Abstract

Heating, Ventilating and Air Conditioning, HVAC, is a huge field. HVAC systems Cooling equipment varies from the small domestic unit to refrigeration machines that are 10,000 times the size.

→ This report will introduce:

- the fundamental concepts that are used by designers to make decisions about system design, operation
- Classification of Air Conditioning Systems
- Types of ventilation and calculation of it
- Types of hoods and design of it
- Process of psychometric process
- Design of operation room and how to control pressure of spaces
- Load calculation based on hap and manual
- Air terminal types and distribution of it
- Air distribution system

Dedication

To our beloved families who gave us lessons in spiritual things.

To Eng / Ahmed Shuhayb

To the entire engineering faculty members in particular.

And mostly especially to our Almighty Allah our God

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Nomenclature

Cfm	Cubic Feet Per Minute
Kw	Kilo Watt
Btu	British Thermal Unit
Ft	Feet
Fpm	Feet Per Minute
In	Inch

Abbreviation

Wbt	Wet Bulb Temperature
Dbt	Dry Bulb Temperature
Ach	Air Change Per Hour
Rh	Relative Humidity
Qt	Heat Transmission Load
Qs	Sun Heat Gain
Qp	Occupants Load
Qinf	Infiltration Load
Hap	Hourly Analysis Program

chapter (1): introduction

1.1 What is HVAC?

Hvac is an abbreviation for Heating Ventilation and Air-conditioning, is a huge field Where each term can be defined individually

- Heating → It is defined as a process of increasing the temperature of a substance or a body
    - heating often used in cold climates to heat private houses and public building
    - Heating systems usually comprise of a boiler, furnace, heat pump
- Cooling → It is defined as a process of decreasing the temperature of a substance or a body
    - Cooling is designed to keep the indoor conditions as per human comfort.
    - Cooling systems usually comprise of a chiller, indoor units like fan coil unit or air handling unit to cool the room air
- Ventilation → It is defined as a process of exchanging air between the outdoors and the conditioned space for the purposes of diluting the gaseous contaminants in the air and improving or maintaining air quality
    - Ventilation can be achieved either through natural ventilation or mechanical ventilation.
- air conditioning → Air conditioning for any space means the control of the followings:
    - Temperature
    - humidity
    - Air Purity, and cleaning
    - Air distribution inside the space

- Human Comfort Condition:

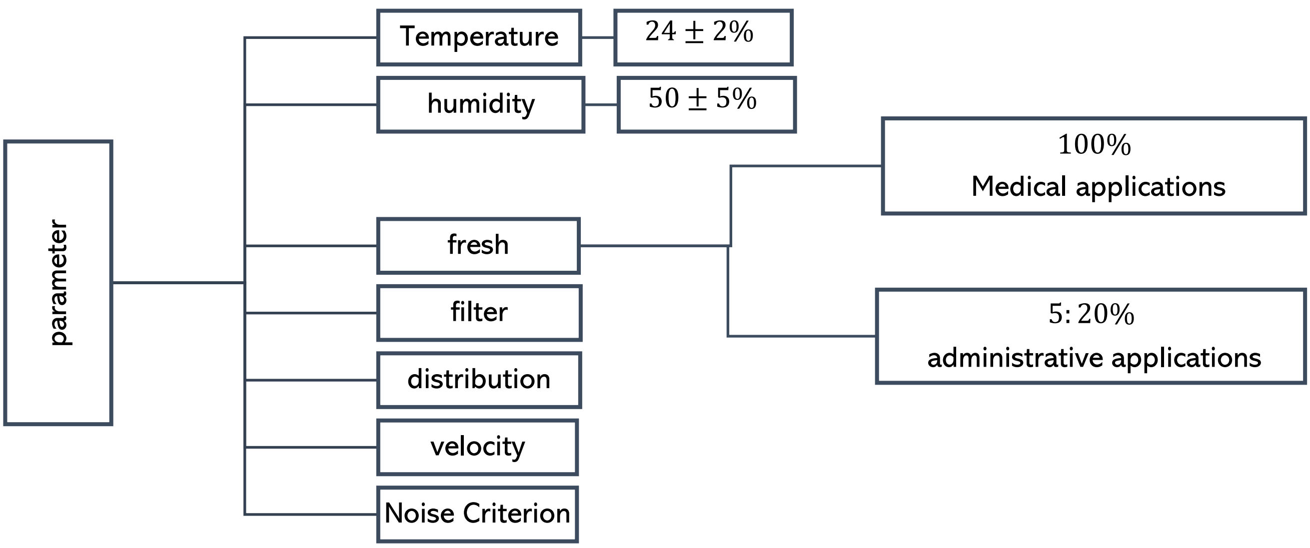


Figure 1-1 : Human Comfort Condition

1.2 simple vapor compression cycle

1.2.1 component of cycle

- The cycle has four basic components
    1 Compressor
    2 Condenser
    3 Expansion valve
    4 Evaporator

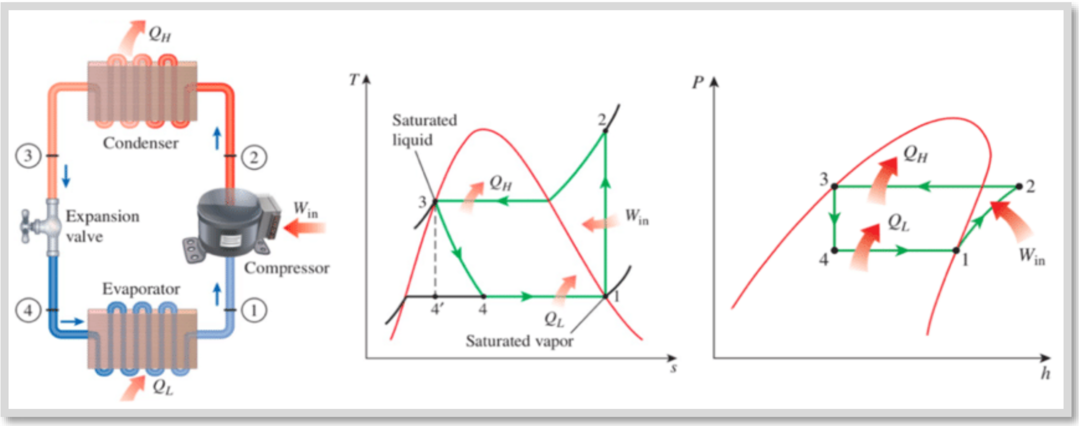


Figure 1-2 : simple vapor compression cycle

1) Compressor

It is a device used to increase the pressure of the refrigerant from evaporator pressure to condenser pressure.

2) Condenser

- Condensation is a process of change of phase from vapor form to liquid form,
    - A condenser converts the refrigerant gas at high pressure & temperature into a liquid refrigerant at nearly high pressure & temperature by releasing the latent heat from the gas.
- Condensers are classified as:
    1. Air Cooled
    2. Water Cooled

3) Expansion Device

The function of expansion device is to decrease the pressure & temperature of the liquid refrigerant by the process of expansion.

- Types of Expansion Devices:
    1. Capillary Tubes
    2. Thermostatic Expansion Valves
    3. Electronic Expansion Valves

4) Evaporator

Evaporation is a process of change of phase from liquid to gas form

- The evaporator converts the liquid refrigerant at low pressure & temperature into its vapor form by absorbing the latent heat from the room air

- Types of Evaporators:
  1. Air Cooled
  2. Water Cooled

1.2.2 vapor compression cycle process

- The processes shown in (Figure 1-2) can be summarized in the following table (Table 1-1): -

Table 1-1:Vapor Compression Refrigeration Cycle

1→2 Isentropic compression inside the compressor (S=C).
2→3 Heat rejection inside the condenser at (P=C).
3→4 Isentropic expansion inside the expansion (S=C).
4→1 Heat addition inside the evaporator at (P=C)

1.2.3 Working of Refrigeration Cycle:

1. The refrigerant comes into the compressor as a low-pressure vapor, it is compressed and then moves out of the compressor as a high-pressure vapor.
2. The vapor then flows to the condenser here the vapor condenses to a liquid and gives off its heat to the outside air
3. The liquid then moves to the expansion valve under high pressure this valve restricts the flow of the fluid, and lowers its pressure as it leaves the expansion valve
4. The low-pressure liquid then moves to the evaporator, where heat from the inside air is absorbed and changes it from a liquid to a vapor.

1.3 Psychometrics

Psychometric is the science, which deals with the thermodynamic properties of the mixture of the dry air and the water vapor, which is called moist air.

→ Moist air: is considered as a binary (or two components) mixture of dry air and water vapor.

1.3.1 Properties of Moist air

1) Dry- bulb temperature (dbt)
2) Wet- bulb temperature (wbt)
3) Dew point Temperature (dpt)
4) Relative Humidity (RH)
5) Humidity ratio (w)
6) Enthalpy of Moist Air (h)
7) Specific Volume, m3/kg

1.3.1.1 Dry- bulb temperature (dbt)

The dry bulb temperature is the temperature of moist air measured by any ordinary thermometer or by thermocouple.

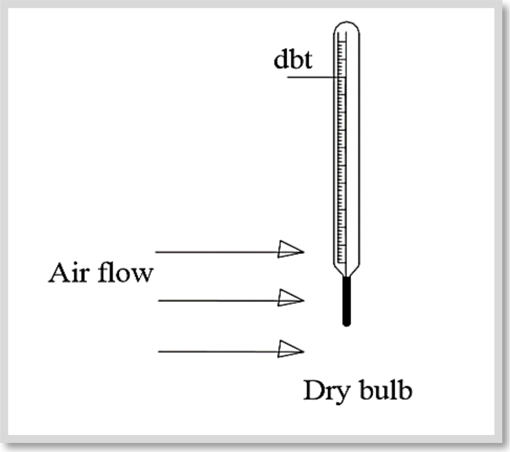


Figure 1-3 : dry -bulb temperature

1.3.1.2 Wet- bulb temperature (wbt)

The wet bulb temperature is the temperature measured by a thermometer, which its sensor bulb coated by wetted wick and is exposed to the current of moving air and The difference between the dry-bulb and wet bulb temperature is known as wet-bulb depression, Wet bulb depression becomes zero when the air is completely saturated.

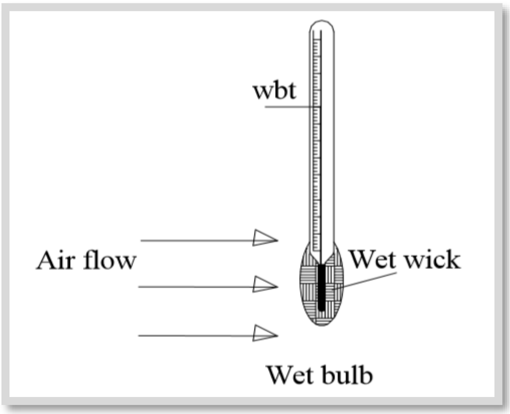


Figure 1-4 : wet bulb temperature

1.3.1.3 Dew point Temperature (dpt)

The dew point temperature is the temperature at which the water vapor in the air starts to condensate

Dew point temperature is equal to the steam table saturation temperature

1.1.1.1 Humidity ratio (w) (Specific Humidity)

It is defined as the mass of water vapor present per kg of dry air

$$w = \frac{m_w}{m_a} = \frac{\text{mass of water vapor in air}}{\text{mass of air}}$$

1.3.1.4 Relative Humidity (RH)

The relative humidity is defined as the ratio of actual mass of water vapor in air to the mass of water vapor if the air is saturated (max) at the same dry bulb temperature.

$$RH = \frac{m_v}{m_{v.s}}$$

1.3.1.5 Enthalpy of Moist Air (h)

The enthalpy is the total energy of a thermodynamic system and the enthalpy of moist air is the sum of the enthalpy of dry air and the enthalpy of water vapor associated with dry air

1.3.1.6 Specific Volume, m3/kg

The volume of the mixture per kg of dry air in the mixture

1.4 Psychometric charts

Psychometric charts provide a graphical representation of the thermodynamic properties of moist air, various air conditioning processes and, variable air conditioning cycles, where dry bulb temperature, and humidity ratio are basic coordinate, as the chart is a representation of dry-bulb temperature versus humidity ratio and all other properties are shown by different lines on the chart.

1.4.1 Determination of Thermodynamic Properties on Psychometric charts

There are seven thermodynamic properties groups of moist air shown on psychrometric charts shown in (Figure 1-5). They are:

1) Dry- bulb temperature (dbt)
2) Wet- bulb temperature (wbt)
3) Dew point Temperature (dpt)
4) Relative Humidity (RH)
5) Humidity ratio (w)
6) Enthalpy of Moist Air (h)
7) Specific Volume, m3/kg

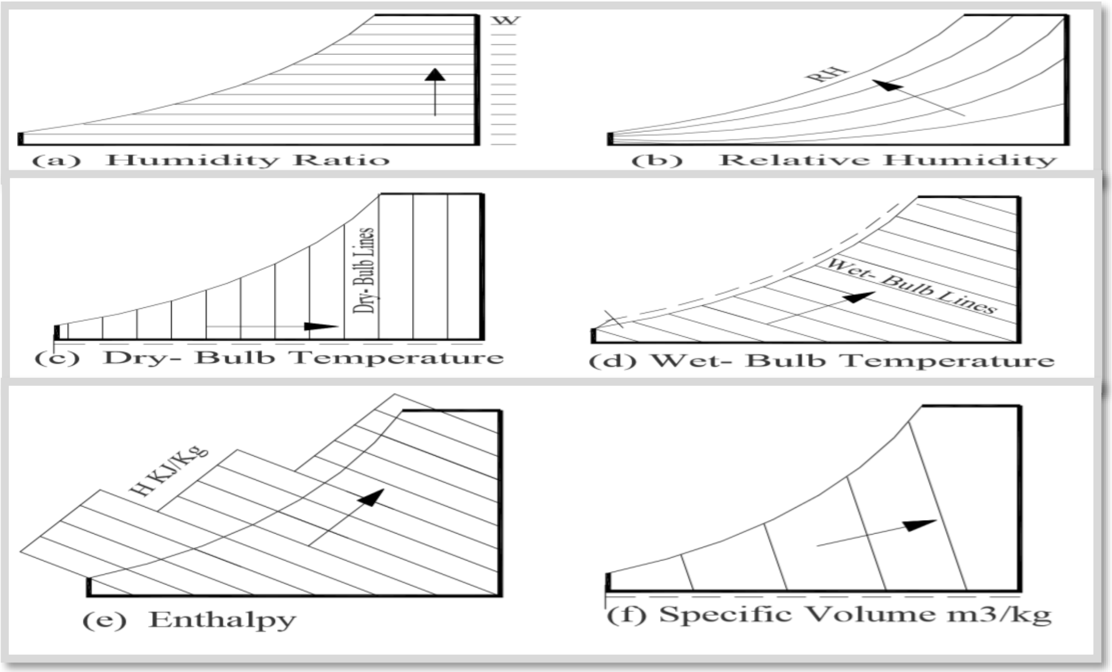


Figure 1-5 : Thermodynamic Properties on Psychometric charts

1.5 Psychometric processes

An air conditioning process determines the change in thermodynamics properties of moist air between the initial and final states of conditioning and also the corresponding energy transfer between the moist air and medium such as water, refrigerant, or moist air itself during the change.

