hourly analysis program (HAP), and it is explained in some detail how it works. The third section presents the results of the heat load calculations.

3.1. Data Assembly

One of the objectives of the work is to compare different dehumidification solutions to a specific production hall. The dimensions and the required indoor conditions are inspired by a past project of RACE.

3.1.1. Characterization of the production hall considered in the case study

The general layout of the production hall is illustrated in Figures 11 and 12, where the configuration of the building, its orientation and external shading from other building plans and specifications.

Figure 11 shows a layout of several buildings. The one that is intended to dehumidify is the Production Hall. As it is possible to verify, the Production Hall has a floor area of 190 m per 30 m. Giving it a total area of 5700 m^2.

On Figure 12 it's presented its average ceiling height of 13.2 m with a 28.9° of roof slope. Giving it a roof gross area of 6510 m^2.

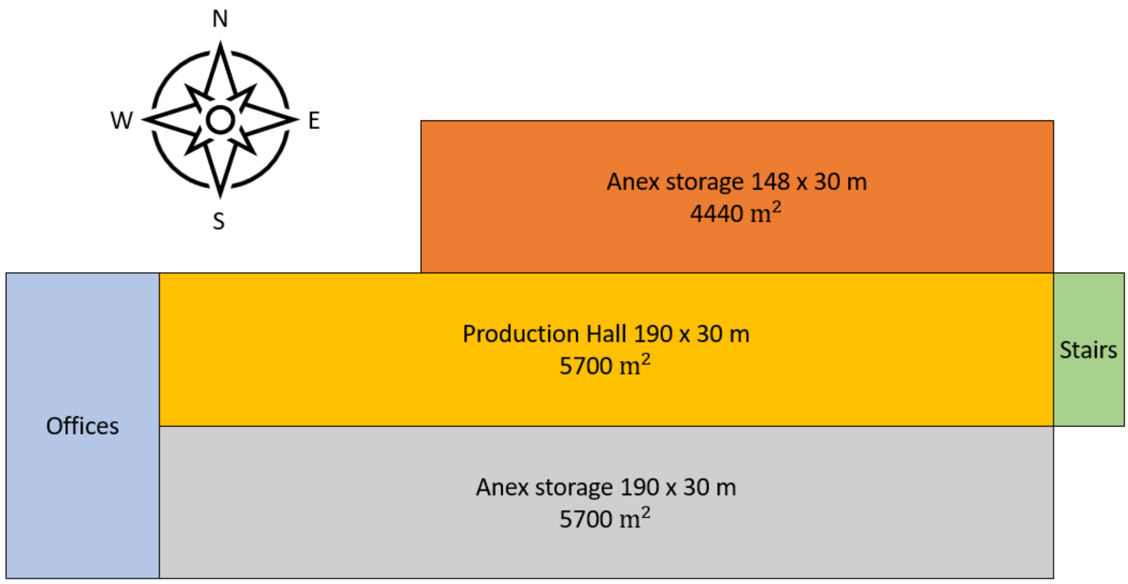

Figure 11– Illustration of the floor map area of the production hall and other buildings, presented as an example to the calculations of the heat loads [8].

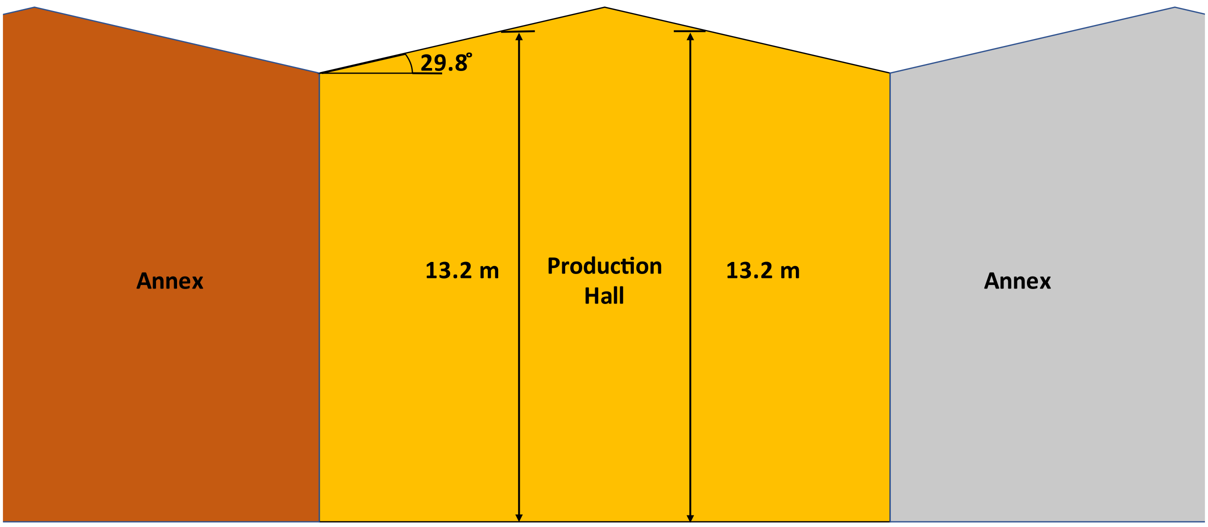

Figure 12 – Illustration of the production hall’s frontal view and its average ceiling height and the roof slope [8].

3.1.2. Location of the production hall and characterization of the outdoor and indoor conditions

Outdoor design conditions are important weather information for HVAC design purposes at a particular location. Data on climatic variables like air temperature, humidity, wind conditions and solar radiation. It is important information because it is the basis for heating, ventilating, and air-conditioning (HVAC) systems design and energy estimation for buildings [16].

Annex B shows the outdoor climatic design conditions for Porto, Portugal, which is the location of the production hall. The data was obtained from a weather station located in Porto. It is presented in a table and is available on the ASHRAE website, for major urban centers all over the world [7].

According to ASHRAE, the period of record used for the calculation of weather data presented in Annex B, is 25 years (1990 to 2014) [7]. The actual number of years used in the calculations for a given station depends on the amount of missing data and it may be as little as 8 years [7]. The first and last years of the period of record used to calculate design conditions are listed in the top section of the tables of climatic design conditions, as shown in Annex B.

Simple design conditions were obtained by pooling the hourly data into frequency tables and then deriving from the pooled data the design condition likely to be exceeded by a certain percentage of the time [7]. Mean coincident values were obtained by double-binning the hourly data into joint frequency matrices, then calculating the mean coincident value corresponding to the simple design condition [7].

The section to focus on, from Annex B, is the annual design conditions on the top of the table. Annual Heating and Humidification Design Conditions for the calculation of heating heat loads. The annual cooling, dehumidification, and enthalpy design conditions section is important for the calculation of cooling heat loads.

The most important information to extract from the annual heating and humidification design conditions section are as follows:

- Coldest month (1 = January and 12 = December). On this case the coldest month for Porto is January.
- Dry-bulb temperature, T_{db}, of 1.9 and 3.1 ºC corresponding to 99.6 and 99.0% annual cumulative frequency of occurrence (cold conditions), respectively.
- Dew-point temperature, T_{dp}, of -4.5 and -2.5 ºC, corresponding to 99.6 and 99.0% annual cumulative frequency of occurrence. Corresponding humidity ratio (*HR*), grams of moisture per kg of dry air and mean coincident dry bulb temperature (MCDB) in degrees Celsius.

The most important information to extract from the annual cooling, dehumidification, and enthalpy design conditions section are as follows:

- Hottest month for Porto is August.
- Daily temperature range for hottest month, is 9 ºC [defined as mean of the difference between daily maximum and daily minimum dry bulb temperatures for hottest month].
- Dry-bulb temperature of 30.3 °C, 28.1 °C and 26 °C corresponding to 0.4, 1.0, and 2.0% annual cumulative frequency of occurrence (cooling section conditions), respectively. Also, a mean coincident wet-bulb temperature, of 19.3 °C, 18.7 °C and 18 °C, respectively.
- Dew-point temperature, T_{dp}, of 19.1 °C, 18.3 °C and 17.9 °C, corresponding to 0.4, 1.0, and 2.0% annual cumulative frequency of occurrence, respectively. Corresponding humidity ratio and mean coincident dry-bulb temperature.

With this information it is possible to define all outside conditions require to determine a viable dehumidification solution.

On the other hand, indoor conditions are defined entirely by the client. In this example, the client wants a range of temperature of 17 to 25 ºC and a relative humidity of 10 to 65 %. In Figure 13 is illustrated the comfort zone, delimited with black lines, on a psychometric chart.

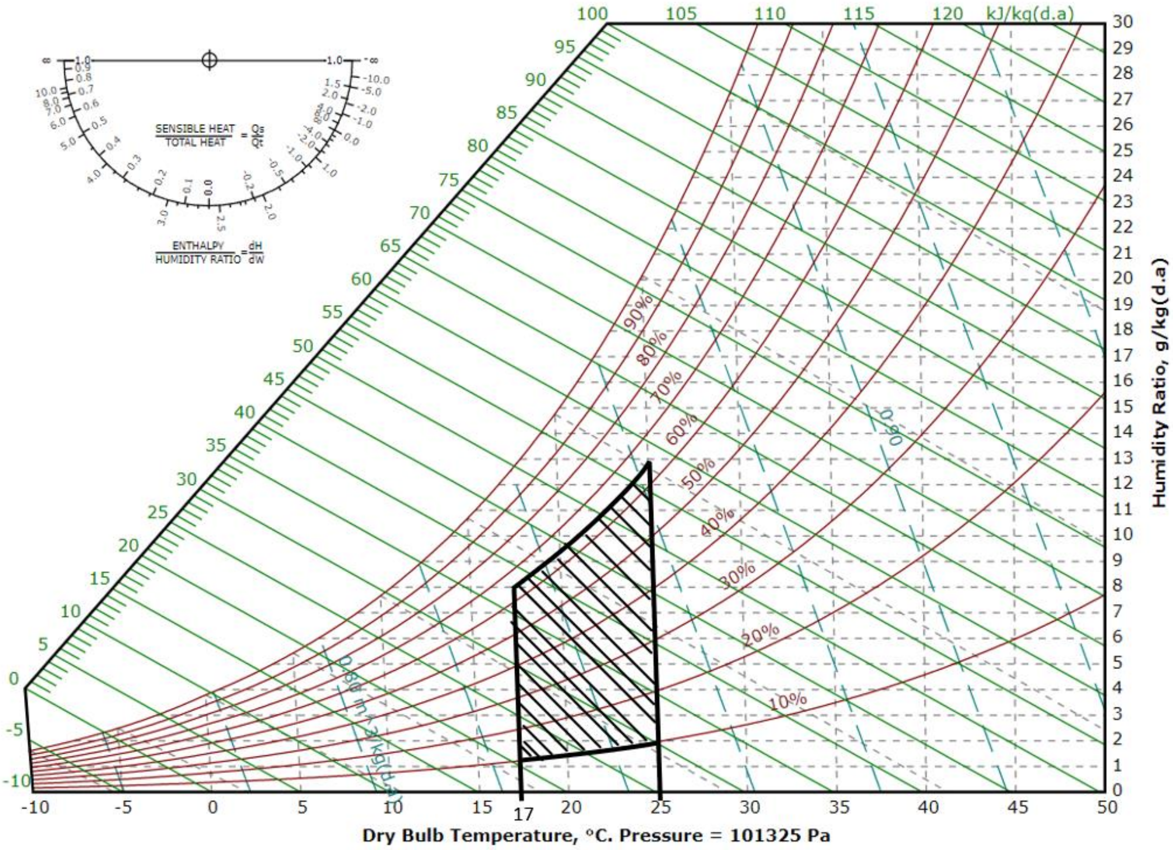

Figure 13– Illustration of the design indoor conditions provided by the client, using apsychometric chart.

3.1.3. Internal heat gains

As part of the heat loads calculations is it essential to obtain information about the occupation density and a schedule of lighting, occupancy schedule, equipment and appliances, and any other processes that contribute to the internal thermal load.

In Table 2 is demonstrated the schedule of hourly and daily behaviour in percentages, of the equipment usage. As well as the Table 3 for the lighting usage. Table 4 for occupation and Table 5 for production. However, for production, it will be presented as occupied or unoccupied.

Table 2 - Schedule of hourly and daily behaviour in percentages, of the equipment usage.

Hour	Percentage of usage [%]	Hour	Percentage of usage [%]
0	0	13	100
1	0	14	100
2	0	15	100
3	0	16	100
4	0	17	100
5	0	18	0
6	0	19	0
7	0	20	0
8	30	21	0
9	50	22	0
10	100	23	0
11	100		
12	50		

Table 3 - Schedule of hourly and daily behaviour in percentages, of the lighting usage.

Hour	Percentage of usage [%]	Hour	Percentage of usage [%]
0	0	13	100
1	0	14	100
2	0	15	100
3	0	16	100
4	0	17	100
5	0	18	100
6	50	19	0
7	100	20	0
8	100	21	0
9	100	22	0
10	100	23	0
11	100		
12	100		

Table 4 - Schedule of hourly and daily behaviour in percentages, of occupation.

Hour	Percentage of usage [%]	Hour	Percentage of usage [%]
0	0	13	100
1	0	14	100
2	0	15	100
3	0	16	100
4	0	17	100
5	0	18	0
6	0	19	0
7	0	20	0
8	30	21	0
9	50	22	0
10	100	23	0
11	100		
12	50		

Table 5 - Schedule of hourly and daily behaviour in terms of occupation, of the production.

Hour	Occupied	Hour	Occupied
0	✖	13	✔
1	✖	14	✔
2	✖	15	✔
3	✖	16	✔
4	✖	17	✔
5	✖	18	✔
6	✖	19	✔
7	✖	20	✖
8	✔	21	✖
9	✔	22	✖
10	✔	23	✖
11	✔		
12	✔		

3.2. Hourly analysis program (HAP) procedure

This section is dedicated to the demonstration of the program HAP [9]. It is one of the key elements on this thesis. It is very important that the calculations of heat loads be as accurate as possible.

In this subsection, it is demonstrated on what the program consists of and how it works. Every step is explained, and the solutions are also interpreted. It is important to mention that this subsection only concentrates on the calculation of heat loads. The selection and study of the dehumidifiers is indicated on chapter 4.

3.2.1. Introduction in HAP

Hourly analyses program (HAP) is a computer tool which helps designing HVAC systems for any type of building. HAP is useful for estimating thermal loads and designing HVAC systems. It is also helpful in simulating energy consumption and calculating energy costs. It is a program that is ASHRAE-endorsed for load calculations that uses the transfer function load method and detailed energy simulation techniques for the energy analysis [9].

For better understanding the principles of thermal calculation method applied in this program, it is presented in the Annex A a full description, from the program itself, about the Transfer function load method.

3.2.2. Calculation procedure

The procedure for the heat load calculation on HAP is laid out in this section. To create simulations and analyses on the program, there are some aspects in the program that need to be characterized. They are as follows:

- **Weather:** Definition of the temperature, humidity, and solar radiation conditions the building is subjected to, during a year. The user can choose a city directly from the program's weather database, or weather parameters can be manually inserted using the weather input form [9].
- **Space:** A space is a region of the building constituted of one or more heat flow elements and served by one or more air distribution terminals. Usually, a space represents a single room. In this case study, the space in question is the entire production hall. To define a space, all elements which affect heat flow in the space

must be described. Elements include walls, windows, doors, roofs, skylights, floors, occupants, lighting, electrical equipment, miscellaneous heat sources, infiltration, and partitions. [9]

- **Systems:** In this field components the user can define the equipment and controls that provide cooling and heating to a region of a building. There are several examples of air systems that the user can choose from on the air systems input form. [9]

Figure 14 shows a diagram of the modelling process to create a HVAC project in HAP. After cretaing a new project, the first step is to define the parameters on the weather window.

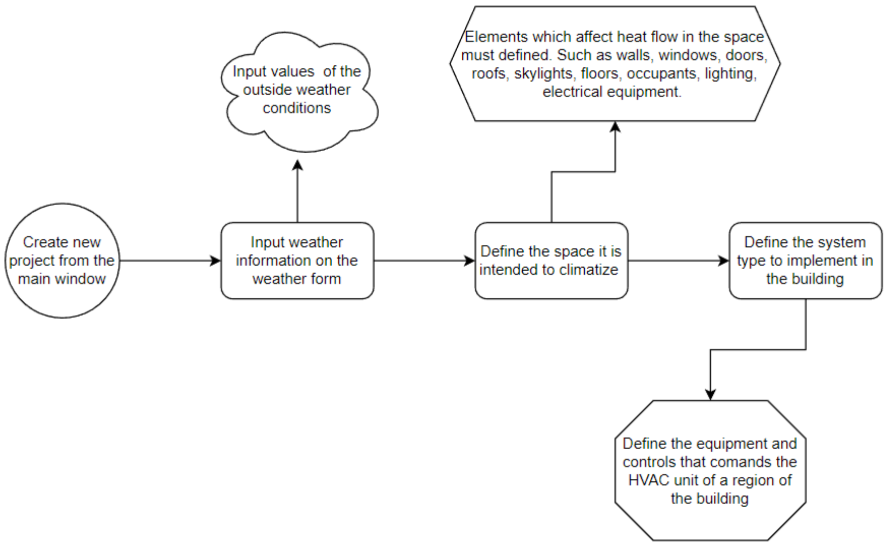

Figure 14 - Diagram of the HAP modelling process to create a project.

In Table 6 is demonstrated the values to input in the program for the outside weather in Porto.

Table 6 - Weather data inputted in HAP for the design weather conditions for the building site.

Location	Portugal	**Summer DB**	21.1 ºC
City	Porto	**Summer coincident WB**	19.9 ºC
Latitude	41.2º	**Summer Daily range**	9 K
Longitude	8.7º	**Winter DB**	1.9 ºC
Elevation	77 m	**Winter coincident DB**	-1.4 ºC

After defining the weather data, it is necessary to specify certain information about the walls, roof, and windows of the Production Hall. That type of information is important to calculate the heat loads. In Table 7 is illustrated the data of the materials used for the constructions of the walls and roof. Each row of the Table contains data for a separate layer in the walls and roof. In Table 8 is presented the information about the windows. All the data that is shown in those Tables are client given information.

For this case study, the value that the author defined was the outside surface colour of the walls and roof as light, with an absorptivity of 0.450, because in Porto can be considered low air pollution. That means that the surfaces which are not expected to darken through weathering [9]. It is recommended using the dark surface colour for exterior wall surfaces in urban areas where air pollution is more prevalent or to wall surfaces which are expected to darken due to climate conditions [9].

Table 7 – Wall and Roof information, used to define the wall and roof assembly, inputted in HAP

	Layers: Inside to outside	**Thickness [mm]**	**Density [kg/m^3]**	**Specific Heat [kJ/(kg·K)]**	**R value [(m^2·K)/W]**	**Weight [kg/m^2]**	Overall U value [W/(m^2·K)]	Absorptivity
Wall properties	Inside Surface resistance	0	0	0	0.12	0	0.38	0.45
	25 mm stucco	25.4	1858.1	0.84	0.04	47.2		
	203 mm HW concrete block	200	977.1	0.84	0.19	195.4		
	RSI-2.3 batt insulation	101.6	8	0.84	2.26	0.8		
	Outside surface resistance	0	0	0	0.06	0		
	Totals	327	-	-	2.66	243.4		
Roof properties	Inside Surface resistance	0	0	0	0.12	0	0.23	0.45
	RSI-1.2 board insulation	30	1858.1	0.84	0.04	47.2		
	RSI-2.5 board insulation	60	977.1	0.84	0.19	195.4		
	Outside surface resistance	0	0	0	0.06	0		
	Totals	327	-	-	2.66	243.4		

Subsequently, Table 8 contains information describing the window's assembly [9]. It is possible to insert the overall U value and shade coefficient of the window.

Table 8 - Window data, inputted in HAP, that defines one window assembly.

Window Properties	
Height	5.8 m
Width	6.8 m
U	2.1 W/(m^2·K)
Shade coefficient	0.37

The next step is to define the space properties. In this section of the program, it is required to the user to define all the building characteristics and internal design conditions.

The Space data, shown in the Figures 25 to 29, is used to define the characteristics of all heat flow elements in a space. The Space Form from HAP contains the following essential components:

- **General**: It is presented in Table 9 and contains the space name, total floor area, average floor to ceiling height and the building weight as well the ventilation requirements for the space.

Table 9 - General information about the building space that defines several key values which describe the space as a whole and the ventilation requirements for the space.

General properties	Floor area	5700	$\mathbf{m^2}$
	Average ceiling height	13.2	$\mathbf{m^2}$
	Building weight	341.8	kg/m^2
	Outside air ventilation requirements		
	Requirement 1	5	L/s/person
	Requirement 2	0.3	$L/(s{\cdot}m^2)$

As it is possible to visualize in Table 9, the "Floor Area" and "Avg Ceiling Height" are data about the building itself. The section "Building Weight" consists of the overall weight of the interior walls, floors, ceilings, and contents of the building. The value of this component influences how heat gains are converted into loads. Heavy buildings tend to absorb and store heat for longer periods than light buildings. As a result, there is more of a lag between the time a heat gain occurs and the time it becomes an air-conditioning load [9]. As a standard procedure for all buildings in Portugal the building weight was set by the author as medium, which corresponds to 341.8 kg/m^2.

The OA Ventilation Requirements section contains two values which together define the outdoor ventilation requirements for the space: "OA Requirement 1" and "OA Requirement 2". All two of these inputs comply with the ANSI/ASHRAE Standard 62.1-2013 edition for minimum ventilation rates in a breathing zone of a Production Hall.

- **Internals**: It is presented in Table 10 and contains information about internal heat gains from lighting, electrical equipment, and occupants. The information in Table 10 is all client given information

Table 10 – Information about internal heat gains from lighting, electrical equipment and occupants.

Internals	**Lighting**		
	Fixture type	Recessed, unvented	
	wattage	8	W/m^2
	Eletrical equipment		
	Wattage	415,5	kW
	People		
	Occupancy	70	m^2/person
	Activity level	Medium work	
	Sensible	86,5	W/person
	Latent	133,3	W/person

As shown in the Table 10, data is divided into three groups, each dealing with a different type of internal heat gain. “Lighting” section describes the characteristics of overhead lighting fixtures. The values on this table are all clients given information. The schedule used is the one presented in Table 3. The schedule for “Electrical Equipment” is presented in Table 2 and the schedule for the “People” section is presented in Table 4.

- **Walls, Windows, Doors:** It’s presented in Table 11 and contains data for vertical wall exposures. In this case study, the writer considers that the Production Hall has no windows or doors exposed to solar loads.

Table 11 - Information about vertical wall exposures.

Walls	**Exposed Wall gross area**		
	North	139	m^2
	North	1223	m^2
	West	240	m^2
	South	2688	m^2
	South	462	m^2
	East	210	m^2

- **Roofs, Skylights**: It's presented in Table 12 and contains information about horizontal or sloped roofs and any skylights which are part of these roof exposures.