The commonly six processes shown in (Figure 1-6) described in the next section include:

1) Sensible heating
2) Sensible cooling
3) Cooling and dehumidification
4) Cooling and Humidification
5) Heating and Humidification
6) Heating and dehumidification

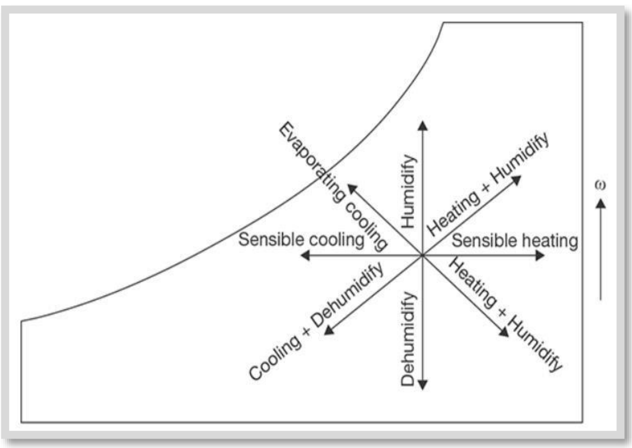


Figure 1-6 : psychometric processes

1.5.1 Sensible Heating Process

- Sensible heating process means the heating of the air without the addition or remove of moisture (w1=w2) shown in (Figure 1-7).
- Heating can be provided using heating coil or electric heater.
- In this process: dbt2 > dbt1, h2 > h1, Rh2 < Rh1

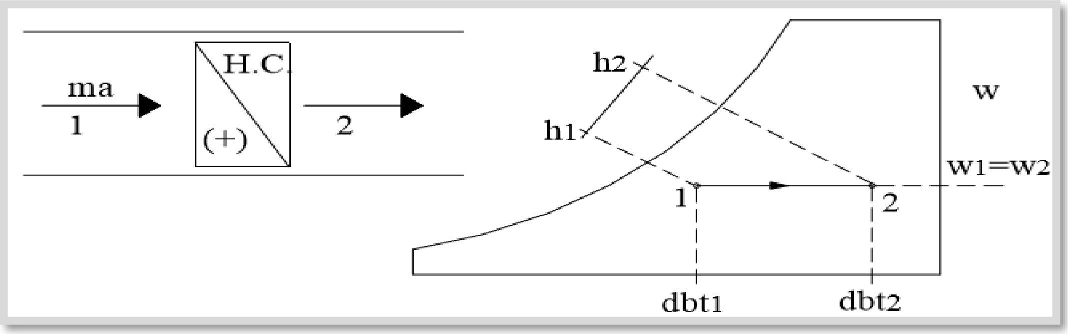


Figure 1-7 : Sensible heating process

1.5.2 Sensible Cooling Process (cooling above dpt) dbt2 > dpt

- Sensible Cooling process means the cooling of the air while maintaining a constant humidity ratio (w1=w2) of the moist air shown in (Figure 1-8).
- The sensible cooling process occurs using cooling coil.
- In this process: dbt2 < dbt1 , h2 < h1 ,Rh2 > Rh1

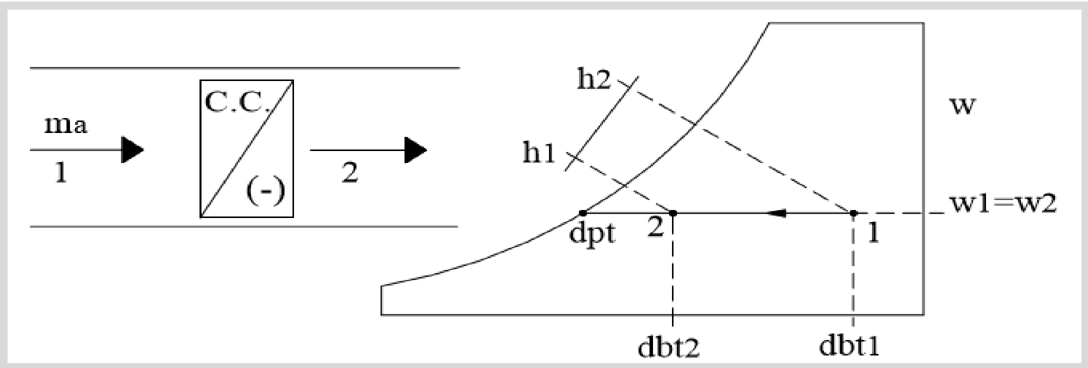


Figure 1-8 : Sensible Cooling Process

1.5.3 Cooling with Dehumidification Process (Cooling below dpt)

- The removal of the water vapor from the air is termed as dehumidification of air
- The cooling and dehumidification of air is only possible if the air is cooled below the dew point temperature (dpt) shown in (Figure 1-9)
- It is necessary to maintain the coil surface temperature (Ts) below dew point temperature of air for effective dehumidification.
- can be provided using Water cooling coil with chilled water flowing inside the coil’s tubes.

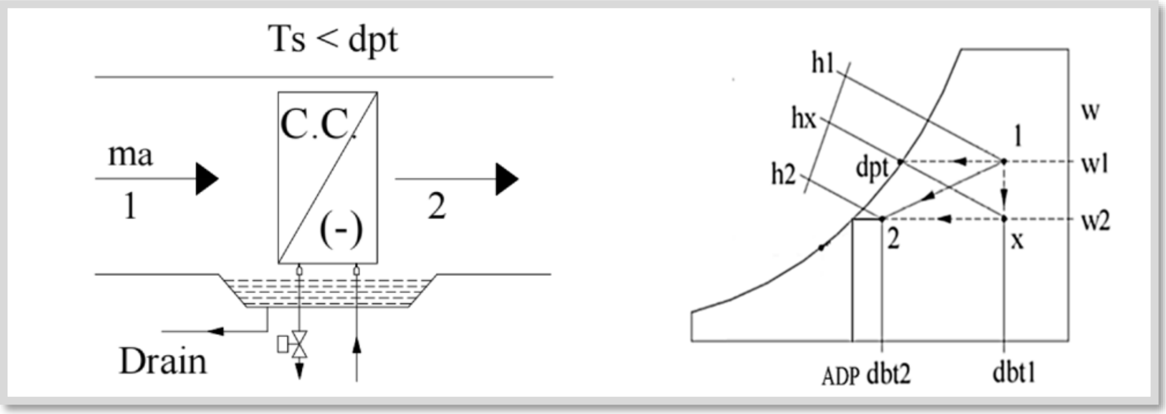


Figure 1-9 :Cooling with Dehumidification Process

1.5.3.1 Apparatus Dew Point Temperature (ADP)

The temperature (dbt3 or T3) corresponding to the point 3, shown on the saturation curve is known as "Apparatus Dew Point." (ADP), which is also equal to the coil surface temperature (Ts) shown in (Figure 1-10)

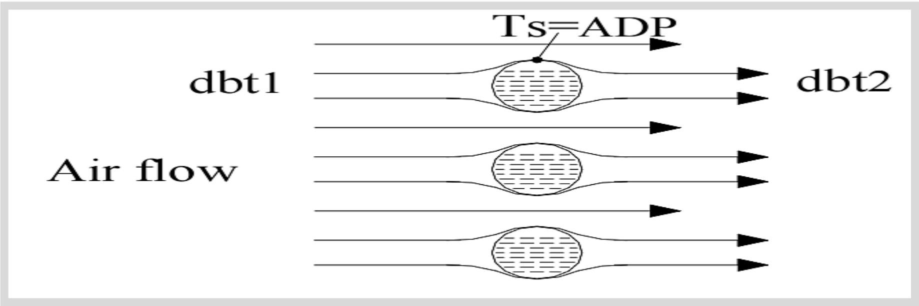


Figure 1-10 : Apparatus Dew Point Temperature

1.5.3.2 Coil By Pass Factor (BF)

- As the moist air at temperature (dbt1) passing over a cooling coil at a surface temperature (Ts) which equal the ADP.
- Part of the air shall contact the coil and other part shall pass without contact the surface shown in (Figure 1-10)
- thus, the final mixture condition of the air leaving the coil shall be at a temperature dbt2 which is greater than Ts (dbt2> Ts)
- The by-pass factor is an important tool for measuring the performance of the coils The performance of the coil is better with lower bypass factor
    - Sensible Heat Ratio (SHR)
- The ratio $\frac{Qsen}{Qtotal}$ is called sensible heat ratio or sensible heat factor and it is given as:

$$SHR = \frac{Qsen}{Qsen + Qlat}$$

1.5.4 cooling and humidification by using (Air Washer – evaporative cooling)

- Cooling above dpt
- temperature of water droplets lower than dry bulb temperature of air
- In this process Tw < T1

In an air washer, a small pump supplies the water at a temperature (Tw) to banks of spraying nozzles under high pressure. The un-evaporated water is collected and re- circulated again and again for humidification purpose shown in (Figure 1-11)

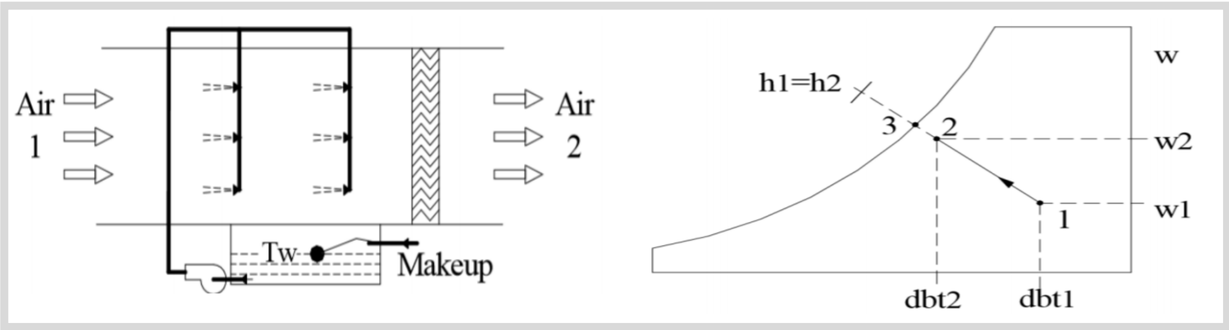


Figure 1-11 : cooling and humidification by using Air Washer

1.5.5 Heating and humidification by using Air Washer

- Temperature of water droplets greater than dry bulb temperature of air shown in (Figure 1-12)
- Tw > T1

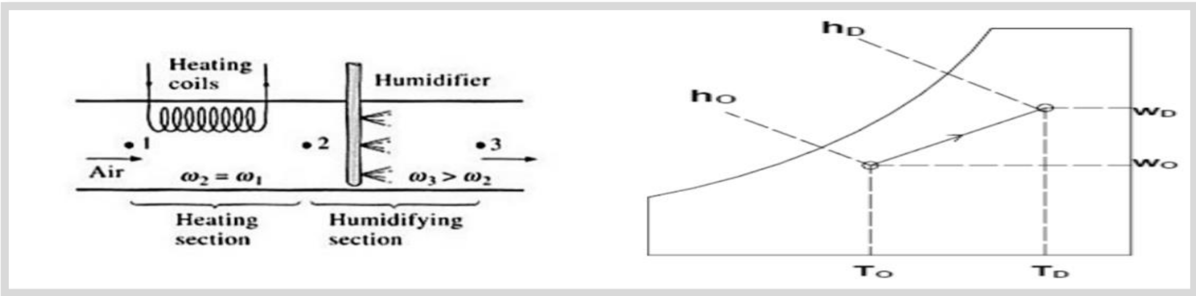


Figure 1-12 : Heating and humidification by using Air Washer

1.5.6 Heating and dehumidification (Chemical Dehumidification)

- The process in which the air is heated and at the same time moisture is removed from it is called as heating and dehumidification process. This process is obtained by passing the air over certain chemicals like alumina and molecular sieves.

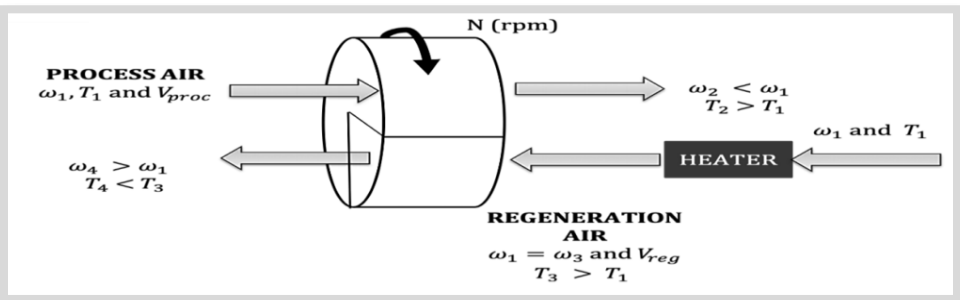


Figure 1-13 : Heating and dehumidification process

chapter (2): Air Conditioning Systems

2.1 General Classification of Air Conditioning Systems

The Air Conditioning Systems can be classified into two main categories

1) direct expansion DX unit
2) chilled water system
   - → In The Direct Expansion (DX) direct heat transfer between air and Freon this used in small and medium load
   - → In chilled water system no direct contact between air and Freon and the heat exchange process is carried out by means of a medium called chilled water and used in large load

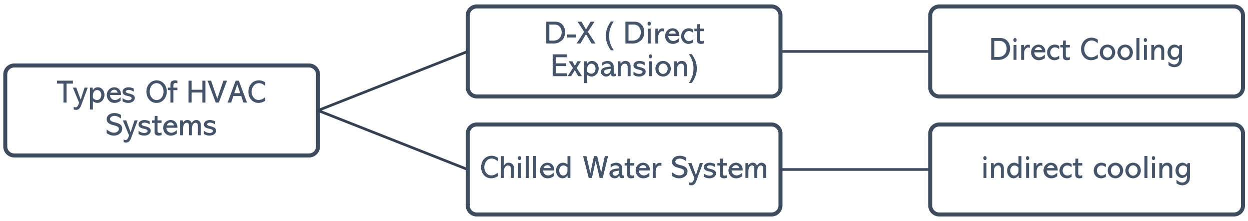


Figure 2-1:DX Units vs Chilled Water System

2.1.1 Comparison between HVAC systems

2.1.1.1 Why use chilled water in chiller system?

1) Water has a much higher heat capacity, and specific heat, than air, meaning it takes more energy to heat water than it does to heat air Water has a specific heat of 4.186 $\frac{J}{g\,k}$ versus air, which has a specific heat of 1.005 $\frac{J}{g\,k}$
2) Less running cost
3) The density of water is equal 1000 kg/m^3 and the density of air is equal 1.21 kg/m^3 also we know that $\rho = \frac{1}{v}$ this meaning the specific volume of water is lower than air It makes me able to transfer a lot of energy in a small diameter This can be summarized in the following (**Error! Reference source not found.**)

Table 2-1 : Difference Between DX Units and Chilled Water Units

Direct Expansion System (dx)	Chilled Water System
• Air is cooled by Freon • High running cost and Low Initial Cost • Easy maintenance	• Cooling is done by chilled water which is Cooled by Freon • Low running cost and High initial cost • Complicated design & maintenance

2.1.1.2 Difference between a package and a split system

→ split system

The split system consists of Two main components shown in (Figure 2-2):

1) outdoor compressor/condenser
2) indoor air (evaporator).