Table 12 - Information about horizontal and sloped roof exposures and the skylights that are exposed to solar loads.

Roofs/Skylights	**Exposed Roof/Skylight gross area**		
	Horizontal	6510	m^2
	Skylight quantity	28	

- **Infiltration:** It's presented in Table 13 and contains infiltration specifications.

Table 13 - Information about infiltration airflow in the Production Hall.

Infiltration	ACH	0,6	
	Gross airflow	12567,9	L/s

Infiltration results from the leakage of outdoor air into the space. This typically occurs because of leakage around windows and doors, and leakage from the opening and closing of doors in the space [9].

The central portion of the infiltration tab contains a table of infiltration specifications. Rows in the table contain infiltration data for design cooling (only affect sizing of cooling equipment), design heating (only affect sizing of heating equipment) and energy analysis conditions (these values only affect the calculation of annual energy use and operating cost) [9].

Columns in the table contain infiltration specifications in three different units of measure. When you enter an infiltration rate in one column, values for the other two columns in the row are calculated automatically [9]. Items in the L/s column define the infiltration rate as a gross airflow [9]. The L/s/sqm column define infiltration in terms of airflow per unit of exterior wall area [9]. The ACH column define infiltration in terms of air changes per hour [9]. When infiltration is entered as ACH, the program uses this ACH value plus the space volume to calculate the total infiltration airflow [9]. Space volume is derived from the total space floor area and the average floor to ceiling height defined on the General data tab [9]. The ACH value is client given information and it’s the same for all design calculations.

The infiltration tab contains one additional input describing how infiltration occurs. That is the “Infiltration occurs” section. It was selected the “Only When Fan Off” option because it can be assumed that infiltration will only occur during the "unoccupied" hours when the system is off.

- **Floors:** It’s presented in Table 14 and contains information about heat transfer through slab floors, basement floors or floors above an unconditioned region.

Table 14 - Information about heat transfer through floors.

Floors	Floor type	Slab-On-Grade Floor	
	Area	5700	m^2
	Total U value	0,568	$W/(m^2 \cdot K)$
	Exposed perimeter	273	m

In this case study, the author selected the "Slab-On-Grade Floor" type, which means that heat transfer is through the floor to the soil below [9]. In other words, the Production Hall is directly on top of the soil and not above basements of other conditioned or unconditioned spaces.

Below the floor type, contains data describing the heat transmission characteristics of the chosen floor type [9] and its values are obtained from the building floor measurements.

After filling the Space form with the necessary data, it is then possible to attend the System Form. Firstly, the program will direct the user to the General tab of the System Form presented in Figure 15.

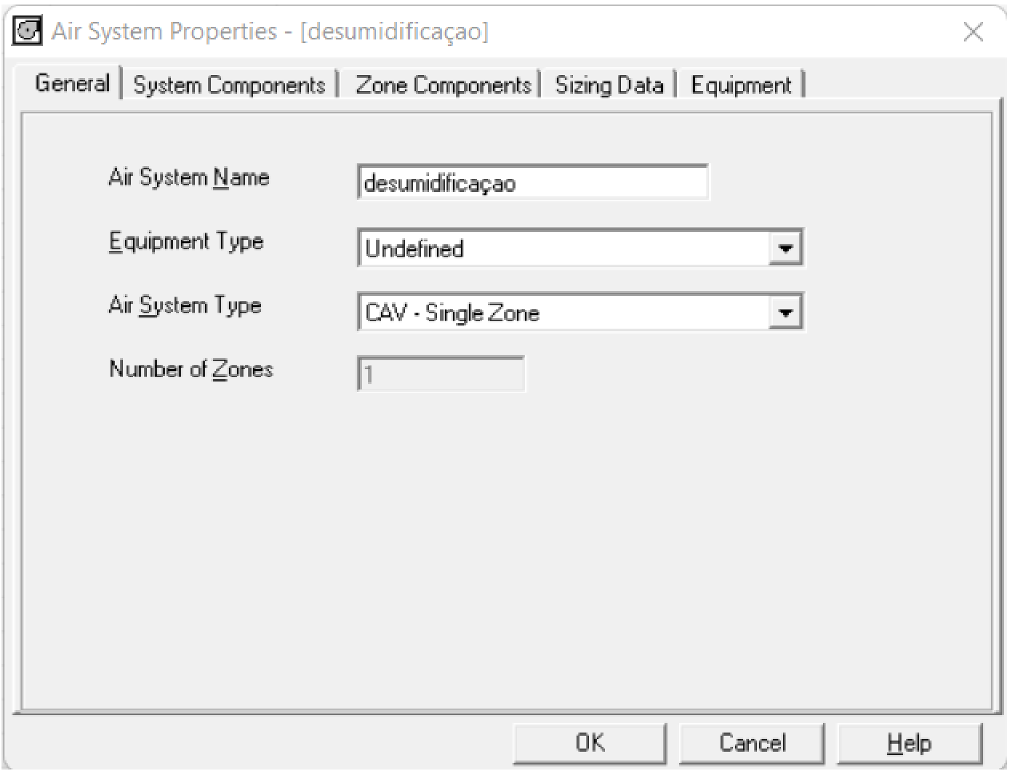

Figure 15 - General tab of the System form that contains several values which describe the nature of the system. Contains the system name, its equipment classification and system type, and the number of zones [9].

Two important fields of data that are presented in this tab are the "Equipment Type" and "Air System Type". The first one defines the equipment classification for the HVAC system that's being considered. It's possible to choose from several equipment classifications, but the author chose the "Undefined" classification. This option allows the user to circumvent the choice of an equipment type. It is often used for preliminary block load estimates in which equipment type is not yet relevant [9].

The second one, defines the kind of air system being used. The choice of a system type will influence system sizing calculations, such as operating cost studies, the system operation and energy consumption levels during the whole year energy simulation the program performs [9]. It is an important detail; however, it does not influence the focus of this thesis. For someone that intends to build an entire plant, it is required to specify the systems it wishes to implement for costs analyses. In this dissertation, the objective is another. For that reason, the author selected a constant air volume system for a single zone ("CAV – single zone"), presented in Figure 16. It's the most elemental structure that the program offers. CAV, uses a central air handler to provide a constant volume of conditioned air to the zone

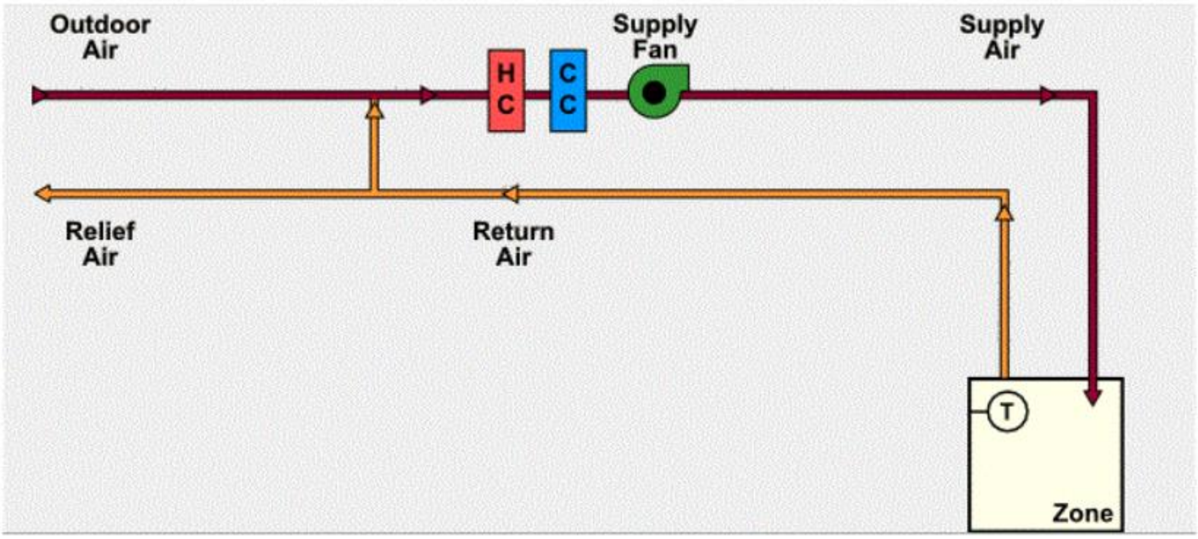

Figure 16 - CAV single zone air system [9].

The rest of the tabs of the System form are not relevant on this research except for one. The “System Component” tab, presented in Figure 17. Yet not all fields all important. This contains information about components in the system such as fans and coils, and information about the distribution duct system

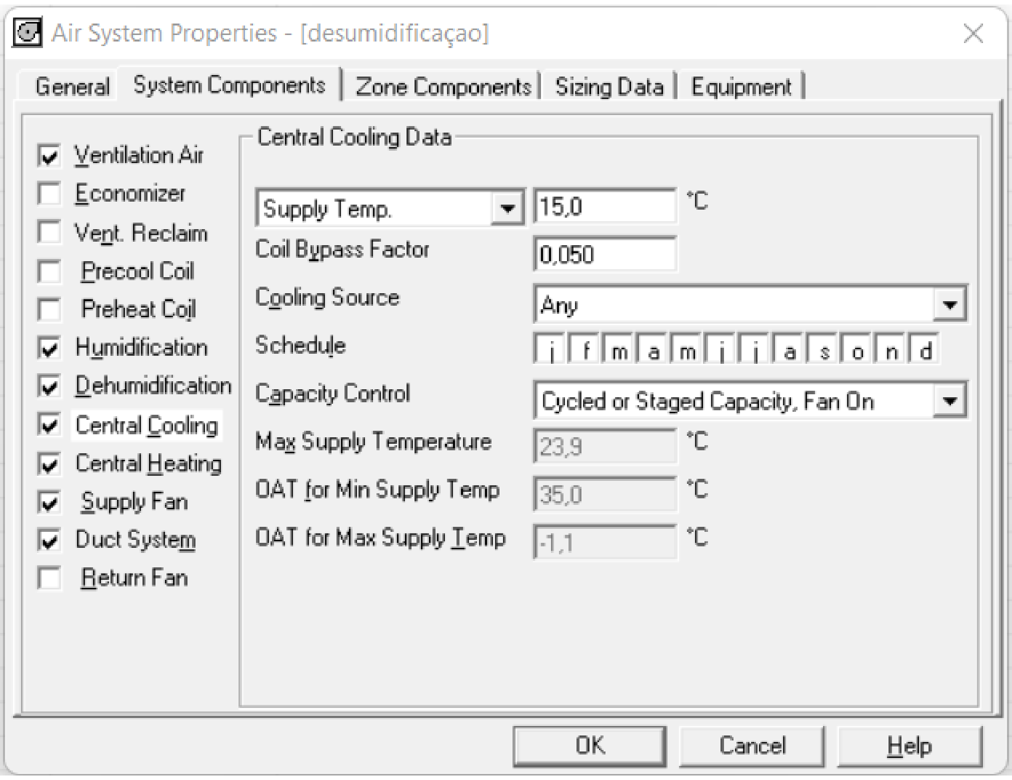

Figure 17 – The System Components Tab Form [9].

The most important item is the “Central Cooling Data” section. In this section it can be defined the supply temperature to the room and the coil bypass factor. For the supply temperature, the writer specified a temperature of 15 ºC. The reason why, is that is good practice to give a 10 ºC difference between the desired room temperature and the outlet air temperature of the ducts. As for the coil bypass factor, it will be explained in full detail in Chapter 4. It’s an important feature of the cooling coils that greatly influences dehumidification, but for heat loads calculations is not relevant.

3.2.3. Heat Loads results

After all the data collection the next step is to generate a diagnosis report on the program. The content on Table 15, presents the heat loads results.

Table 15 – Heat loads results summary generated by HAP [9].

Zone loads	Details	Sensible (kW)	Latent (kW)
Window & Skylight Solar Loads	1104 m^2	158	-
Wall Transmission	4962 m^2	-8.31	-
Roof Transmission	5406 m^2	10.8	-
Skylight Transmission	1104 m^2	-10.9	-
Floor Transmission	5700 m^2	0	-
Heat loss to annexs	1332 m^2	-6	-
Lighting	46 kW	36.7	-
Electric Equipment	42 kW	367	-
People	81	4.76	10.9
Total Zone Loads		560	10.9

In the most humid month, August, peak conditions, the sensible heat load is approximately 560 kW, and the latent heat load is 10.9 kW.

4. Characterization of the dehumidifiers

In this chapter the author distinguishes the two types of dehumidifiers selected for this study. The desiccant dehumidifier and the cooling coil dehumidifier. Both are strong candidates for a successful climatization control.

In this chapter it is presented the calculations necessary to arrive to the best option on each case. On Chapter 5 is demonstrated a simple costs analysis, to decide on the best apparatus.

This chapter is divided into two sections. First section, a presentation of the calculations made by HAP for the cooling coil performance on air conditioning, and some theory for this apparatus. Secondly, desiccant dehumidifiers calculations made by hand.

4.1. Cooling coil dehumidifier

Before presenting the calculations for the selection of a cooling coil for dehumidification purposes, it is important to understand what influences the results in this apparatus. For a better understanding of what happens in a cooling coil in the middle of an air conditioning process, the following subsection describes the characteristic psychometric performance of this equipment.

After that theoretical rundown, another subsection presents the results generated by HAP demonstrating the best working conditions for the cooling coil.

4.1.1. Coil characteristics

As it is presented in Chapter 2, in the normal operation of coils, the air is forced over the surface of the coil and passes over a series of tubes. Those tubes contain a refrigerant fluid passing through them at a lower temperature than the air (in the case of cooling and dehumidification).

The amount of coil surface not only affects the heat transfer but also the bypass factor of the coil [11]. The bypass factor represents that portion of air which passes through the cooling coil completely unaltered [11]. So, the higher the bypass factor the higher amount of air remains thermodynamically unchanged. In fact, an increase of the bypass factor, which means less rows of coil and less surface area, wider spacing of coiling tubes, results in a deficit in heat transfer, because there is less available apparatus surface for the heat transfer. Also, a larger bypass factor means that the refrigerant temperature needs to be lower to produce the desire affect. To achieve the appropriate indoor conditions more air must be supplied. With that in mind, the system needs to have a larger fan and fan motor [11].

A decrease of the bypass factor means that the coil temperature can be higher. There is less air being thermally unaltered. So, it is possible to have a system where the fan size and motor can be smaller, because less air is required to produce the desired indoor conditions. the heat transfer is increased due to the bigger coil surface (more coil rows) [11].

However, having a higher bypass factor is not enterally bad. It depends on the application of the apparatus. When a low bypass factor coil is implemented, one of the solutions that can be implemented, to reduce the bypass effect, is to slow the air speed through the apparatus [11], so that more air can contact the surface rows of the battery. In fact, it is more appropriately used when it needs to win over high latent loads. In Table 16, it is presented various bypass factor values and where is typically applicable.

Table 16 – Standard bypass factors for several applications [11].

COIL BYPASS FACTOR	TYPE OF APPLICATION	EXAMPLE
0.30 to 0.50	A *small* total load or a load that is somewhat larger with a low sensible heat factor (high latent load).	Residence
0.20 to 0.30	Typical comfort application with a *relatively small* total load or a low sensible heat factor with a somewhat larger load.	Residence, Small Retail Shop, Factory
0.10 to 0.20	Typical comfort application.	Dept. Store, Bank, Factory
0.05 to 0.10	Applications with high internal sensible loads or requiring a large amount of outdoor air for ventilation.	Dept. Store, Restaurant, Factory
0 to 0.10	All outdoor air applications.	Hospital Operating Room, Factory

From Table 16, it is possible to conclude that for this case study the cooling coil BF needs to be between 0.05 and 0.1. As the heat loads results show, there is a high value of sensible load (560 kW) in comparison to the latent load (10.9 kW), and a large amount of outdoor air for ventilation is surely necessary to insufflate. Not only because of the dimensions of the Production Hall but also to win over those heat loads.

For a better understanding of the apparatus behaviour, there are some psychometric contents that can be introduced. Those contents are the coil process line, effective surface temperature (EST) and the contact factor (CF).

In figure 18 it is illustrated a line depicted on a psychometric chart which represents the changes in the air properties as it passes through the cooling and dehumidifying coil. It is labelled as the coil process line. The inclination and direction of this line depends on the coil configuration, air velocity, and refrigerant temperature [12]. In truth it is an arched line, however, it is possible to define a straight line on the chart that, although it is not the true coil process line, can be called the coil process line anyway, because it also enables to select a coil or check the performance [12].

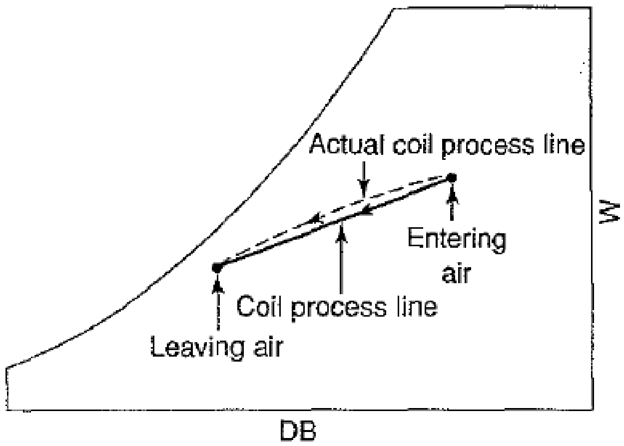

Figure 18 - Coil process line [12].

In other words, the coil process line definition is straight line drawn between the air conditions entering and leaving the coil, as shown in Figure 18. Note that "DB" stands for Dry bulb temperature in ºC, and "W" stands for Humidity Ratio in kg_v/kg_{da}.

Another important content to know about cooling coils is the contact factor (CF). It is the opposite of the bypass factor (BF). As previously mentioned, the BF is the amount of air that passes through the cooling coil and does not touch the surface of the coil. It is therefore uncooled. The CF is the amount of air that touches the surface and is thus cooled [12]. In Figure

19, shows where the BF and CF are situated. Note that from the figure there are three new variables: *a*, *b*, and EST.

EST stands for effective surface temperature. It represents an average surface temperature of the coil to which the air that contacts the surface is cooled. It is also called the apparatus dew point [12]. For applications involving cooling and dehumidification, the effective surface temperature is the point where the coil process line crosses the saturation line on the psychometric chart [11], as it is shown in Figure 19. As such, this effective surface temperature is also considered the apparatus dew point [11].

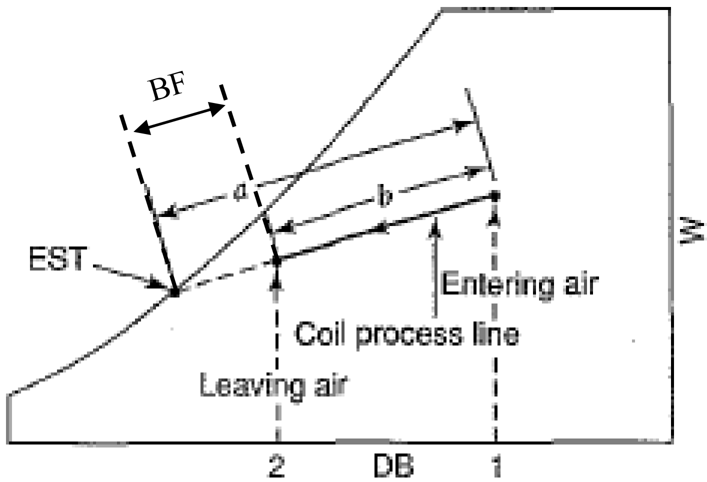

Figure 19 – Representation of a coil process in a psychometric chart [12].

Variables *a* and *b*, represent equations that relate the BF with the CF, as such:

$$a = CF + BF \tag{5}$$

$$b = CF \tag{6}$$

where:

$$a = 1$$

Is possible to conclude from Figure 19, that as the inclination of the coil process line increases, so does the BF, and the EST lowers consequently. As previously mentioned, if the bypass factor increases the coil surface temperature as to decrease so it can be able to achieve the desired conditions. A problem with an EST very low is the occurrence of surface freezing.

In some cases, freezing occurs when a dehumidification coil's surface temperature falls below 0 ºC, but the exact value depends on the design of the coil, its operating dew point, and the amount of cooling total load. The possibility of this type of surface freezing is greater if a bypass is used because it causes less air to be passed through the coil [13].

4.1.2. Cooling coil data

In this subsection, a series of results generated by HAP are presented, to create performance graphs for a certain cooling coil. After all the inputs in the program demonstrated in chapter 3, there is one value that is the key value for the selection of an apparatus. That is the bypass factor. In Figure 17 is demonstrated an input field named "Coil Bypass factor" where different BF values where selected.

Results generated from HAP simulations are presented in Table 17 for different BF values, for a constant room supply temperature of 15 ºC.

Table 17 – Results generated from HAP reports.

Total coil load [kW]	**Sensible coil load [kW]**	**Air quantity $[\mathbf{m^3/h}]$**	**Sensible heat ratio**	**Coil EST [ºC]**	**Resulting ϕ [%]**	**Bypass factor**
536.1	508.2	190.5×10^3	0.948	15.9	59	0.05
536.5	508.2	190.5×10^3	0.947	15.8	59	0.06
536.8	508.2	190.5×10^3	0.947	15.7	59	0.07
537.2	508.2	190.5×10^3	0.946	15.6	58	0.08
537.7	508.2	190.5×10^3	0.945	15.5	58	0.09
538.1	508.2	190.5×10^3	0.944	15.4	57	0.1

To better understand the contents on Table 17 the Figures 20 to 23 illustrate the results. All data correspondent to those figures are in line with the theory presented in section 4.1.1.. Figure 20 demonstrates that, as the bypass factor increases so does the coil load. That happens because since there is more air passing unchanged through the battery, the EST needs to be lower to give the same performance, as it is observable from Figure 22. So that means that the refrigeration cycle needs a higher load to correspond that temperature drop.

It is important to notice that the total coil load values generated, represent the peak load. Throughout the month of August, the cooling coil doesn't constantly operate under such load and that will be considered on the costs analyze. Also, those total coil loads don't match the thermal heat loads presented in Chapter 3. That's because HAP also considers the influence of ventilation loads into the sensible and latent heat loads, therefore directly changing the necessary peak total coil loads for the different bypass factor values.