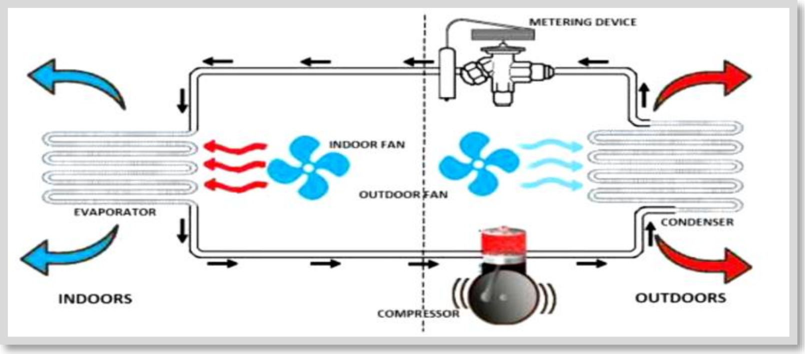


Figure 2-2 : split system component

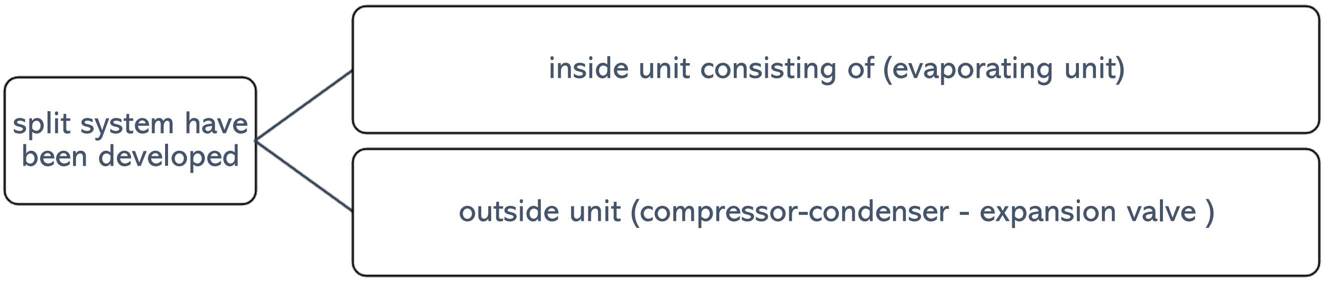


Figure 2-3 : split system

→ Package system

Package units will contain all of their parts in one outdoor unit like Window Unit shown in (Figure 2-4)

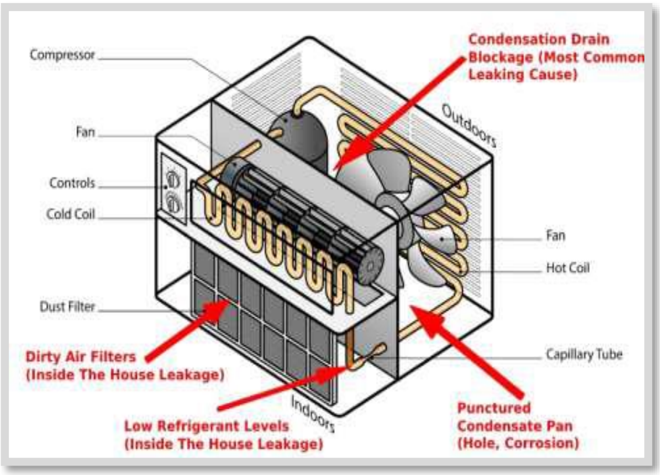


Figure 2-4 : Package system component

2.1.1.3 Advantages and Disadvantages of Window/split Air Conditioners

Advantages of the Window type are the disadvantages of the split type shown in (Table 2-2).

Table 2-2 : Advantages and Disadvantages of Window/split Air Conditioners

Split A/C		Window A/C	
Advantages	Disadvantages	Advantages	Disadvantages
• No noise	• higher Cost.	• Low Cost.	• Noisy
• Higher Capacities.	• high pipes Loss.	• Easy Maintenance. Portable	• Limited Capacities.
• No Decoration Problems	• Maintenance is harder	• Low pipes Loss.	• Decoration Problems
	• Not Portable		

2.2 Classification of Air Conditioning Systems in details

The Air Conditioning Systems can be classified shown in (Figure 2-5)

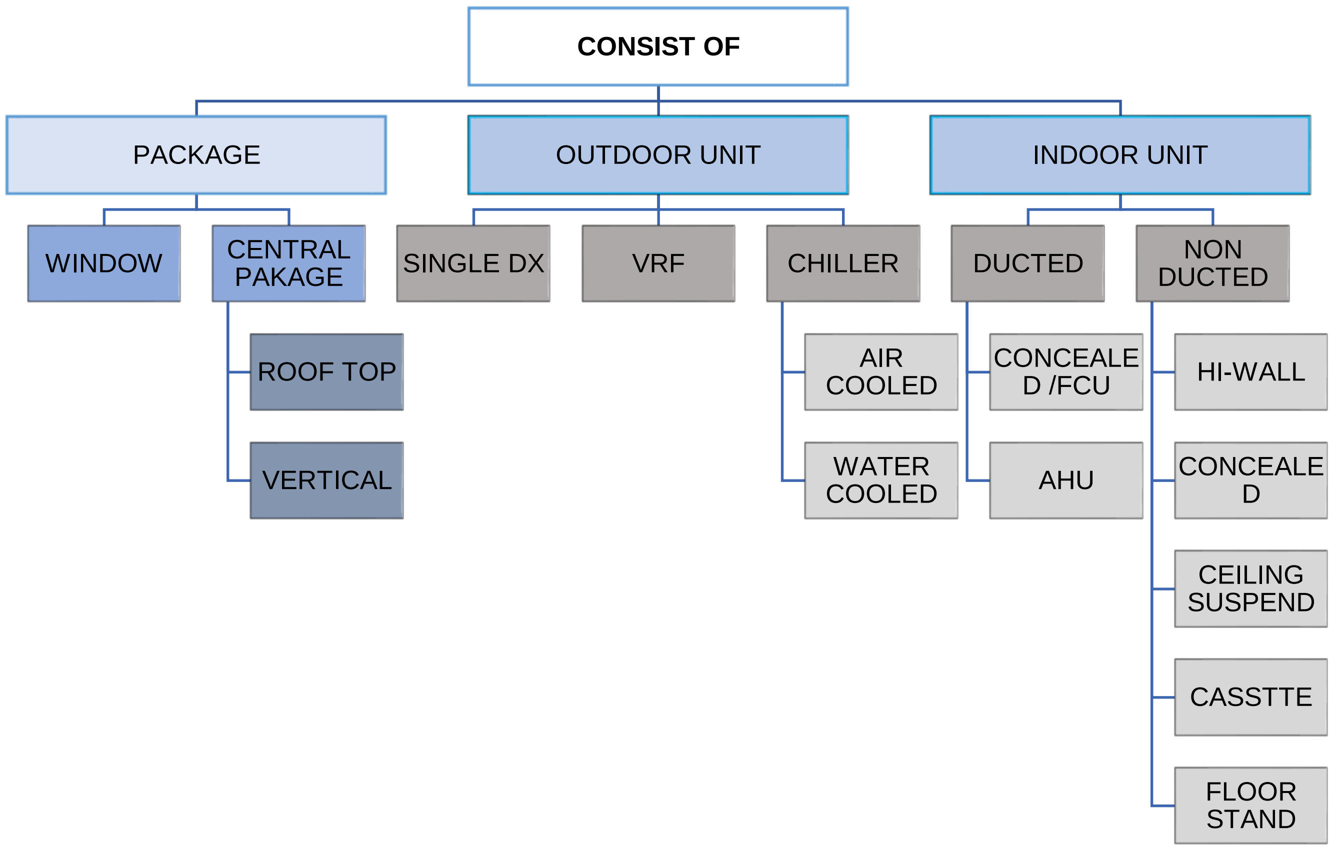


Figure 2-5 : Classification of Air Conditioning Systems in details

2.3 Package units

2.3.1 Window unit

- All component in one piece
- Cooling Capacity Up To 4 TR
- Easy To Install But there is a noise
- Easy To Maintenance
- Don’t need ducts

Figure 2-6 : Window Unit

2.3.2 Central packaged unit

→ The central package can classify in (Figure 2-7)

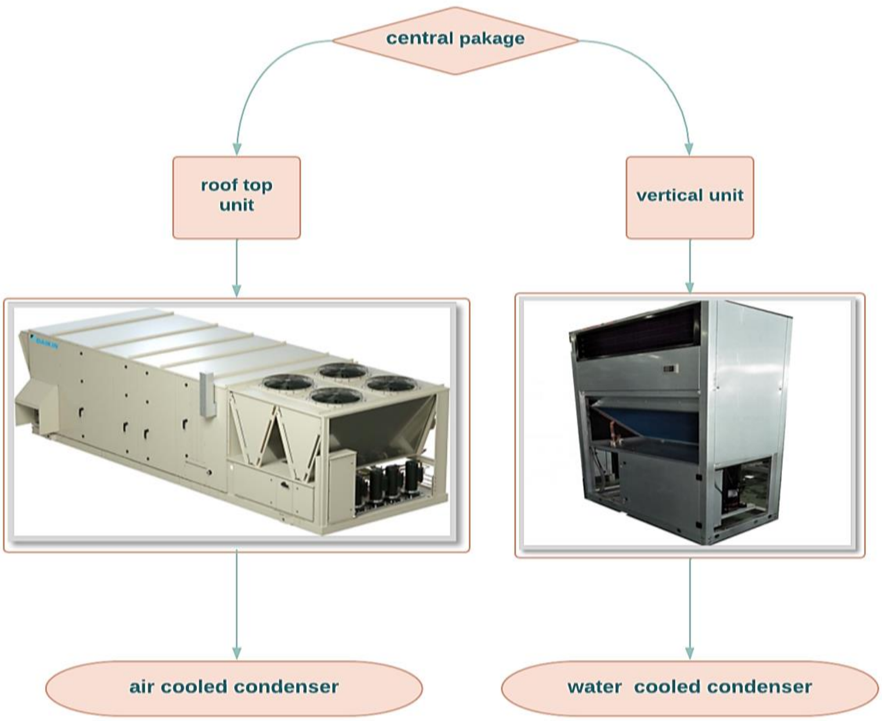


Figure 2-7 : roof top unit vs vertical unit

2.4 Outdoor unit

- → The outdoor unit, which is a (compressor expansion valve - condenser), which is to cool the evaporator coil
- → The outdoor unit could be (single DX or VRF or Chiller) Shown in (Figure 2-8)

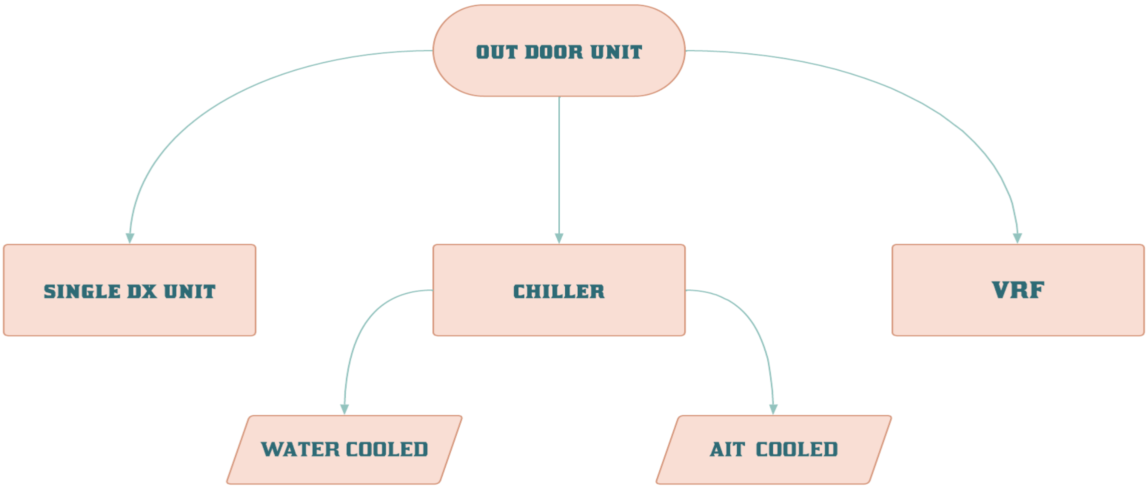


Figure 2-8 : outdoor units

2.4.1 Single Dx unit

- → Thermal zone: A thermal zone is referred to a space or group of spaces within a building with similar heating and cooling requirements. Each thermal zone must be 'separately controlled' if conditions conducive to comfort are to be provided by an HVAC system.
- → Any area that requires different temperature, humidity and filtration needs shall be categorized as an independent zone
- → One zone equal one thermostat
- → each indoor unit has outdoor unit and Preferably use in single zone

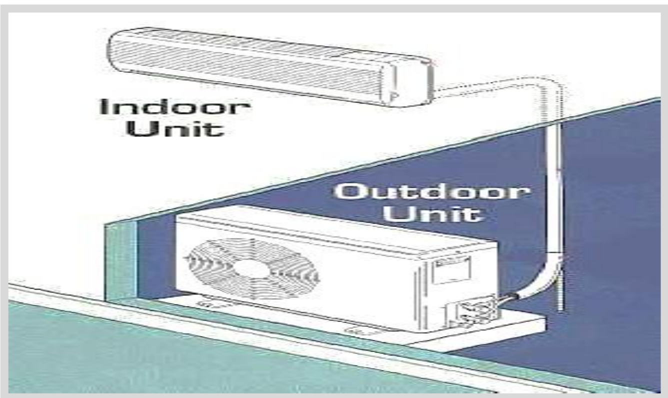


Figure 2-9 :Single Dx unit

2.4.2 Variable refrigerant flow (VRF)

→ Variable refrigerant flow (VRF) is an air-condition system configuration where there is one outdoor condensing unit and multiple indoor units.