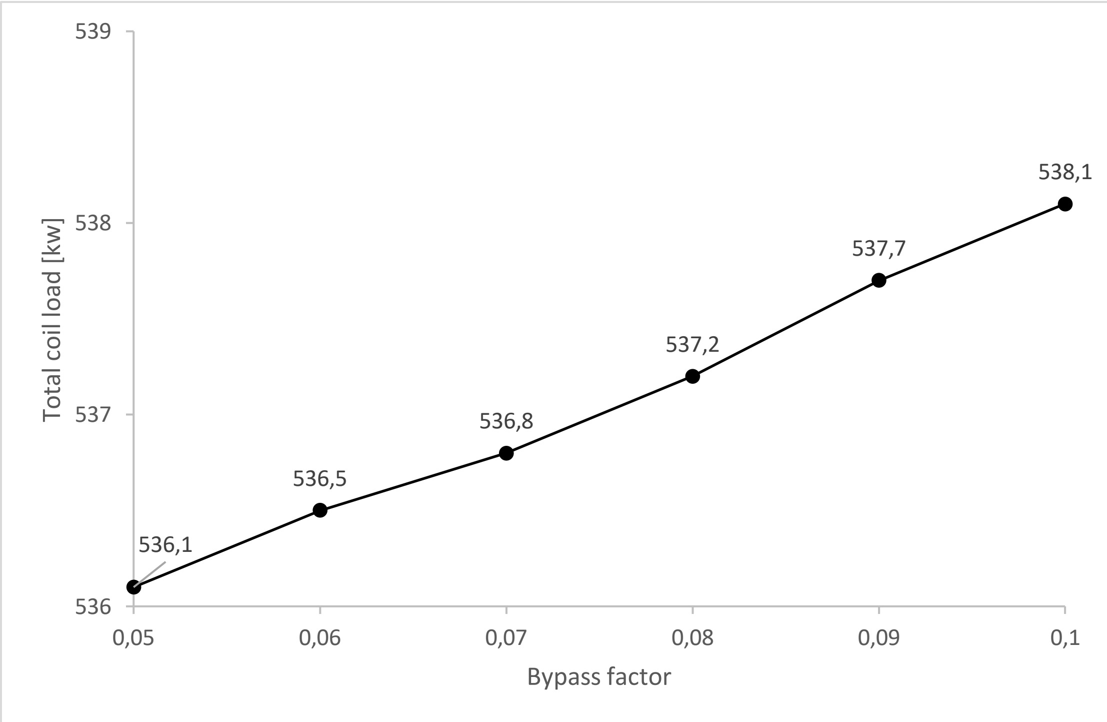

Figure 20 – Total coil load as function of the bypass factor.

Since that by increasing the bypass factor the EST decreases, that causes an increment of the inclination of the coil process line. In reaction to that factor, the latent heat load directly increases, as it is demonstrated in Figure 21. The sensible heat ratio relates the sensible heat load with the total heat load. The lower the ratio, the higher is the latent heat load that the coil needs to deal. Therefore, the higher the cooling coil load needs to be to win over that increment of latent heat.

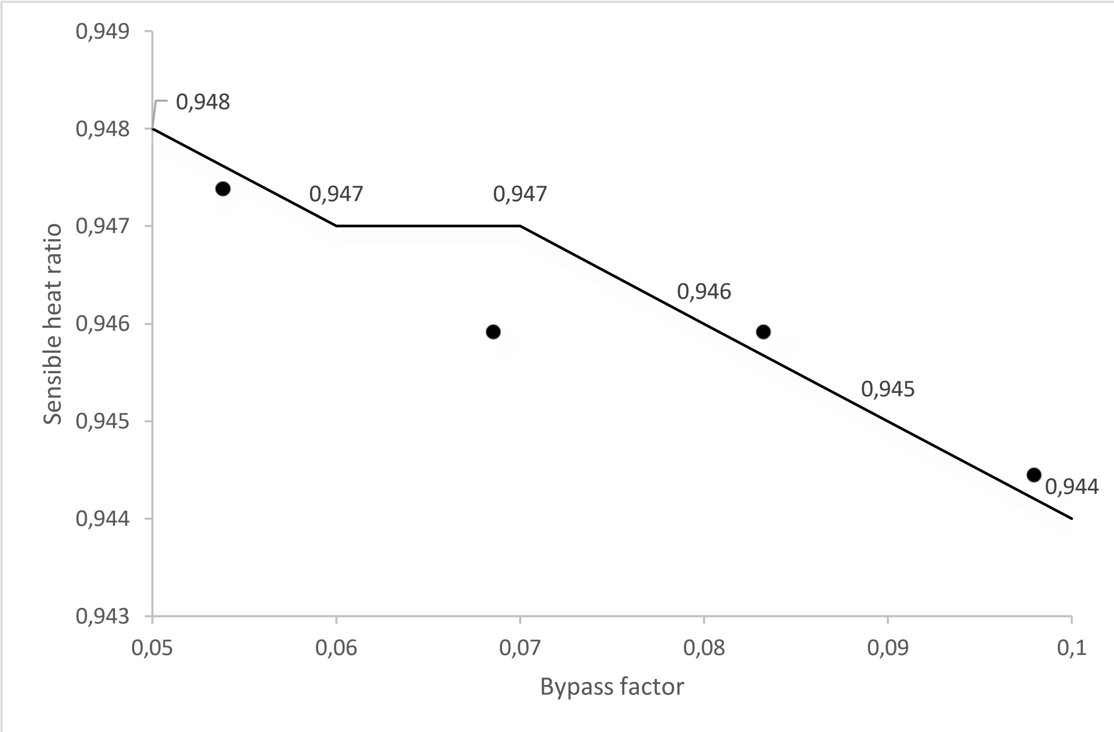

Figure 21 – Sensible heat ratio as function of the bypass factor.

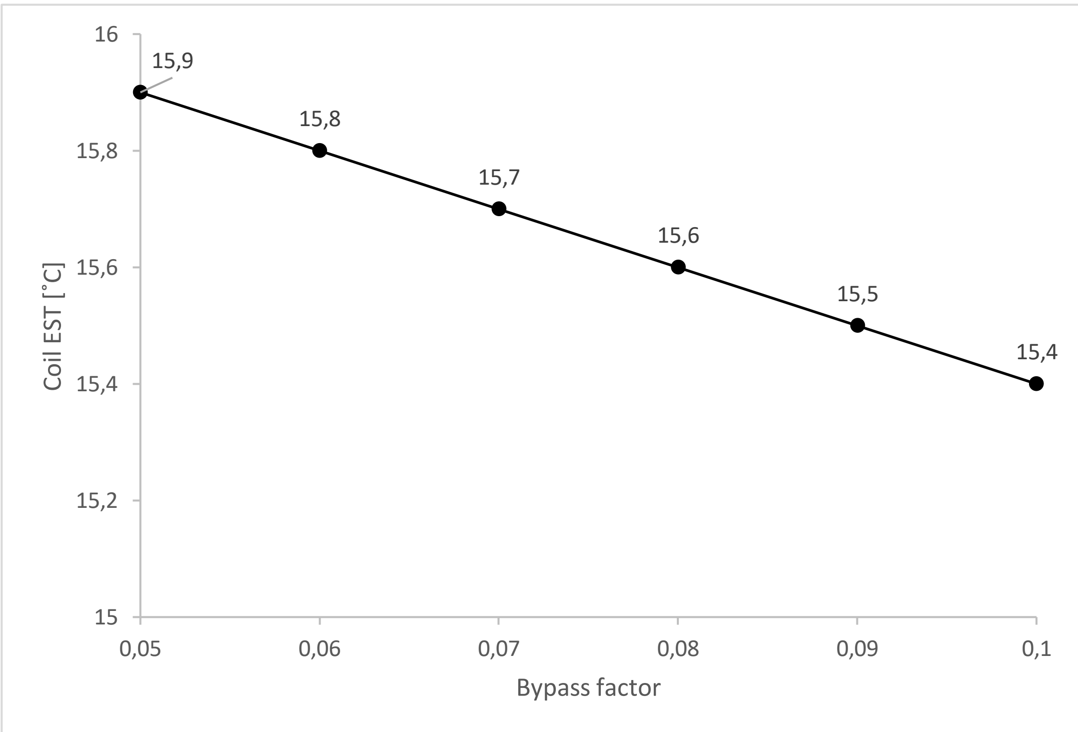

Figure 22 – Coil surface temperature as function of the bypass factor.

Another conclusion that is observable from Figure 23 is the resulted ϕ after the air passes through the apparatus. In fact, the higher the bypass factor, the lower the ϕ it can achieve. This information poses a contradiction. The higher the bypass factor the higher the load, but also the better it reduces the ϕ. So, the choice of a cooling coil is defined by a compromise between the cooling coil load that the author is willing to provide and the BF that is also willing to accept.

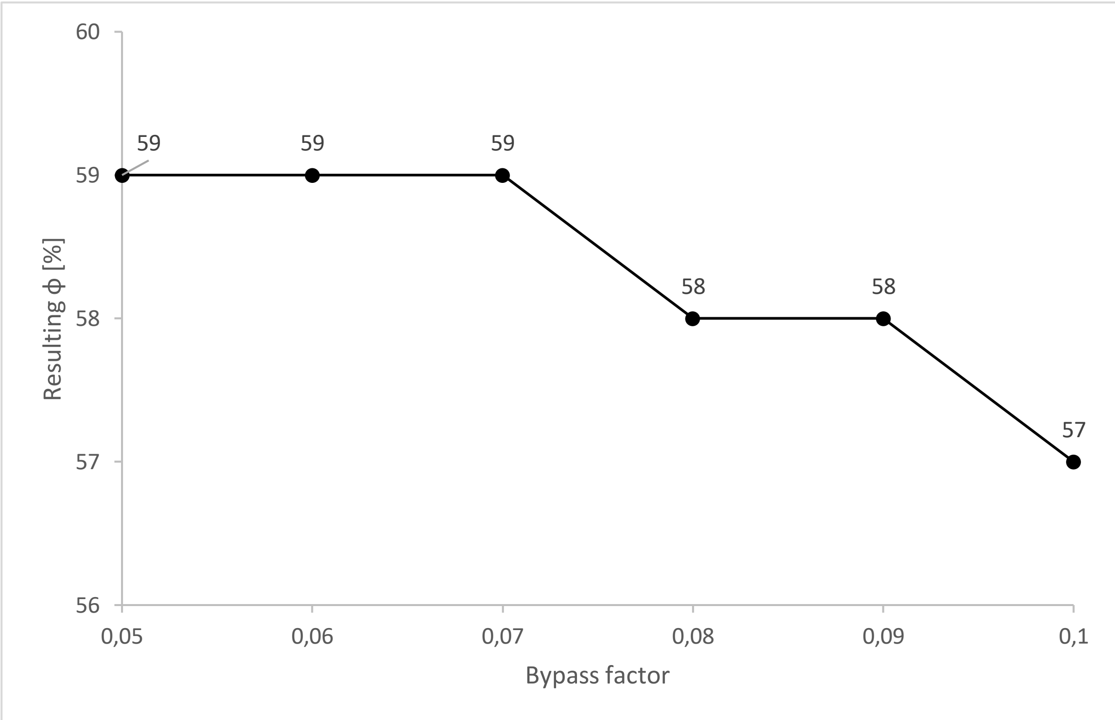

Figure 23 – Relation between the leaving humidity ratio and the increase of the bypass factor.

Given the results, the author decided that the best bypass factor, in this case study, is 0.1, because it provides the lower ϕ. Therefore, it minimizes the possibility that the apparatus does not dehumidify the air to below 65% of ϕ. So, for the investment analyse, the author will search the market for air units that provide a peak load of 538.1 kW to the cooling coil.

The psychometric path, generated by HAP, of the considered cooling coil is presented in Figure 24. As it is observable from the Figure, the contact of the mixed air (point 2 with a dry bulb temperature of 24.4 ºC) with the cooling coil surface (point 4 with a temperature of 15.4ºC) results on an insufflation air temperature of 16.3 ºC (point 3). Insufflating air at that temperature results on a room air temperature of 24.5 ºC. Note that point 2 is the mixture of the outside air (point 1 with a temperature of 21.1ºC) with the return air of the room.

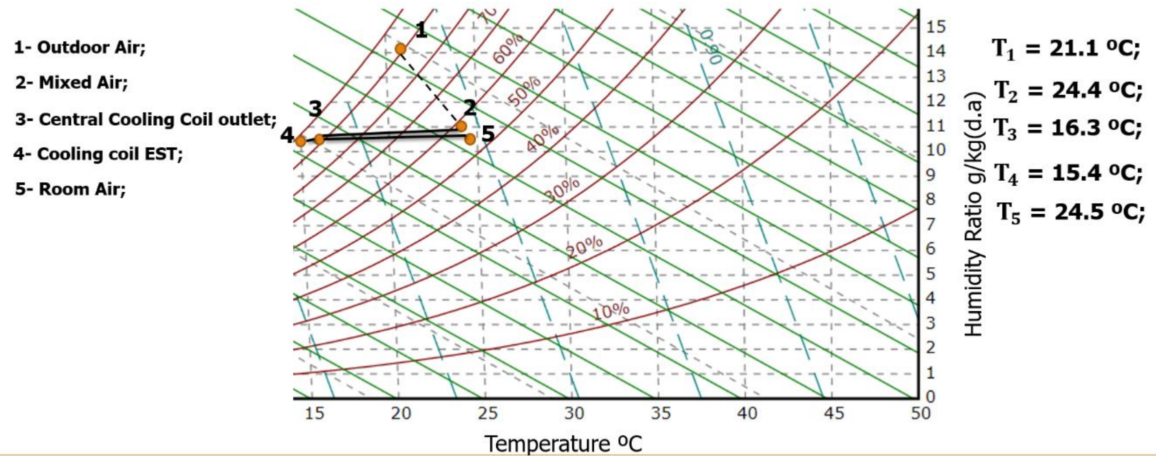

Figure 24 - Psychometric path of the cooling battery with a bypass factor of 0.1.

4.2. Desiccant dehumidifier

For the selection of a desiccant dehumidifier, the calculations are easier and direct. To size it, it is only necessary to know the dehumidification capacity in kg per hour. For that, it is required to firstly know the entry air properties and the desired exit humidity ratio, with the assistance of the psychometric chart. In Table 9 it is presented such properties. The properties identified with the number 1 are related with the entry, the others identified with the number 2 are related with the exit.

Table 18 – Entry and exit air properties passing through the desiccant dehumidifier.

Entry			**Exit**		
T_1	21.1	ºC	T_2	21.1	ºC
ϕ_1	89.7	%	ϕ_2	65	%
w_1	14.16	$^{\mathrm{g}}/_{\mathrm{kg(d.a)}}$	w_2	10.19	$^{\mathrm{g}}/_{\mathrm{kg(d.a)}}$
v_1	0.85	$^{\mathrm{m^3}}/_{\mathrm{kg(d.a)}}$	v_2	0.85	$^{\mathrm{m^3}}/_{\mathrm{kg(d.a)}}$

After defining the air properties, the next step is to begin calculations. It starts by calculating the difference between the entry humidity ratio and exit humidity ratio, Δw, as it is presented in equation 7, as well as the average between specific volumes, $\bar{v}$, as it is presented in equation 8.

The difference between humidity ratios, Δw, is defined as:

$$\Delta w = w_1 - w_2 \tag{7}$$

$$\Delta w = 3.97 \ \mathrm{g/kg(d.a)}$$

The average between specific volumes, $\bar{v}$, is defined as:

$$\bar{v} = \frac{v_1 + v_2}{2} \tag{8}$$

$$\bar{v} = 0.85 \ \mathrm{m^3/kg(d.a)}$$

After that, the flow volume that is necessary to ventilate the entire production hall, as it is presented in equation 9.

$$\dot{V} = V \times ACH \tag{9}$$

where:

$\dot{V}$ – Flow volume ($\mathrm{m^3/h}$)

V – Volume of the conditioned space ($\mathrm{m^3}$)

ACH – Air changes per hour ($1/\mathrm{h}$)

From equation 9, it results in:

$$\dot{V} = 75205.8 \times 0.6 \approx 46 \times 10^3 \ \mathrm{m^3/h}$$

After the flow volume is obtained, then the next step is to obtain the mass flow of the dry air, $\dot{m}_{d.a}$ in $\mathrm{kg_{d.a}/h}$. It is presented in equation 10.

$$\dot{m}_{d.a} = \dot{V} \div \bar{v} \tag{10}$$

where:

$$\dot{m}_{d.a} = 45123.48 \div 0.85 \approx 53 \times 10^3 \ \mathrm{kg_{d.a}/h}$$

Finally, it is possible to calculate the mass flow of the condensate, $\dot{m}_v$, in $\mathrm{kg_v/h}$. By multiplying the mass flow of the dry air with the difference between humidity ratios, as it is presented in equation 11.

$$\dot{m}_v = \dot{m}_{d.a} \times \Delta w \times 0.001 \tag{11}$$

From equation 11 we get:

$$\dot{m}_v = 53 \times 10^3 \times 3.97 \times 0.001 = 210 \ \mathrm{kg_v/h}$$

In conclusion, it is required a desiccant dehumidifier that is capable to condensate a mass flow of 210 $\mathrm{kg_v/h}$ and insufflates a flow volume of approximately $46 \times 10^3 \ \mathrm{m^3/h}$.

5. Basic cost analysis

Both dehumidifiers present the capacity to climatize the air to the intended indoor conditions. Although, each performs better under different insufflation temperatures, they can both be used as a solution to this case study. However, there is a variable in this problem that truly defines the appropriate solution. That is the cost of investment and its operation. The monetary factor is what separates the two and ensures the correct investment.

The HAP program can provide cooling load values for the cooling coil dehumidifier throughout the entire year. By generating an annual energy simulation, the results are presented in Figure 25.

To calculate the electrical consumption, it was simply dividing the cooling coil load to the Energy Efficiency Ratio (EER). The EER indicates the ratio of heating or cooling provided by a unit to the amount of electricity supplied to generate it. The equipment's manufacturer states that the EER of this equipment is 3.20. The results of the monthly electrical consumption are also presented in Figure 25.

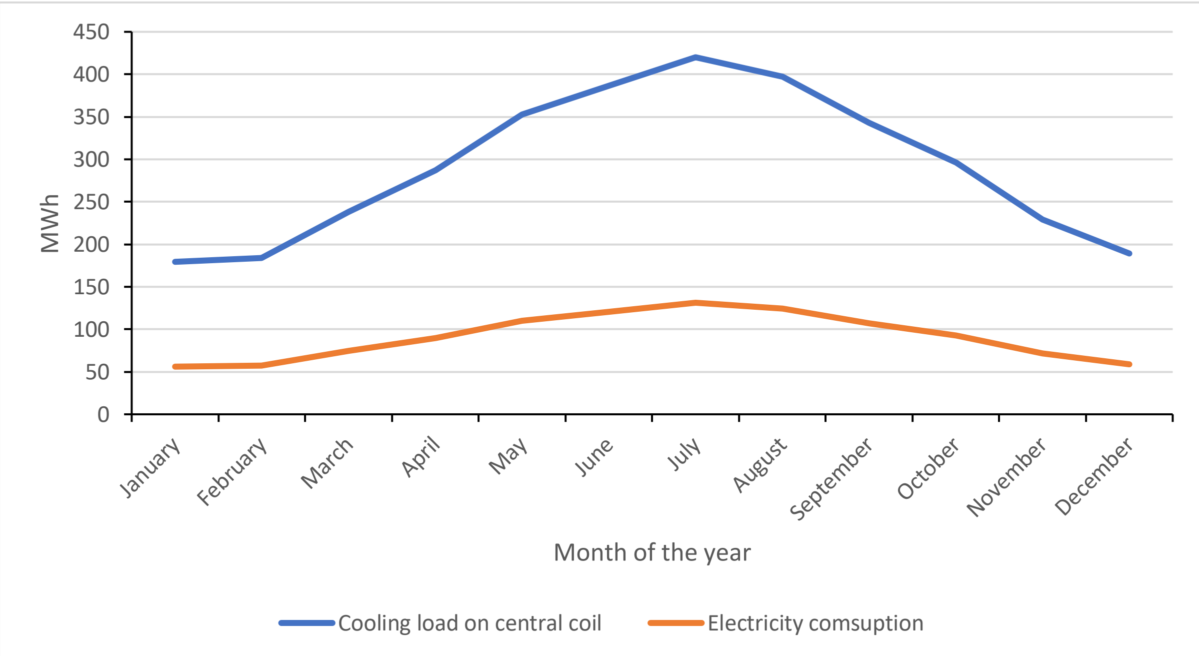

Figure 25 - Cooling load values and electricity consumption of the cooling coil dehumidifier throughout the entire year.

For both dehumidifiers, the price per kwh considered was of 0.145 €/kwh, since it is the cost in Portugal for production facilities. In the case of the cooling coil dehumidifier, the monthly electrical cost was calculated by multiplying the electrical consumption by 0.145 €/kwh, presented in the Figure 26.

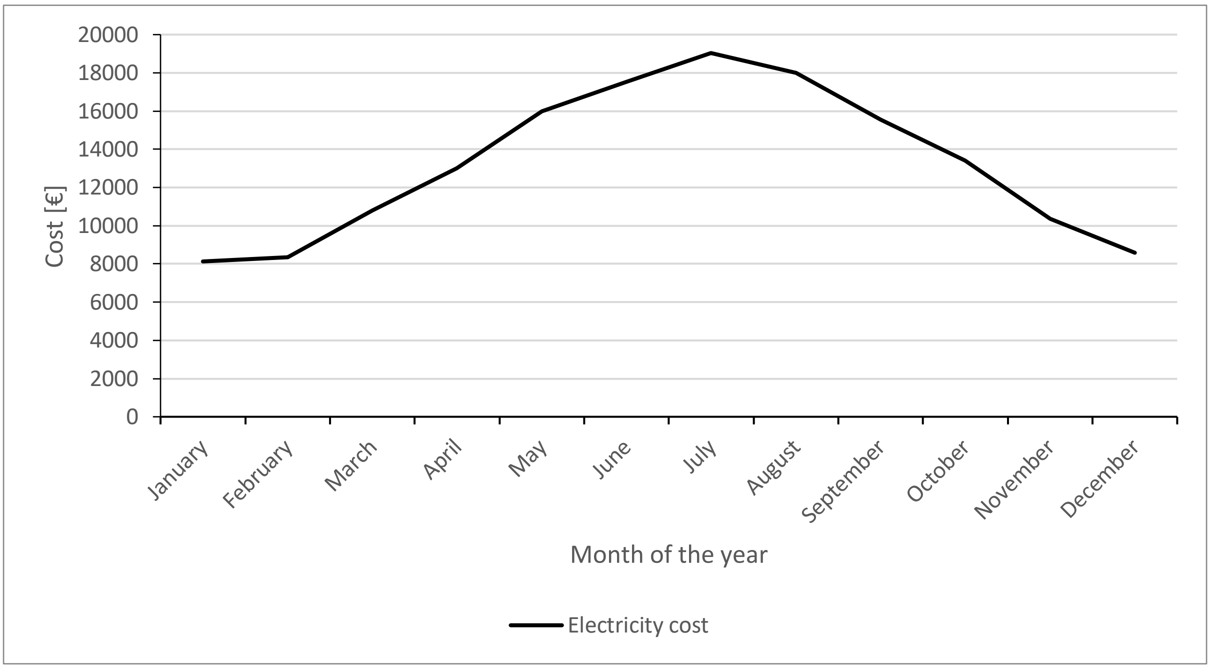

Figure 26 – Electricity cost in € throughout a year of operation, for the cooling coil dehumidifier.