→ The term variable refrigerant flow refers to the ability of the system to control the amount of refrigerant flowing to the multiple evaporators (indoor units), enabling the use of many evaporators of differing capacities and configurations connected to a single condensing unit.

→ The control is achieved by continually varying the flow of refrigerant through an electronic expansion valve

→ indoor units are linked by a control wire to the outdoor unit which responds to the demand from the indoor units by varying its compressor speed to match the total cooling requirements.

→ A separation tube has 2 branches and the separation tube can be used for branches.

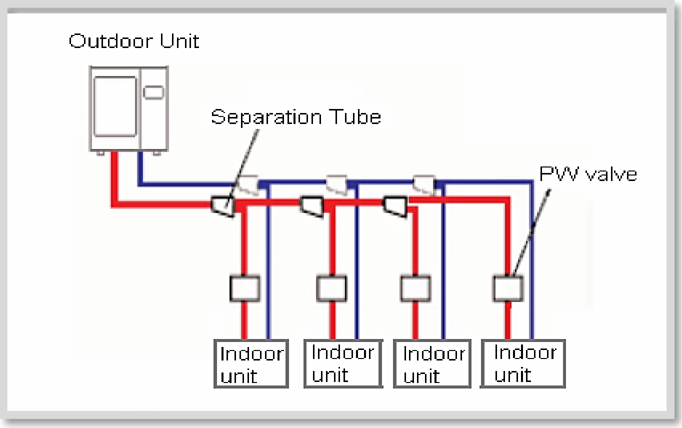


Figure 2-10 : Variable Refrigerant Flow system

→ Variable refrigerant flow (VRF) systems operate differently. Compressors in VRF systems are driven by an inverter. When loads in the system decrease, the inverter slows the compressor until its output matches the load, eliminating the frequent cycling common in conventional systems. Eliminating the cycling improves part load operating efficiency and extends the service life of the compressor shown in (Figure 2-11)

→ VRF systems use R-410A refrigerant this is non-toxic and non-flammable (Environmental Protection)

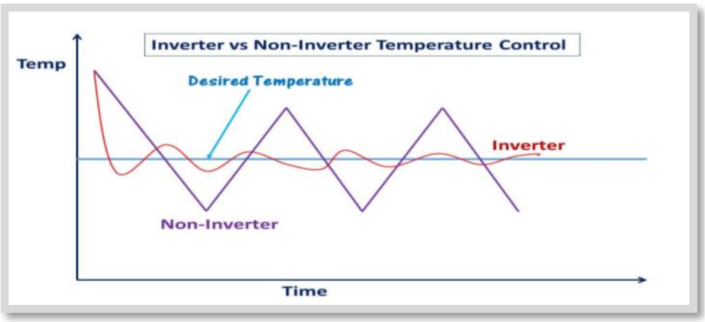


Figure 2-11 : inverter and non-inverter system

2.4.3 Chiller system

2.4.3.1 How Does a Chiller Work?

→ process begins with the primary return bringing warm water to the chiller. Heat is transferred in the evaporator to the refrigerant, and the water runs through the primary supply to the cooled space. The refrigerant moves through the compressor to raise the pressure and temperature, and then it reaches the condenser.

→ The evaporator cools the chilled water and then goes to the AHU and then the chilled water cools the mixing air shown in (Figure 2-12)

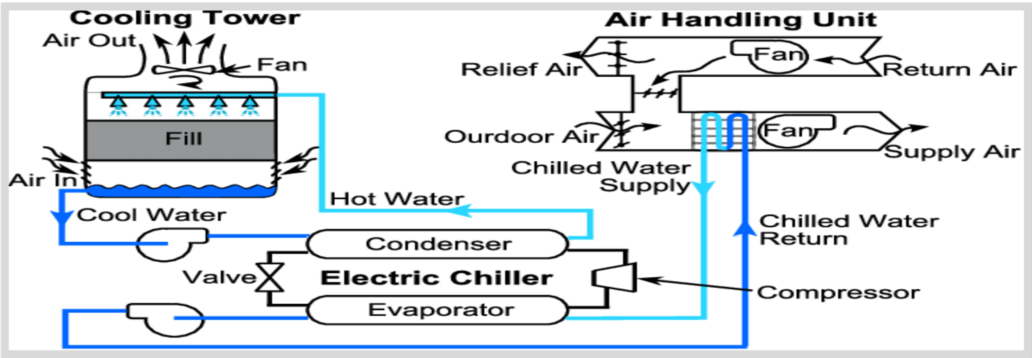


Figure 2-12 : working of chiller system

2.4.3.2 Different between air cooled condenser and water cooled

There are two types of chillers shown in (Figure 2-13): -

1) water-cooled
2) air-cooled

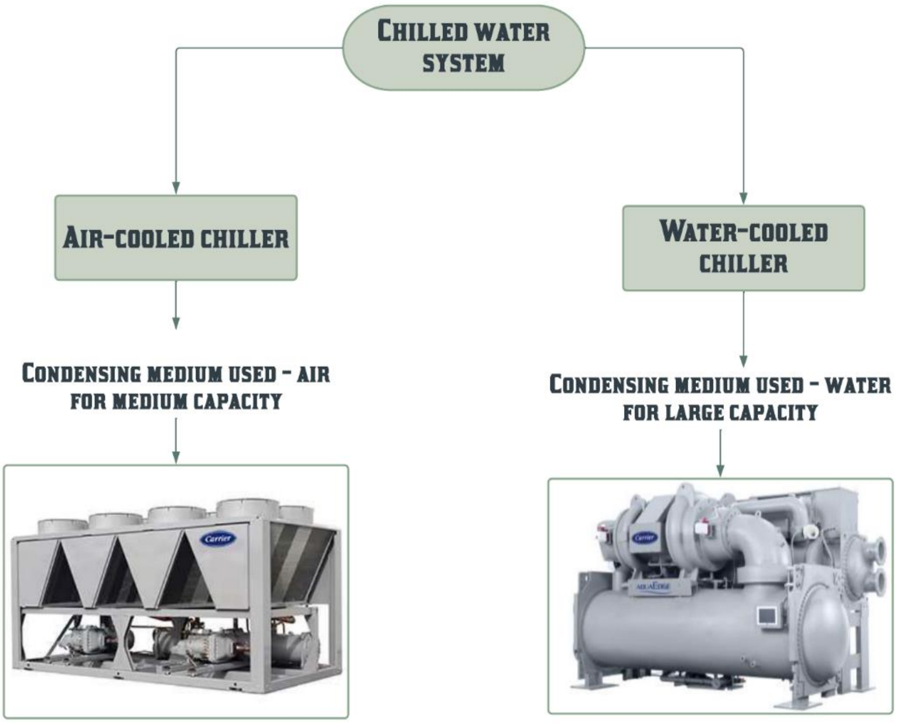


Figure 2-13 : air cooled condenser vs water cooled

2.4.4 How to selection the outdoor unit

The selection of the outdoor unit based on :

1) Total project load
2) Variable load proportion

2.5 indoor unit

The selection of indoor unit based on application such as medical application required total fresh air then The best choice is AHU and so on shown in (Figure 2-14)

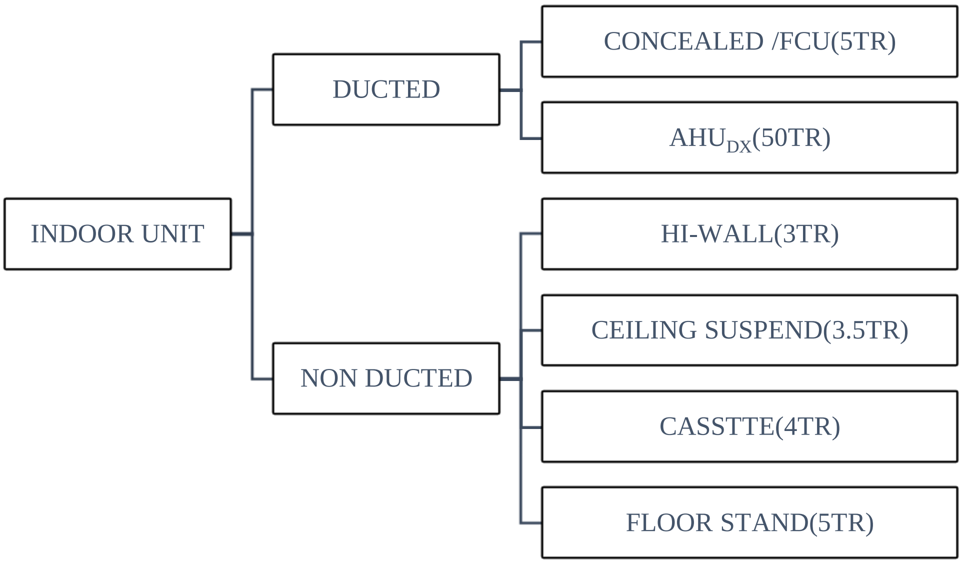


Figure 2-14 : classification of indoor unit

→ The different type of indoor unit shown in (Figure 2-15)

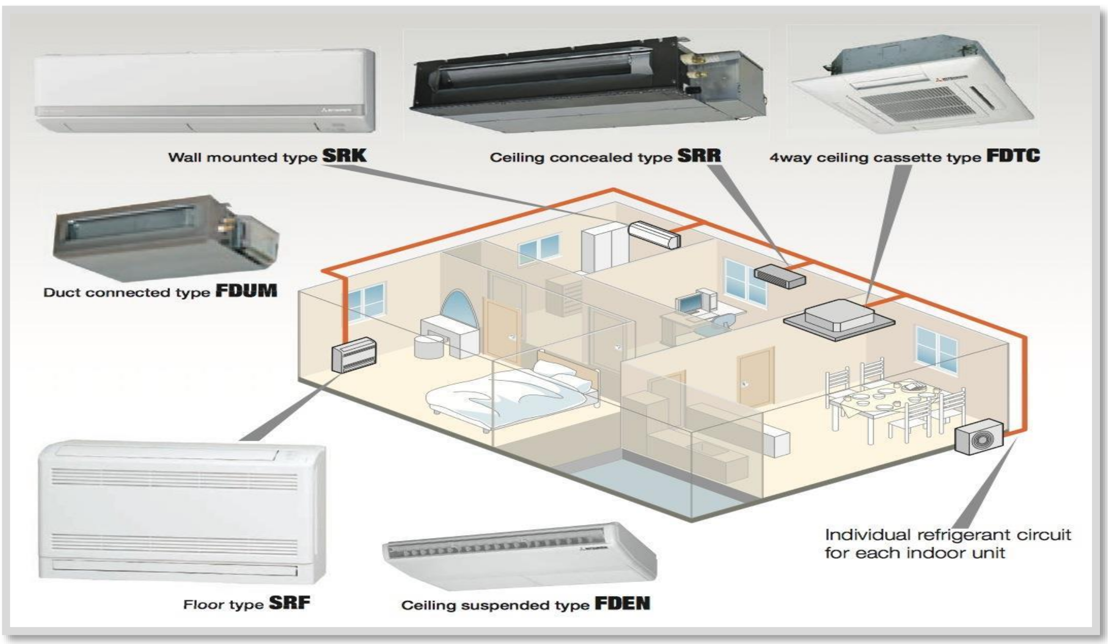


Figure 2-15 : type of indoor unit

2.5.1 Air handling unit AHU

An Air Handling Unit or AHU is a device that can handle, regulate, circulate, and conditioning the Air of an HVAC System shown in (Figure 2-16)

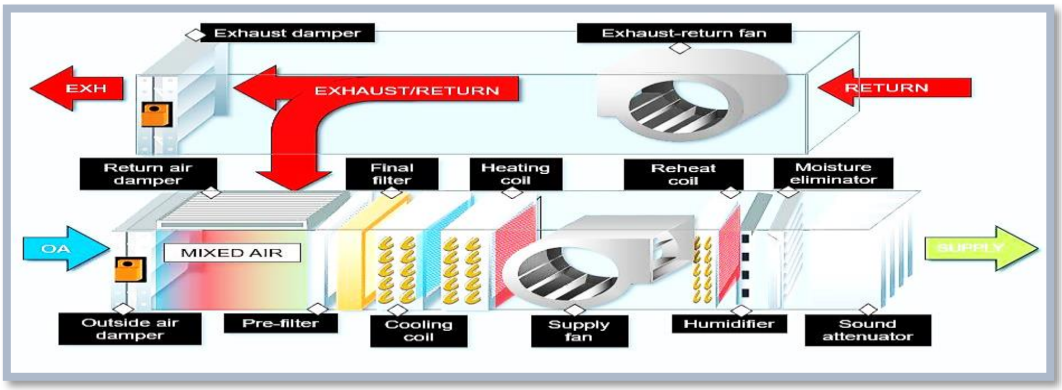


Figure 2-16 : Air Handling Unit

2.5.1.1 components of air handling unit

1) Duct

There are three ducts are there two for input air (outside air + inside air) and another for output air supply. Ducts are designed with proper shape and size according to airflow volume

2) Dampers

The main function of the damper is to start or stop the flow of air. Using a damper, the flow of air can be controlled according to requirement.