For the dehumidification through condensation the author researched the market and found a rooftop unit capable of handling the total coil load required. It has a nominal cooling capacity of 199.5 kW and a discharge air flow of 47800 m^3/h. Since the required air flow is $190.5 \times 10^3\ m^3/h$, it will be needed 4 of those rooftop units.

For the desiccant dehumidifier, there is one available on the market that meets the requirements. It has a capacity to discharge $30 \times 10^3\ m^3/h$ at a 345 kW load. It has a condensate mass flow capacity of 220 kg_v/h. Since the necessary flow volume to insufflate the production hall is about $45 \times 10^3\ m^3/h$, it is needed two of these units. However, this desiccant dehumidifier doesn't operate constantly at the peak load of 345 kW. So, the author defined that 40% of 345 kW, is an appropriate constant load value to consider on the costs analysis. Considering 40% of the peak load to obtain a value of constant load is a rule of thumb in the HVAC industry. Since the author couldn't obtain more information about the electricity consumption, the costs analyses for the desiccant dehumidifier will be calculated by multiplying the cost of electricity with an average value of operational load.

The monthly cost of the desiccant dehumidifier was calculated with that value, multiplied by 8 hours of usage and 22 days, and multiplied by the monthly load and the number of equipment's required.

On Table 19 it is presented a comparison between the two dehumidifiers, in terms of investment costs, load capacity, the number of equipment's required and a monthly cost of electricity for their operation.

Also, it was considered a useful life of the equipment of 20 years so, at the bottom of the table it is presented an of electricity consumption cost and a total cost for two decades of operation. Those values where obtained by simply multiplying the annual electricity cost and the total cost in 1 year by 20, respectively.

Moreover, there was a few assumptions considered in this costs analysis to reduce the complexity. They are as follows:

- The regeneration energy of the desiccant dehumidifier was considered to be free.
- The costs of maintenance were considered to be the same on both equipment's, therefore they were not considered for the calculations.
- The electricity cost of 0.145 €/kwh was considered to be fixed throughout the years.
- It was considered that the initial cost (investment) was with own funds, meaning that it wasn't considered annual interest rates of loans.

Table 19 – Costs comparison between the rooftop unit and the desiccant dehumidifier.

	Roof top unit	**Desiccant dehumidifier**
Equipment investment cost (€)	37900	150000
Peak load capacity (kw)	538.1	345
Constant load capacity (kw)	215.24	138
Number of equipment	4	2
Total equipment investment cost (€)	151600	300000
Annual electricity cost (€)	1625478	84522
Total cost in 1 year (€)	1777078	384522
Electricity cost after 20 years (€)	32509547	1690445
Total cost after 20 years (€)	32661147	2074967

From Table 11 it is concluded that the desiccant dehumidifier cheaper. Despite having the higher equipment cost, after a year of operation the electricity costs are much lower in comparison to the rooftop unit.

In fact, subtracting the electricity cost of 20 years of both equipment's, by choosing the desiccant dehumidifier, the savings are around 30819103 €.

6. Conclusions and perspectives of possible future works

Climate changes, increased innovation, and growing attention to more energy efficient solutions is motivating companies to invest in HVAC installations.

A case study for air conditioning in a production hall was presented in this thesis. The case study focused on two types of dehumidifiers: dehumidifying cooling coil and desiccant dehumidifier. To assess which solution was the most viable, an costs analysis was carried out, concluding, under certain assumptions, that the desiccant dehumidifier was the most economically viable.

To assist the study, a computer program called Hourly Analysis Program (HAP) was used, which calculated the thermal heat loads and the cooling coil total load. Moreover, the desiccant dehumidifier calculations were defined by hand.

To increase the accuracy of this investment analysis, a more detailed study of the HVAC system is necessary, not only for the inclusion of other types of equipment in the study, but also for the inclusion of the estimated cost of electricity consumption for the remaining 11 months of the year, for the desiccant dehumidifier. For that it is necessary a detail technical data to be provided by the suppliers of such equipment.

It would increase the quality of the costs analyses, to study the sensibility of the electricity cost throughout the years.

It would also increase the accuracy of the analyses if there was more information about the leaving temperatures on the desiccant dehumidifier. Many articles and studies state that this equipment cannot deliver air at low temperatures, forcing the implementation of a chiller or a heat recovery wheel to reduce the insufflation temperature. That information can only be provided by the manufactures.

Also, it would be very beneficial to carry out a study of the implementation of both dehumidifiers. Comparing the results between a mixed system and the other two dehumidifying equipment's studied in this thesis would be very advantageous.

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Annex A: Principles of the Transfer Function Load Method explanation from HAP [9]

Principles of the Transfer Function Load Method

This help topic explains the key principles used to calculate building loads with the Transfer Function Method. This method is used in the program for cooling design calculations and for energy simulation load calculations. The explanation of Transfer Function Method principles requires covering a series of topics, including discussion of:

1. Necessary considerations for analyzing heat transfer in buildings.
2. Principles of the Heat Balance Method, which is the most rigorous load method available.
3. Fundamental principles of the Transfer Function Method which evolved from the Heat Balance Method.
4. Fundamental procedures used to calculate Transfer Function loads.
5. Examples illustrating how loads are calculated using Transfer Function procedures.

.Each topic will be covered in a separate section below. A final section will summarize the discussions.

A. Heat Transfer Considerations for Buildings

Most who study the process of calculating loads for commercial buildings reach the conclusion that the task is difficult and challenging because it involves:

a. The three modes of heat transfer: conduction, convection and radiation.
b. Many heat sources whose heat contributions vary with time and are usually not in phase with each other.
c. Storage and discharge of heat from massive building elements such as walls, roofs and floors. This results in transient rather than steady-state heat transfer processes.
d. Heat transfer processes that are interrelated.

To accurately estimate building loads, all of these factors must be considered.

To analyze building heat transfer, several key terms are used:

1. **Heat Gain** is the total quantity of heat entering a room from a specific heat source via the different conduction, convection and radiation modes of heat transfer.
2. **Load** is the heat transferred to room air which must be removed by the air-conditioning equipment if the zone is maintained at a constant air temperature.
3. **Heat Extraction** is the quantity of heat actually removed by the equipment as the zone temperature varies.

To illustrate building heat transfer in a room and the use of these terms, consider the simple example of lighting. Heat is gained from lights in the form of convected heat and radiated heat. Convected heat warms the air in the room and thus immediately becomes a load. Radiated heat is absorbed by the surfaces of walls, floors, ceilings and furnishings in the room. This causes the temperature of each surface to rise. Heat is then re-radiated to other surfaces in the room, is convected to air in the room, and is conducted into the mass adjacent to each surface. The absorption, storage, re-radiation and convection processes occur over a period of time. Consequently, lighting heat gain occurring during the current hour will be converted to a load during the current hour and a number of following hours. This is a transient heat transfer process. Finally, cooling equipment is sometimes not able to remove the entire load from the room as it occurs. As equipment responds to varying loads, the temperature in the room floats within the thermostat throttling range. Over a long period of time, the total heat extracted by equipment may equal the total load, but due to equipment and thermostat behavior, heat extraction and loads are not equal at every point in time.

Calculations required to account for the key aspects of these thermal processes can be complex and lengthy. The next section discusses the Heat Balance Method, which is the most rigorous approach to solving the building heat transfer problem. This will lead to a discussion of the Transfer Function Method which evolved from the Heat Balance Method.

B. The Heat Balance Method

The most rigorous method of calculating building loads is the Heat Balance Method. This method evaluates each conduction, convection, radiation and heat storage process occurring in a building using the fundamental laws of heat transfer and thermodynamics. Equations are written for each "node" in a room, where a "node" represents a surface or mass element associated with the room. For a mass element, the heat balance specifies in simple terms that:

(Rate of Heat In) - (Rate of Heat Out) = Energy Storage Rate

Since a surface has no mass, its heat balance equation for a surface in the room is:

(Rate of Heat In) - (Rate of Heat Out) = 0

To define the rate of heat in and out at each node, all convection, conduction and radiation processes must be represented. Typically these take the form of differential heat transfer equations or linearized forms of these equations. By solving all heat balances for a room simultaneously, the total rate of heat convection to room air can be determined. This defines the room load.

To illustrate how this works, consider the calculation of loads due only to heat gain from lighting. The heat transfer processes involved with converting lighting heat gain to load will be described first, followed by a summary of how the Heat Balance Method would be used to solve the heat transfer problem.

One portion of lighting heat gain is convective. Thus, it warms the room air directly and is immediately converted to a room load. The remainder of the heat gain is thermal radiation which strikes various surfaces in the room. Each surface absorbs a portion of the radiated heat and reflects the rest to other surfaces. The absorbed portion warms each surface which initiates further heat transfer processes. For a wall, for example, this can create a thermal gradient between the surface and the interior mass of the wall. Therefore, some heat might be conducted from the surface into the wall mass where it is stored. When conditions change later, causing the wall surface to cool, the stored heat is discharged by being conducted back to the wall surface. The warm wall surface also can create a temperature gradient between the wall surface and room air leading to convection which converts the heat to a load. Finally, the elevated surface temperature causes a thermal radiation exchange between the wall surface and other surfaces in the room, where similar processes are occurring. These processes are continually occurring and changing.

Using the Heat Balance Method to solve this heat transfer problem would require writing heat balance equations for each surface and mass element considering each process involved:

1. Convection of heat from the light to room air.
2. Radiation from the light to each surface in the room.
3. Conduction, convection and radiation exchange for each surface in the room.
4. Conduction and heat storage for each mass element in contact with a surface.

Once equations have been formulated, they must be solved simultaneously each hour to evaluate the ebb and flow of heat in the room.

Note that the processes described above are not isolated. Each surface in the room is also receiving heat from other sources such as people, appliances, solar heat gain and heat conducted through walls and roofs. To determine the total heat convected to room air (i.e., the load), all processes must be simultaneously evaluated.

The results of such an analysis can be highly accurate, insofar as input data describing sources of heat gain and the physical and geometric properties of the room allow. While computer programs have been written to perform these calculations, they typically require powerful computer hardware, detailed input data and long calculation times. Using desktop computers for typical design applications, the Heat Balance Method tends to be difficult to use. However, its comprehensive approach to modeling the actual heat transfer processes in a building are desirable. The next section describes the Transfer Function Method which evolved from the Heat Balance Method and, using key assumptions, shortens calculation times and is able to provide sophisticated, accurate building load estimates.

C. Fundamental Transfer Function Principles

The Transfer Function Method is the culmination of work first published in 1967 by two scientists working for the Canadian National Research Council. The method is based on an idea known as the "Response Factor Principle". This principle states that for a specific room, the thermal response patterns (i.e., how a heat gain is converted to load over a period of time) for each specific type of heat gain will always be the same. That is, for a specific room, a 1000 BTU/h heat gain through an exterior wall will cause the same response over a period of hours as a 2000 BTU/h heat gain. The sizes of the loads will differ, but the heat gain to load conversion pattern will be the same.

The Response Factor Principle is in turn based on three additional principles:

1. *The Principle of Superposition:* The total room load is equal to the sum of loads calculated separately for each heat gain component.
2. *The Principle of Linearity:* The magnitude of the thermal response to a heat gain varies linearly with the size of the heat gain.
3. *The Principle of Invariability:* Two heat gains of equal size occurring at different times will produce the same thermal response in a room.

Together, these principles allow the simplification of the Heat Balance Method analysis for a building:

1. The Principle of Superposition can be used to break the heat transfer problem into manageable units since it allows loads due to each identifiable heat gain component to be computed separately. For example, loads due to an exterior wall heat gain, and a lighting heat gain can be computed separately and then added together to determine the total room load. By contrast, the heat balance method requires all heat gains in a room to be considered simultaneously.
2. The Principle of Superposition also allows the effects of heat gains each hour to be considered separately. For example, a lighting heat gain this hour will cause loads in the current hour and a number of following hours. This is because a portion of the heat gain is thermal radiation which is absorbed by the walls, floor and furnishings of the room, and is then convected to room air over time. When lights are on for several hours, the load in any one hour is due to heat gain during the current hour, and heat gains in a number of previous hours. With the Principle of Superposition, the pattern of loads due to each hourly heat gain can be computed separately and then added together to determine the total lighting load each hour. By contrast, the heat balance method requires the effects of heat gains for the current hour and previous hours to be considered simultaneously.
3. The Principles of Linearity and Invariability permit a vast reduction in the number of calculations needed. Because the pattern of loads resulting from each type of heat gain will be invariable, the pattern only needs to be determined one time via heat balance computations. Then, because the magnitude of each load is a linear function of the size of the heat gain, load patterns due to each hourly heat gain can be easily computed using simple algebraic equations. By contrast, the heat balance method requires solving a series of heat balance equations simultaneously each hour.

D. Fundamental Transfer Function Procedures

With the Transfer Function Method, a general mathematical relationship which defines load as a function of heat gain and time is determined for each heat gain component in a room. This relationship is then used to quickly calculate loads for each hour. The mathematical relationship is expressed in what is called a Room Transfer Function Equation which looks like this:

$$Q_0 = v_0q_0 + v_1q_1 + v_2q_2 - w_1Q_1 - w_2Q_2$$

In this equation:

1. Q represents a load. The subscripts refer to specific points in time. Subscript 0 is the current hour, 1 is the previous hour and 2 is two hours previous.
2. q represents a heat gain. The subscripts 0, 1 and 2 have the same meaning as for loads.
3. v_0, v_1, v_2, w_1 and w_2 are transfer function coefficients. Values of these coefficients vary for each type of heat gain and room due to the different heat transfer processes involved in converting each kind of heat gain into a load. ASHRAE has published tables of these coefficients for different heat gain components, room types, and building weights.

In words, the Room Transfer Function Equation says that the load for the current hour (Q_0) is a function of the heat gain for the current and preceding two hours, plus the loads for the preceding two hours. Because loads for the preceding two hours are themselves dependent on a series of heat gains for prior hours, this hour's load is really dependent on the effects of heat gains from many preceding hours.

To use the Room Transfer Function Equation, transfer function coefficients must first be obtained. Then for any type of heat gain, calculating loads is a two-step process:

1. Determine the heat gains for a series of hours.
2. Use the heat gains and transfer function coefficients together in the Room Transfer Function Equation to calculate the loads.

The following two sections provide examples of how loads for an internal heat gain (lighting) and a transmission heat gain (wall) are calculated.

Annex B: Climatic conditions for Porto, Portugal [7]

PORTO/PEDRAS RUBRAS, Portugal WMO#: 085450

Lat: 41.23N Long: 8.68W Elev: 77 StdP: 100.4 Time Zone: 0.00 (GMT) Period: 82-06 WBAN: 99999

Annual Heating and Humidification Design Conditions

Coldest Month	Heating DB		Humidification DP/MCDB and HR						Coldest month WS/MCDB				MCWS/PCWD to 99.6% DB	
			99.6%			99%			0.4%		1%			
	99.6%	99%	DP	HR	MCDB	DP	HR	MCDB	WS	MCDB	WS	MCDB	MCWS	PCWD
1	1.9	3.1	-4.5	2.6	7.3	-2.5	3.1	7.6	12.3	12.6	10.9	12.5	2.5	90

Annual Cooling, Dehumidification, and Enthalpy Design Conditions

Hottest Month	Hottest Month DB Range	Cooling DB/MCWB						Evaporation WB/MCDB						MCWS/PCWD to 0.4% DB	
		0.4%		1%		2%		0.4%		1%		2%			
		DB	MCWB	DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB	WB	MCDB	MCWS	PCWD
8	9.0	30.3	19.3	28.1	18.7	26.0	18.0	20.7	26.7	20.0	25.1	19.3	23.6	3.8	320

Dehumidification DP/MCDB and HR									Enthalpy/MCDB						Hours 8 to 4 & 12.8/20.6
0.4%			1%			2%			0.4%		1%		2%		
DP	HR	MCDB	DP	HR	MCDB	DP	HR	MCDB	Enth	MCDB	Enth	MCDB	Enth	MCDB	
19.1	14.0	21.7	18.3	13.3	20.9	17.9	13.0	20.5	59.9	26.9	57.3	25.2	55.0	23.7	1870

Extreme Annual Design Conditions

Extreme Annual WS			Extreme Max WB	Extreme Annual DB				n-Year Return Period Values of Extreme DB							
				Mean		Standard deviation		n=5 years		n=10 years		n=20 years		n=50 years	
1%	2.5%	5%		Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max
9.9	8.5	7.4	27.7	-0.7	34.9	1.3	1.7	-1.6	36.0	-2.4	37.0	-3.1	37.9	-4.0	39.1

Monthly Climatic Design Conditions

			Annual	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
Temperatures, Degree-Days and Degree-Hours	Tavg		14.7	9.9	10.6	12.3	13.1	15.2	18.0	19.3	19.6	18.5	16.0	12.9	11.0
	Sd			2.50	2.55	2.67	2.74	2.75	2.87	2.58	2.65	2.59	2.52	2.82	2.56
	HDD10.0		93	33	22	8	4	0	0	0	0	0	0	6	19
	HDD18.3		1495	262	218	187	159	105	39	14	11	27	81	164	229
	CDD10.0		1816	30	37	81	97	162	239	289	298	254	186	93	49
	CDD18.3		174	0	0	1	1	8	28	44	51	32	8	1	0
	CDH23.3		1199	0	0	8	13	64	220	336	346	177	36	0	0
	CDH26.7		364	0	0	0	1	14	66	114	119	46	4	0	0
Monthly Design Dry Bulb and Mean Coincident Wet Bulb Temperatures	0.4%	DB		17.4	19.7	24.3	25.2	28.7	31.5	33.0	33.3	31.0	26.8	21.9	19.1
		MCWB		12.4	13.4	13.9	15.7	17.5	19.2	20.0	19.8	19.2	18.1	15.9	13.5
	2%	DB		15.6	17.2	21.1	22.1	24.9	28.5	29.9	29.9	27.8	24.0	19.8	16.9
		MCWB		12.2	11.9	13.4	14.3	16.2	18.5	19.6	19.4	18.5	16.9	15.3	13.7
	5%	DB		14.7	15.7	18.8	19.9	22.0	25.9	27.1	27.1	25.1	22.0	18.2	15.9
		MCWB		12.2	11.7	12.8	13.9	15.9	17.7	19.1	18.7	18.0	16.6	15.3	13.9
	10%	DB		14.0	14.4	16.9	17.9	19.8	23.1	24.3	24.6	23.0	20.1	17.1	15.0
		MCWB		11.9	11.7	12.1	13.2	15.2	17.5	18.4	18.3	17.6	16.3	14.6	12.9
Monthly Design Wet Bulb and Mean Coincident Dry Bulb Temperatures	0.4%	WB		14.7	14.9	15.7	17.4	19.2	21.1	21.8	21.9	21.0	19.8	18.1	16.1
		MCDB		15.3	16.8	20.4	21.6	24.8	27.7	28.9	27.8	26.9	22.1	18.9	16.5
	2%	WB		13.9	14.0	14.6	15.8	17.5	19.8	20.6	20.6	19.9	18.7	16.9	15.4
		MCDB		14.5	15.4	18.4	19.3	22.2	25.9	27.0	26.2	24.3	21.6	18.4	15.9
	5%	WB		13.2	13.2	13.8	14.9	16.5	18.8	19.7	19.7	19.1	17.8	16.1	14.3
		MCDB		14.0	14.5	16.7	18.0	20.3	23.6	24.9	24.3	22.8	20.4	17.6	15.4
	10%	WB		12.4	12.5	13.1	14.0	15.7	18.0	18.8	19.1	18.5	17.0	15.2	13.4
		MCDB		13.6	13.9	15.7	16.9	18.8	21.9	23.0	23.0	21.7	19.3	16.7	14.7
Mean Daily Temperature Range		MDBR		7.2	7.8	8.5	7.9	7.6	8.0	8.6	9.0	8.6	7.7	7.0	6.7
	5% DB	MCDBR		7.4	9.5	11.9	11.4	11.5	12.6	13.3	13.2	12.1	10.7	7.7	6.9
		MCWBR		5.2	6.0	6.7	6.5	5.9	5.9	6.0	6.0	5.7	5.7	4.9	5.0
	5% WB	MCDBR		4.7	6.2	9.2	9.2	9.0	9.8	10.9	9.9	9.2	8.0	5.6	4.9
		MCWBR		4.4	4.8	5.8	5.9	5.2	5.3	5.7	5.4	5.2	5.0	4.3	4.4
Clear Sky Solar Irradiance	taub			0.332	0.350	0.384	0.378	0.405	0.417	0.410	0.421	0.392	0.378	0.347	0.338
	taud			2.381	2.299	2.146	2.227	2.150	2.141	2.205	2.146	2.256	2.236	2.349	2.347
	Ebn,noon			816	853	860	893	872	859	862	841	842	807	787	771
	Edh,noon			92	111	140	137	150	152	141	146	124	115	94	90

CDD*n* Cooling degree-days base n°C, °C-day
CDH*n* Cooling degree-hours base n°C, °C-hour
DB Dry bulb temperature, °C
DP Dew point temperature, °C
Ebn,noon } Clear sky beam normal and diffuse hori-
Edh,noon } zontal irradiances at solar noon, W/m2
Elev Elevation, m
Enth Enthalpy, kJ/kg
HDD*n* Heating degree-days base n°C, °C-day
Hours 8/4 & 12.8/20.6 Number of hours between 8 a.m. and 4 p.m with DB between 12.8 and 20.6 °C
HR Humidity ratio, g of moisture per kg of dry air
Lat Latitude, °
Long Longitude, °
MCDB Mean coincident dry bulb temperature, °C
MCDBR Mean coincident dry bulb temp. range, °C
MCDP Mean coincident dew point temperature, °C
MCWB Mean coincident wet bulb temperature, °C
MCWBR Mean coincident wet bulb temp. range, °C
MCWS Mean coincident wind speed, m/s
MDBR Mean dry bulb temp. range, °C
PCWD Prevailing coincident wind direction, °, 0 = North, 90 = East
Period Years used to calculate the design conditions
Sd Standard deviation of daily average temperature, °C
StdP Standard pressure at station elevation, kPa
taub Clear sky optical depth for beam irradiance
taud Clear sky optical depth for diffuse irradiance
Tavg Average temperature, °C
Time Zone Hours ahead or behind UTC, and time zone code
WB Wet bulb temperature, °C
WBAN Weather Bureau Army Navy number
WMO# World Meteorological Organization number
WS Wind speed, m/s