3) Mixing Section

It is a place inside an AHU where the outdoor air and indoor or returned air are mixed at a certain ratio

4) Filters

Filter in an AHU has a very important role. The filter removes the dust particle and other contamination from the air

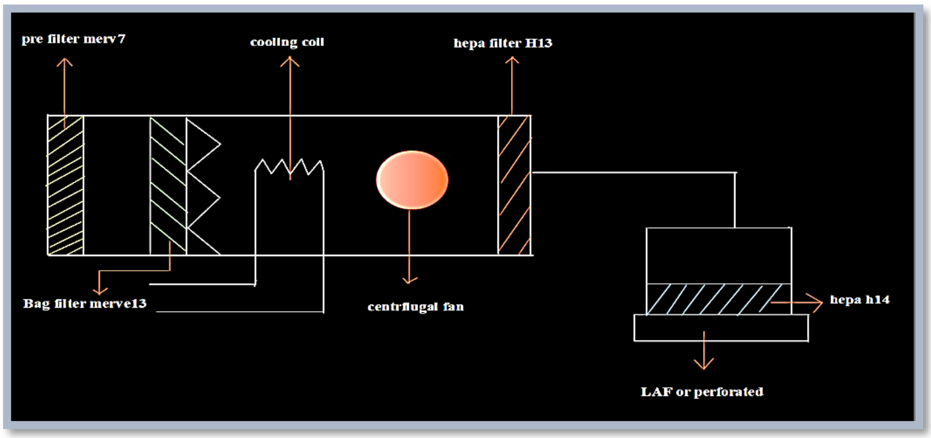


Figure 2-17 : filters in AHU

→ Minimum Efficiency Reporting Values, or MERVs, report a filter's ability to capture larger particles

→ This value of merv is helpful in comparing the performance of different filters

- Pre filter → merv→7
- Bag filter → merv→8:13
- Hepa → merv→14:17
- hepa H13→ Merv 14
- hepaH14→ Merv 17

5) Coils

As per the above diagram, there are three types of coils shown in (Figure 2-16)

1. Preheating coil
2. Cooling coil
3. Reheating Coil.

Generally, AHUs are connected to central cooling(chiller) system. So central cooling system or chiller produces cold water and it circulated through the cooling coil of the AHU. Thus the airflow through the AHU got cooled.

2.5.2 Fan coil unit

→ Component of fan coil unit shown in (Figure 2-18) is:

1) Pre-filter
2) Fan section
3) Cooling coil

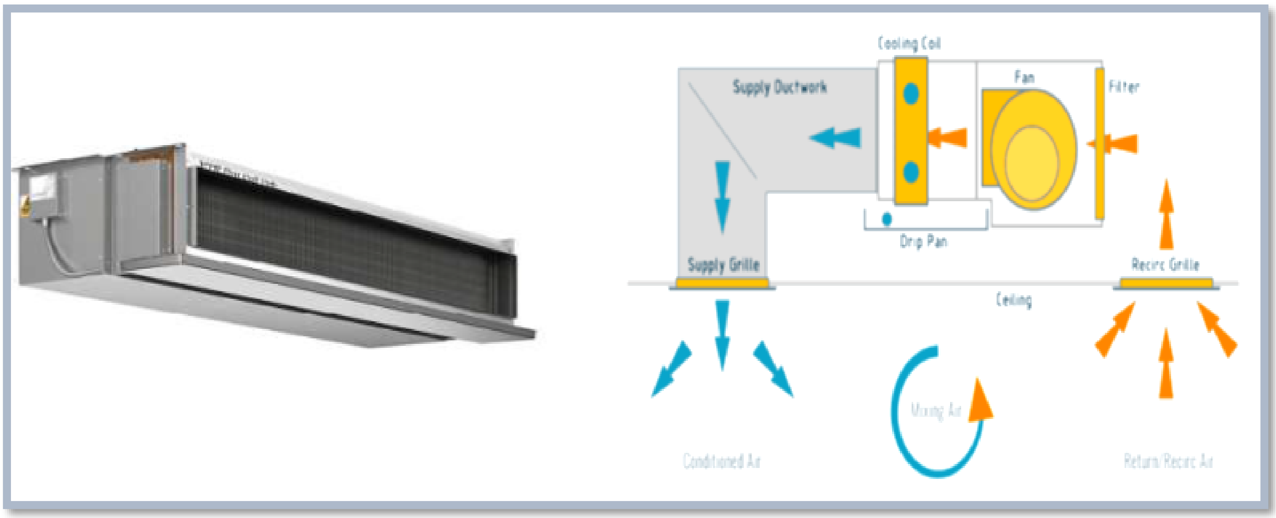


Figure 2-18 : fan coil unit

2.6 Air conditioning cycles

There are different types of arrangement:

1) **Mixing Arrangement:** room return air is mixed with minimum or optimum predetermined amount of outside fresh air.

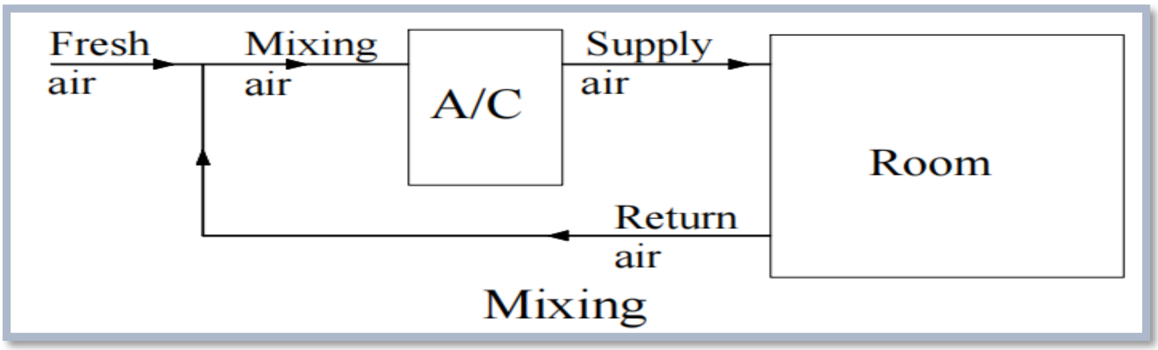


Figure 2-19 : Mixing Arrangement system

2) **All Fresh (100% F.A.) Arrangement: all outside fresh air is used. (More energy consumption)**

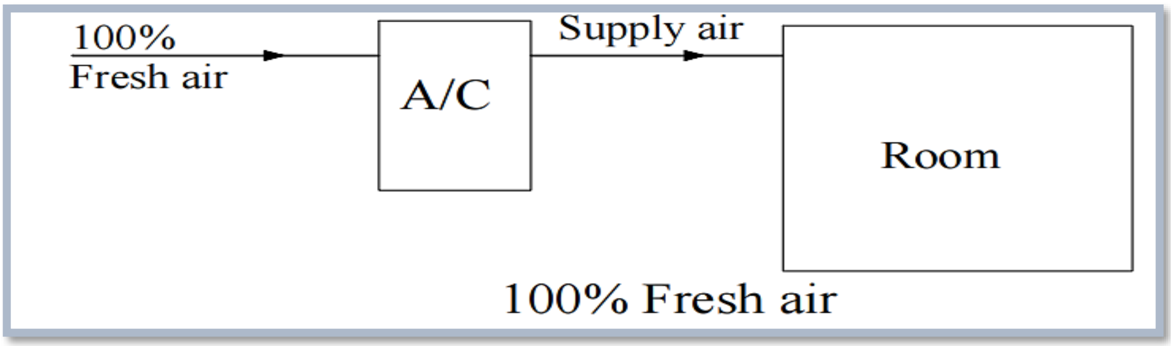


Figure 2-20 : All Fresh Arrangement system

3) **All Return Arrangement:** (less energy consumption - bad for air change and ventilation)

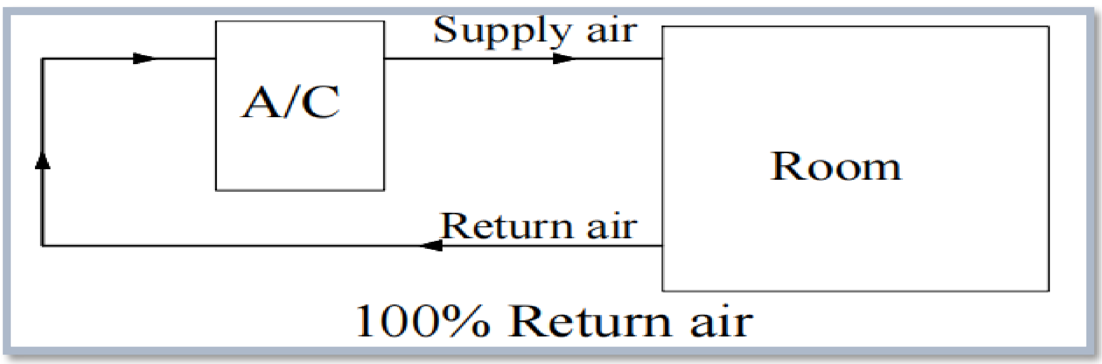


Figure 2-21 :All Return Arrangement system

System using Cooling and Dehumidification coil with mixing between recirculating and outside fresh air

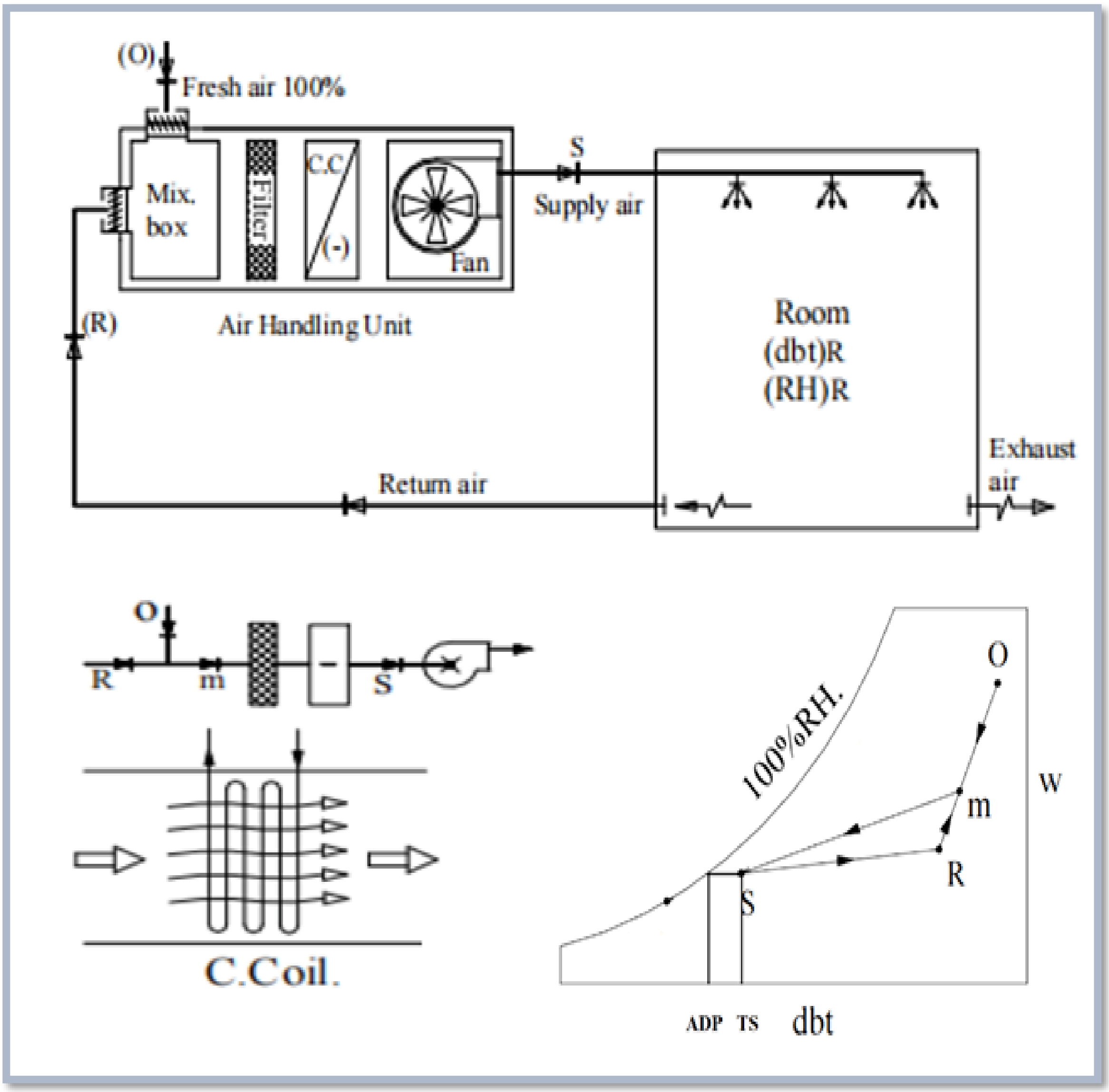


Figure 2-22 : Cooling and Dehumidification mixing system

→ Return air from the conditioned space (R) is mixed with outdoor air at point (O) in the mixing box. The mixture air (m) is the then cooled and dehumidified by passing over the cooling coil which leaves it at point (s)

→ The conditioned air (S) flows through the supply fan, and then enters to the conditioned space through the supply outlets.

Cycle calculations

Room load (S→R)

- Room supply air $(\dot{m}_s) = \dot{m}_R + \dot{m}_O$
- Mixing ratio $= \left(\frac{\dot{m}_0}{\dot{m}_s}\right)$
- Room total heat $=$ Room sensible heat $+$ room latent heat $= \dot{m}_s(h_R - h_S)$

Outside air load(M→R)

- outside total heat $= \dot{m}_m(h_m - h_R)$

Grand load(M→ADP)

- Grand total heat $= \dot{m}_s(h_m - h_{ADP})$

Summer air conditioning system using cooling and dehumidification and 100% outside fresh air

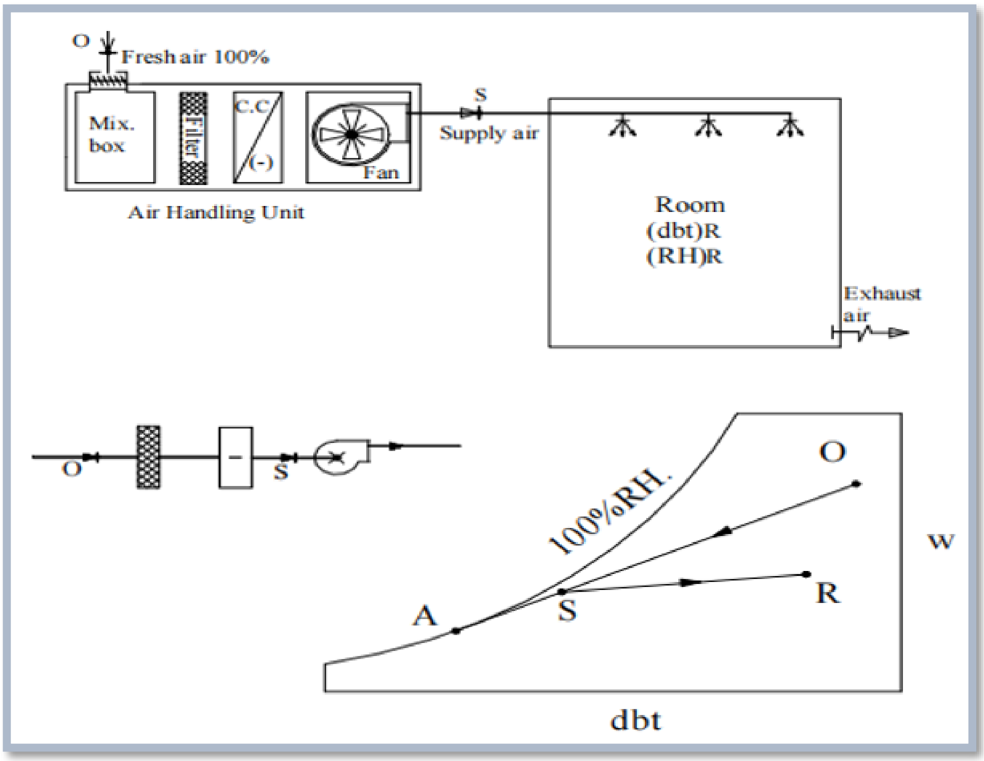


Figure 2-23 : cooling and dehumidification and 100% outside fresh air

→ In this system 100 % outside fresh air is supplied to the air conditioning to be cooled and dehumidifying before supplied to the conditioning space (room)

→ no mixing occurs as the air is exhausted to the outside.

→ This system applied for special cases like surgical rooms, pharmaceutical industry, and some food industries

Cycle calculations

Room load (S→R)

- Room supply air $(\dot{m}_s) = \dot{m}_O$
- Room total heat $=$ Room sensible heat $+$ room latent heat $= \dot{m}_s(h_R - h_S)$

Outside air load(O→R)

- outside total heat $= \dot{m}_S(h_o - h_R)$

Grand load(O→ADP)

- Grand total heat $= \dot{m}_s(h_o - h_{ADP})$

chapter (3): Ventilation systems

3.1 Ventilation definition

Ventilation: It is the process of renewing or changing the air and replacing it with fresh, fresh air inside the space to be ventilated.

Here the characteristics of the air are not changed, but is replaced by natural air with the same temperatures as the outside air. (That is, we remove the air in the place and introduce fresh natural air instead).

3.2 Purpose of ventilation

1) Removing unpleasant odors from the place (such as removing odors in the bathroom or kitchen)
2) There is a lack of oxygen as a result of crowding in the place.
3) Removing the heat load in the place (such as engine rooms and pump rooms).

3.3 Types of ventilation

3.3.1 Natural ventilation

- Natural ventilation is the process of pulling fresh air into a building from the outside, in turn This becomes done, without mechanical assistance
- We rely here on the air that is leaked through the doors and windows, i.e. (there is no interference here on our part) shown in (Figure 3-1)

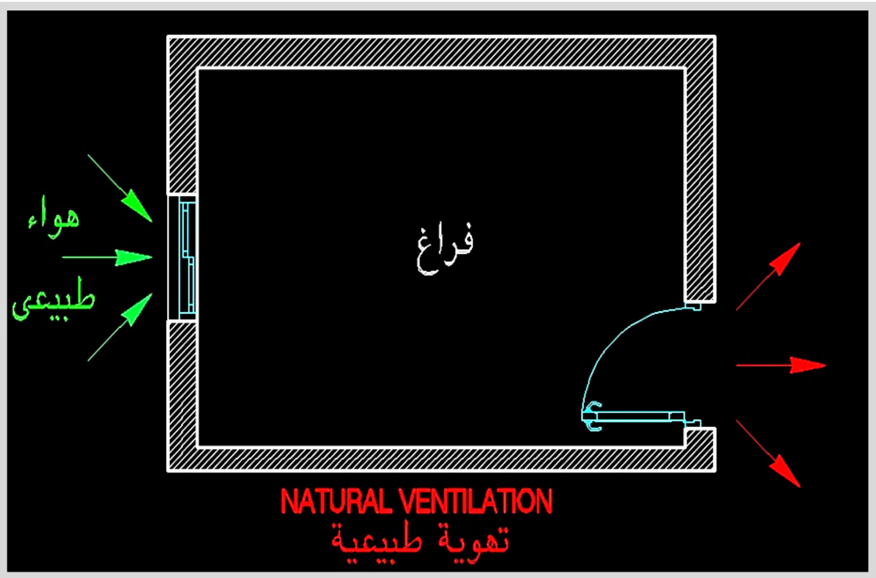


Figure 3-1 : Natural ventilation

3.3.2 Mechanical ventilation

We draw the air in the place and expel it outside through exhaust fans. This expelled air is compensated by introducing fresh air from the outside, and this compensation is also done through one of the two methods:

3.3.2.1 Natural supply

Air enters through openings in the place such as the door, window, or any other openings shown in (Figure 3-2)

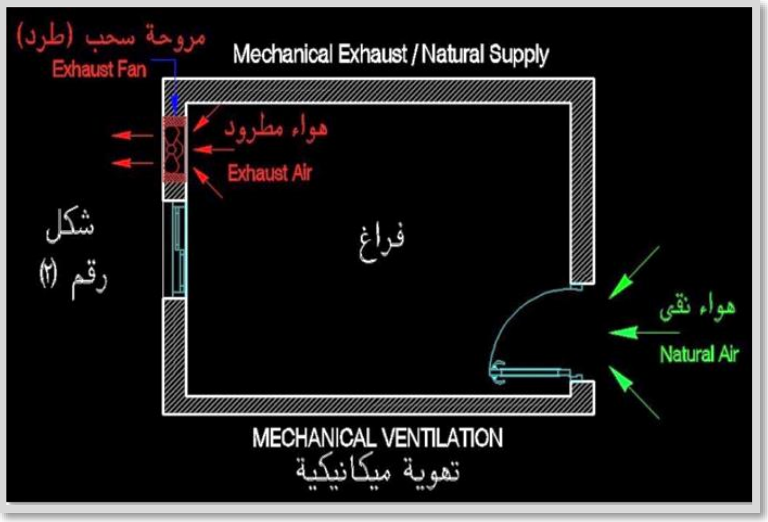


Figure 3-2 : mechanical ventilation by using natural supply

3.3.2.2 Mechanical supply

By installing supply fans, they draw fresh air from outside and enter it to the place to be ventilated shown in (Figure 3-3)

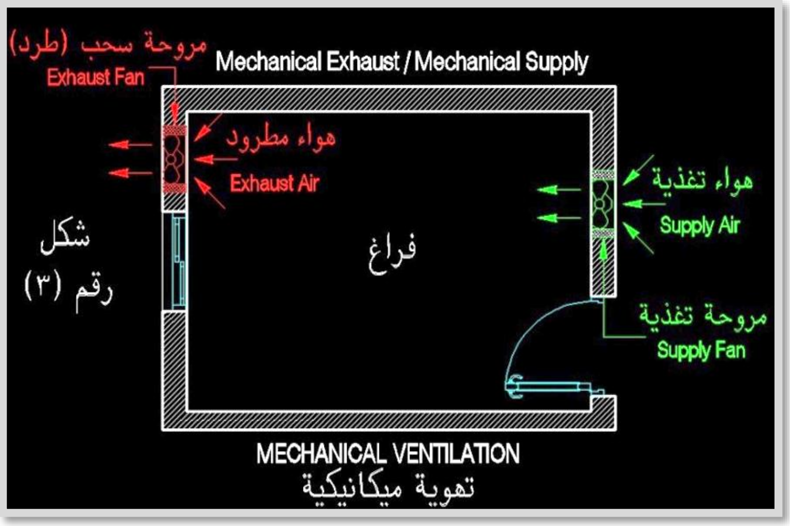


Figure 3-3 : mechanical ventilation by using mechanical supply

→ The most important areas that need ventilation

(Kitchens - bathrooms - garages - warehouses - electrical rooms - transformer rooms - generator rooms)

3.3.2.3 forms of mechanical ventilation

Ventilation is the process of changing and replacing the air with fresh, fresh air (expelling the air in the place, and introducing fresh air in its place), and this is done in one of the ways:

1) Mechanical ventilation using exhaust fans only, and natural air entry (such as bathrooms and kitchens).
2) Mechanical ventilation using only supply fans (for air intake), and natural air exit (such as elevators).
3) Mechanical ventilation using exhaust fans and supply fans (in some cases in garages).

3.4 Negative Pressure and Positive Pressure

3.4.1 Negative pressure

- For example, the value of the exhaust air is (1000 cfm) and the value of the supply air is (850 cfm), meaning that the exhaust air (outside) has a greater value than the supply air, and here a balance must occur between the amount of air entering and leaving to compensate for this difference.
- To compensate for this difference, the air enters naturally from the surrounding areas of the vacuum through the (under the door), and here it is called negative pressure inside this place shown in (Figure 3-4)

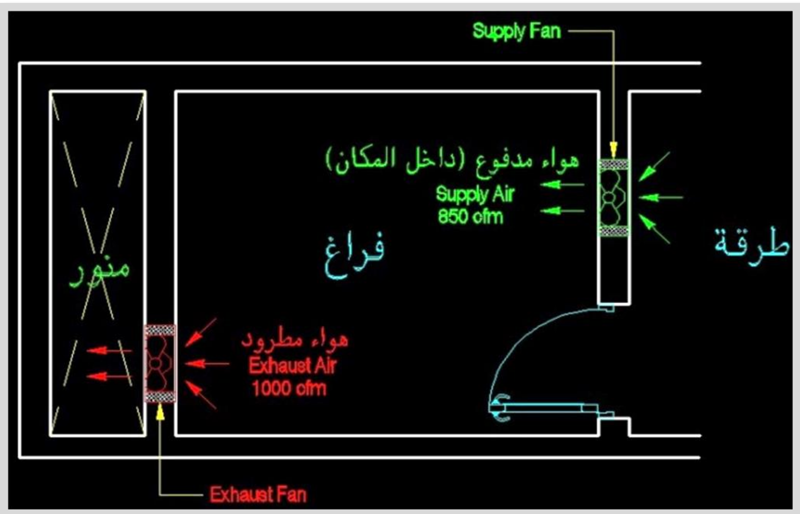


Figure 3-4 : Negative pressure room

3.4.2 Positive pressure

- For example, the value of the exhaust air is (1000 cfm) and the value of the supply air is (1150 cfm), meaning that the exhaust air (outside) has a smaller value than the supply air, and here a balance must occur between the amount of air entering and leaving to compensate for this difference.
- excess air escapes from the door to the outside and here it is called positive pressure inside this place (the value of the air supply is greater than the air exhaust) shown in (Figure 3-5)

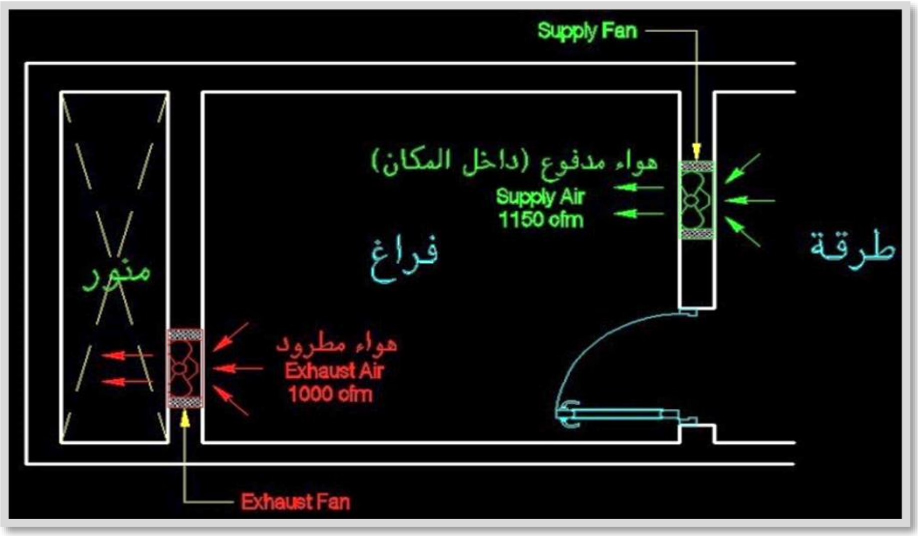


Figure 3-5 : positive pressure room

3.5 Ventilation Calculation

1) Exhaust air

We calculate the amount of air drawn (Exhaust Air) from the place to be ventilated, through the rate of air change per hour for the place to be ventilated (we will change the air how many times per hour) through the following equation:

$$CFM_{exhaust} = \frac{ACH * Area * Height}{1.7}$$

Where:

ACH → air change per hour It is obtained from (Figure 3-6)

Area → The area of the place to be ventilated in square metres (m^2)

Height →The height of the place up to the ceiling to be ventilated in metres (m)

→ Air change per hour of some application shown in (Figure 3-6)

عدد مرات تغيير الهواء في الساعة (ACH)	التطبيق (Application)
10 to 12	حمام (Toilet)
15 to 20	مطبخ خاص (Domestic Kitchen)
20 to 30	مطبخ تجاري (Commercial Kitchen)
4 to 6	مخازن (Stores)
4 to 6	جراجات (Car Park) الحالة العادية
10	جراجات (Car Park) في حالة حدوث حريق

Figure 3-6 : Air change per hour of some application

2) Fresh air

Here, the air is compensated by entering it in a mechanical way by installing supply fans. Here we calculate the value of this air through the following equations

- Pressure of the place to be ventilated (negative):

$$\text{CFM Supply or Fresh} = (0.85 : 0.90) \text{ x CFM Exhaust}$$

- Pressure of the place to be ventilated (positive):

$$\text{CFM Supply or Fresh} = (1.10 : 1.15) \text{ x CFM Exhaust}$$

3.6 Car Park Ventilation

- Due to the large size of the garages compared to bathrooms and kitchens, the garage is ventilated by installing fan and duct s and air outlets on it
- Garage pressure is negative
- The air outlets are distributed on two levels until it is ensured that the air is completely withdrawn from the garage
- We make half of the exits at a high level near the ceiling (high level) And the other half we pull it from a low level (low level) Preferably at a height (60cm :80cm) from the ground shown in (Figure 3-7)

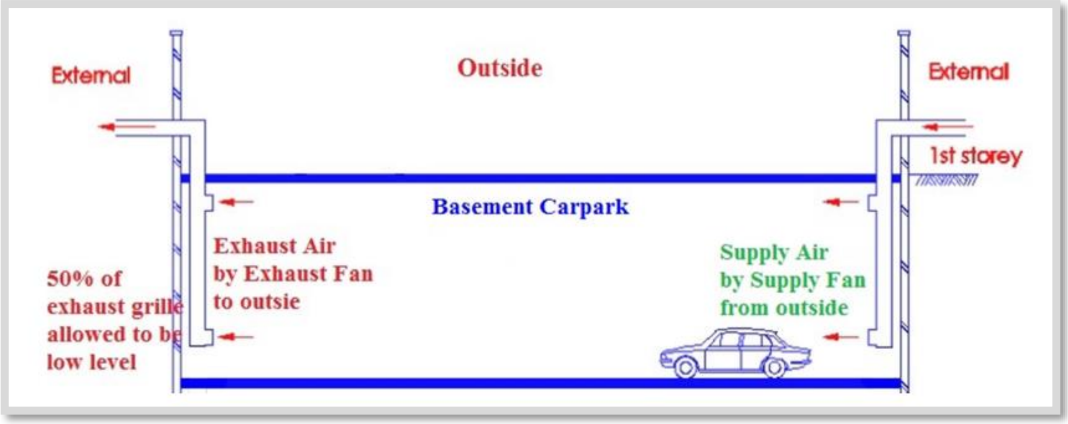


Figure 3-7 : Car Park Ventilation

3.6.1 Fire Mode

- In many cases, it is among the requirements for ventilation of the garages to calculate the ventilation in the event of a fire, and here the ventilation is calculated in the normal case and the case of the fire, and variable speed fans are installed that work in both cases and this is done as follows:
    - Calculate the value of the Exhaust Air at the value of (ACH=6) in the case of normal ventilation, and calculate it again at the value of (ACH=10) in the event of a fire shown in (Figure 3-6)
    - A variable speed fan is selected that operates at the two required speeds, so the fan always runs at the slow speed and draws the amount of air calculated at (ACH=6) and we call this state (Normal Mode)
    - (Smoke Detector) is installed, and when a fire occurs, this (Detector) gives a signal to the fan, so it changes its speed and works at its high speed (High Speed) and draws the amount of air calculated at (10 ACH) and we call this case (Fire Mode)
    - The same previous steps are also done for (Supply Air), and its value is calculated at negative pressure with a value of (85% of cfm exhaust) for both the normal state and the fire state

3.7 Central Kitchen Ventilation

3.7.1 Purpose of hood

The hood is installed in the kitchens above the cooking platform in order to capture the heat generated by the cooking process, as well as to capture grease, combustion products, fumes, smoke, odors and steam, and then expel them outside the kitchen boundaries

3.7.2 Hood types

1) Wall mounted canopy

It is a hood that is installed on the wall above the cooking platform shown in (Figure 3-8)

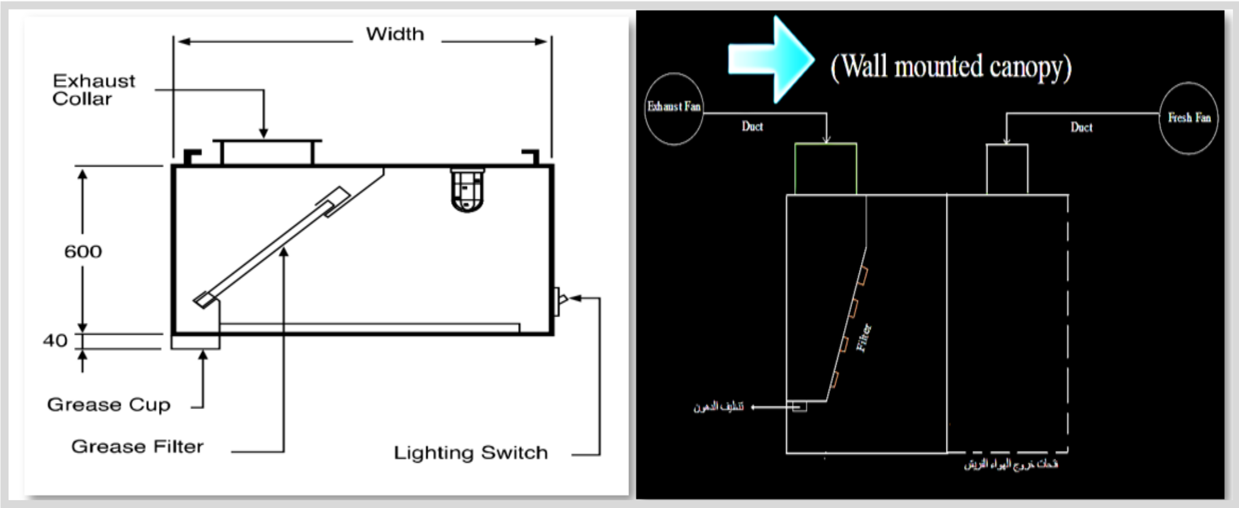


Figure 3-8 : Wall mounted canopy

- air flow pattern of wall mounted shown in (Figure 3-9)

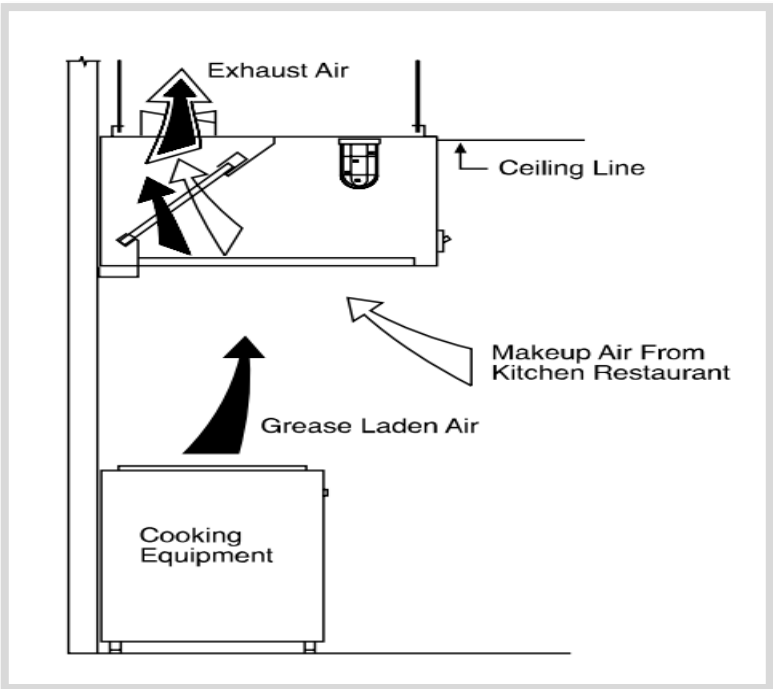


Figure 3-9 : air flow pattern of wall mounted

2) Single island canopy

This type is installed above the cooking platform if it is a single line of appliances, and it is usually open from all sides, meaning there are no walls for the hood shown in (Figure 3-10)

3) Double island canopy

This type is installed above the cooking platform if it is two lines of appliances so that the back end of each line meets each other, and it is usually open from all sides, meaning there are no walls for the hood shown in (Figure 3-10)

4) Back shelf non-canopy

Like a wall mounted, but the difference is that it can't be used other than that it's exhaust only shown in (Figure 3-10)

5) Eyebrow non-canopy

It is installed directly above the oven, exhaust only shown in (Figure 3-10)

6) Pass-Over

It is a hood similar to the back shelf, but its height is low shown in (Figure 3-10)

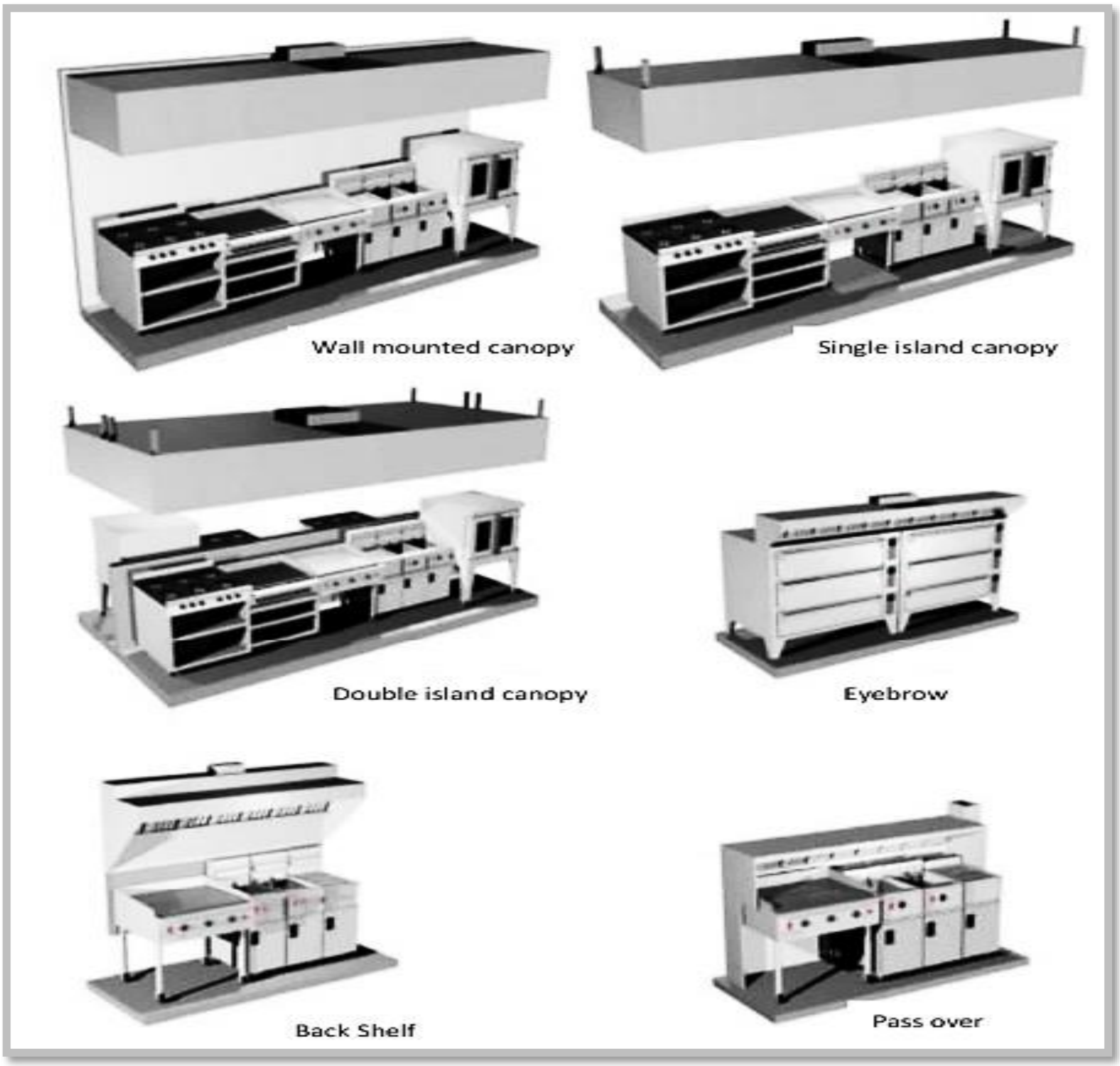


Figure 3-10 :different types of hood

3.7.3 Hood design steps

1) Calculating the dimensions of the hood (length - width - high)
2) Calculation of (cfm fresh – cfm exhaust)
3) The number of fresh and exhaust holes and the area of each hole

→ Very important notes

- Hood height from 50:60 cm
- Do not try to reach the length of the hood more than 4 meters because you will find it difficult to move it

3.7.4 Case study on hood design

→ A hot line consisting of the following shown in (Figure 3-11):

1) Hot top (90*90) cm
2) Griddle (60*90) cm
3) Fryer (60*60) cm

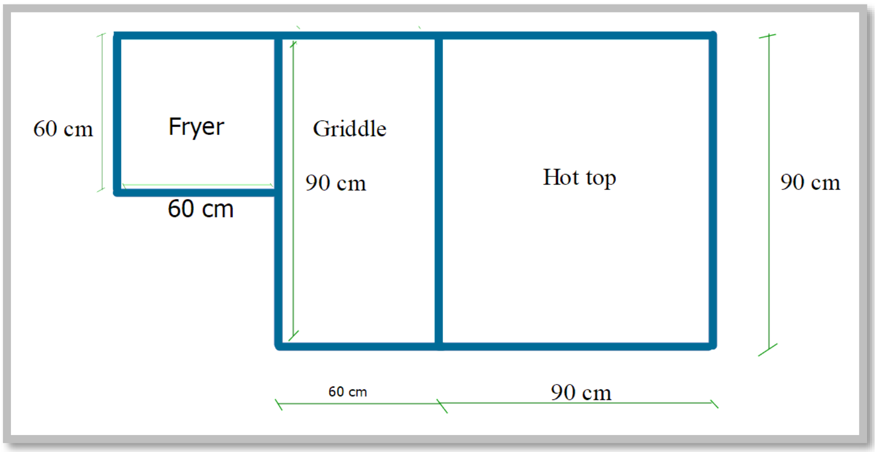


Figure 3-11 : hot line (hot top – griddle – fryer)

Solution

hood (wall mounted canopy)

$\text{Length of equipment's} = 90 + 60 + 60 = 210\text{cm}$

$\text{width of equipment's} = 90 \text{ or } 60 = 90\text{cm (largest width)}$

→ $\text{hood length} = \text{Length of equipment's} + \text{space (1)} = 210 + \text{space (1)}$

→ $\text{hood width} = \text{width of equipment's} + \text{space (2)} = 90 + \text{space (2)}$

→ $\text{hood high} = 40{:}\,60 \text{ cm} = 50\text{cm}$

space (1) and space (2) as shown in (Figure 3-12)

Figure 3-12 : hood space

From (Figure 3-13)

→ space (1) $= (20 + 20) = 40\text{cm}$

→ space (2) $= 30\text{cm}$

Type of Hood	End Overhang	Front Overhang	Rear Overhang
Wall-mounted canopy	6 in. to 12 in. (15-30 cm)	12 in (30 cm)	--
Single island canopy	6 in. to 12 in. (15-30 cm)	12 in. (30 cm)	12 in. (30 cm)
Double island canopy	6 in. to 12 in. (15-30 cm)	12 in. (30 cm)	12 in. (30 cm)
Eyebrow	0 in.	12 in. (30 cm)	--
Backshelf or Pass-over	0 to 3 in. (72 mm)	--	Front set back 6 in. to 12 in. (15-30 cm) in from the front of the cooking surface

الجدول 1.17 أبعاد الهود المطلوبة

Figure 3-13 : dimension of hood

→ hood length $= 210 + \text{space}\ (1) = 210 + (20 + 20) = 250\text{cm}$

→ hood width $= 90 + \text{space}\ (2) = 90 + 30 = 120\text{cm}$

→ hood high $= 40{:}\ 60\ \text{cm} = 50\text{cm}$

CFM exhaust

$$\text{CFMexhaust} = \text{hood length(ft)} * \frac{\text{cfm}}{\text{ft}} * \text{safety factor}$$

- safety factor according to hood type from (Figure 3-14)

Correction Factor According to Hood Type	
Wall Mounted Canopy	1
Single Island Canopy	1.2
Double Island Canopy	1.15

Figure 3-14 : safety factor according to hood type

- $\frac{\text{cfm}}{\text{ft}}$ → Depends on the degree of seriousness of the hood, meaning the equipment under the hood produces high, medium or low temperatures From ashrae shown in (Figure 3-15) (wall mounted – medium duty)→ cfm/ft=210

Hood Type	Extra Heavy Duty		Heavy Duty		Medium Duty		Light Duty	
	cfm/ft	L/s.m	cfm/ft	L/s.m	cfm/ft	L/s.m	cfm/ft	L/s.m
Wall-mounted canopy	385	597	280	434	210	325	140	217
Single island canopy	490	760	420	651	350	542	280	434
Double island canopy (per side)	385	597	280	434	210	325	175	271
Eyebrow Non-canopy	N/A	N/A	N/A	N/A	175	271	175	271
Backshelf / pass-over	N/A	N/A	280	434	210	325	210	325

الجدول 3.17 قيمة تدفق هواء الطرد لواحدة الطول وفق ASHRAE 90.1

Figure 3-15 : exhaust air flow value per unit length

$$\rightarrow \text{cfm}_{\text{exhaust}} = \text{hood length(ft)} * \frac{\text{cfm}}{\text{ft}} * \text{safety factor}$$

$$\rightarrow \text{cfm}_{\text{exhaust}} = 2.5 * 3.28 * 210 * 1 = 1722 \text{ cfm}$$

CFM fresh

CFM Fresh = (0.85 : 0.90) x CFM Exhaust

$$\rightarrow \text{cfm}_{\text{fresh}} = 0.9 * 1722 \text{ cfm} = 1550 \text{ cfm}$$

Number and area of fresh and exhaust air holes

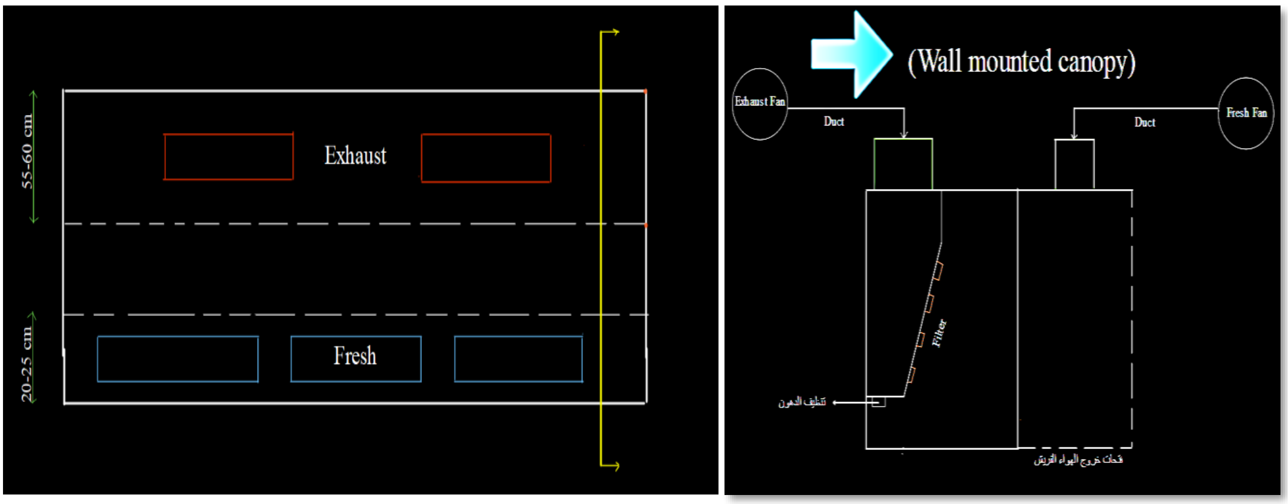


Figure 3-16 : Number and area of fresh and exhaust air holes exhaust

$$\text{cfm for one open} = \frac{1722}{2} = 861\ \text{cfm}$$

$$A = \frac{Q}{V} = \frac{861}{(800:900)} = 1.07\text{ft}^2 = 0.1\ \text{m}^2$$

$$A = y * x$$

Assume $y = 0.2\text{m}$

then $x = 0.5\ \text{m}$

Fresh

$$\text{cfm for one open} = \frac{1550}{3} = 515\text{cfm}$$

$$A = \frac{Q}{V} = \frac{515}{(800:900)} = 0.6\text{ft}^2 = 0.06\text{m}^2$$

$$A = y * x$$

Assume $y = 0.15\text{m}$

then $x = 0.4\text{m}$

chapter (4): Load calculation

- The first step in designing air conditioning for any place is that we calculate the loads in the place

The cooling load for a space can be made up of many components, including:

1) Heat conduction through exterior walls and roofs (transmission load)
2) Solar radiation through transparent surfaces such as windows
3) Heat conduction through interior partitions, ceilings and floors
4) Heat generated within the space by occupants, lights, appliances, equipment and processes
5) Loads as a result of ventilation and infiltration of
6) Other miscellaneous heat gains

→ (Figure 4-1)indicates various components of heat gain sources.

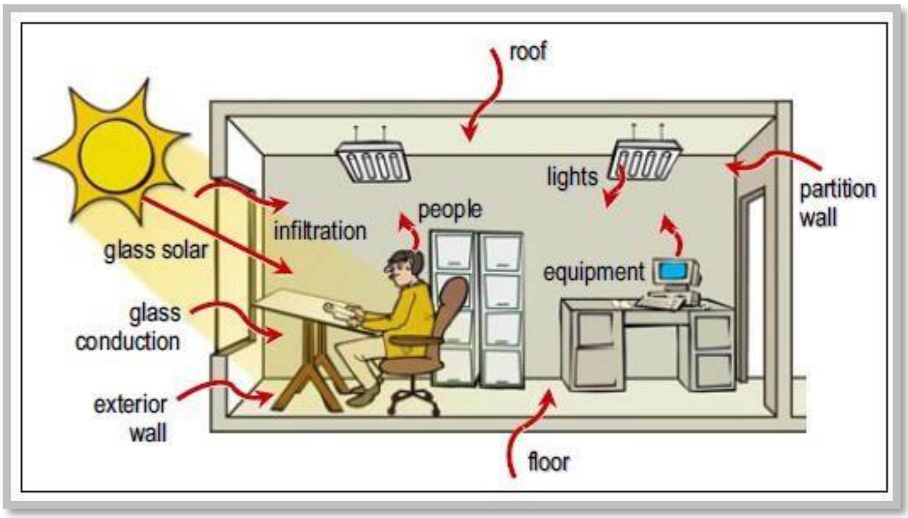


Figure 4-1 : Heat gain sources.

→ Sensible heat

Heat which a substance absorbs, and while its temperature goes up, the substance does not change state

- Heat transmitted thru floors, ceilings, walls
- Occupant's body heat
- Appliance & Light heat
- Solar Heat gain
- Infiltration of outside air
- Air introduced by Ventilation

→ Latent Heat Loads

Latent heat gain occurs when moisture is added to the space either from internal sources (e.g., vapor emitted by occupants and equipment) or from outdoor air as a result of infiltration or ventilation to maintain proper indoor air quality Latent heat load is total of :

- Moisture-laden outside air form Infiltration & Ventilation
- Occupant Respiration & Activities
- Moisture from Equipment & Appliances

→ Loads are calculated using two methods:

1) manual method
2) by using a program called Hourly Analysis Program (HAP)

4.1 load calculation manual method

Loads will be calculated as shown in the (Figure 4-2)

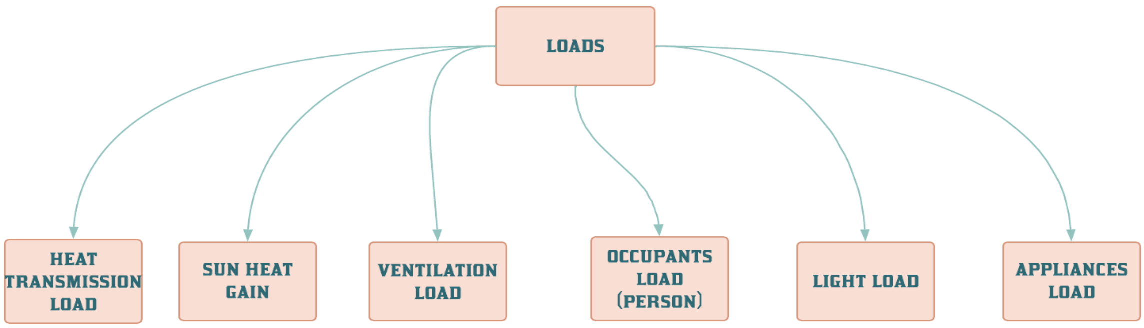


Figure 4-2 : Heat gain for space

→ first step identifies outside condition and inside condition for example the project in Cairo

- From ashrae meteo info shown in (Figure 4-3)
- Outside condition → 42 °C DBT and 26.5 °C WBT
- Inside conditions → 24 °C and 50% RH

Extreme Annual Design Conditions																
Extreme Annual WS				Extreme Annual Temperature				n-Year Return Period Values of Extreme Temperature								
				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
9.3	8.1	7.2	DB	5.5	42.0	2.3	1.7	3.8	43.2	2.4	44.2	1.1	45.1	-0.5	46.4
			WB	3.2	26.5	1.7	0.8	2.0	27.1	1.0	27.5	0.0	28.0	-1.2	28.6

Figure 4-3 : Outside condition from ashrae

4.1.1 heat transmission load (**QT**)

- It is through (walls-windows-doors-ceiling-floors) as a result of the temperature difference between indoors and outdoors
- Well, we know now that it occurs as a result of heat transfer through walls, ceiling, floor, door and windows, and it is calculated through:

$$QT = UA\,(To - TI) = (W)$$

Where:

- ⇒ U: overall heat transfer coefficient (W/m2 K)
- ⇒ A: wall area m2
- ⇒ To: outside air temperature °C
- ⇒ Ti: inside air temperature °C
  - How to calculate the overall heat transfer coefficient

$$u_{wall} = \frac{1}{R_{out} + R_{in} + R_1 + R_2}$$

R_1→ Resistivity of 4 in common brick

R_2→ Resistivity of 6 in face brick

R_{in}→ inside wall resistance

R_{out}→ outside wall resistance

From (Figure 4-4**Error! Reference source not found.**) → $u_{wall} = \frac{1}{R_{out}+R_{in}+R_1+R_2} = \frac{1}{0.33297+0.68500+0.79365+0.63918}$

$$= 0.408\ \frac{Btu}{hr * ft2°F} = 2.317\ \frac{W}{m^2 \cdot K}$$

- $u_{ceiling} = \frac{1}{R_{out}+R_{in}+R_1} == \frac{1}{0.33297+0.68500+0.36} = 0.729\frac{Btu}{hr*ft2°F} = 4.141\frac{W}{m^2 \cdot K}$
  - R_1→ Resistivity of 10mm gypsum board
- $u_{floor} = \frac{1}{R_{out}+R_{in}+R_1} == \frac{1}{0.68500 + 0.68500 + 1.111} = 0.403\frac{Btu}{hr*ft2°F} = 2.288\frac{W}{m^2 \cdot K}$
  - R_1→ Resistivity of 200mm HW concrete block
  - $R_{out} = R_{in}$
- u_{window} →single glass =1.04 $\frac{Btu}{hr*ft2°F}$ =5.6 $\frac{W}{m^2 \cdot K}$

Thermal Properties and Code Numbers of Layers Used in Calculations of Coefficients for Roof and Wall

Description	Code Number	Thickness and Thermal Properties L	K	D	SH	R	WT	WT × SH
Outside surface resistance	A0					0.333		
1-in. Stucco (asbestos cement or wood siding plaster, etc.)	A1	0.0833	0.4	116	0.20	0.208	9.66	1.93
4-in. face brick (dense concrete)	A2	0.333	0.75	130	0.22	0.444	43.3	9.53
Steel siding (aluminum or other lightweight cladding)	A3	0.005	26.0	480	0.10	0.0002	2.40	0.24
Finish	A6	0.0417	0.24	78	0.26	0.174	3.25	0.85
Air space resistance	B1					0.91		
1-in. insulation	B2	0.083	0.025	2.0	0.2	3.32	0.17	0.03
2-in. insulation	B3	0.167	0.025	2.0	0.2	6.68	0.33	0.07
3-in. insulation	B4	0.25	0.025	2.0	0.2	10.03	0.50	0.10
1-in. insulation	B5	0.0833	0.025	5.7	0.2	3.33	0.47	0.10
2-in. insulation	B6	0.167	0.025	5.7	0.2	6.68	0.95	0.19
1-in. wood	B7	0.0833	0.07	37.0	0.6	1.19	3.08	1.85
2.5-in. wood	B8	0.2083	0.07	37.0	0.6	2.98	7.71	4.63
4-in. wood	B9	0.333	0.07	37.0	0.6	4.76	12.3	7.38
2-in. wood	B10	0.167	0.07	37.0	0.6	2.39	6.18	3.71
3-in. wood	B11	0.25	0.07	37.0	0.6	3.58	9.25	5.55
3-in. insulation	B12	0.25	0.025	5.7	0.2	10.0	1.42	0.28
4-in. clay tile	C1	0.333	0.33	70.0	0.2	1.01	23.3	4.66
4-in. l.w. concrete block	C2	0.333	0.22	38.0	0.2	1.51	12.7	2.54
4-in. h.w. concrete block	C3	0.333	0.47	61.0	0.2	0.71	20.3	4.06
4-in. common brick	C4	0.333	0.42	120	0.2	0.79	40.0	8.00
4-in. h.w. concrete	C5	0.333	1.0	140	0.2	0.333	46.6	9.32
8-in. clay tile	C6	0.667	0.33	70	0.2	2.02	46.7	9.34
8-in. l.w. concrete block	C7	0.667	0.33	38.0	0.2	2.02	25.4	5.08
8-in. h.w. concrete block	C8	0.667	0.6	61.0	0.2	1.11	40.7	8.14
8-in. common brick	C9	0.667	0.42	120	0.2	1.59	80.0	16.00
8-in. h.w. concrete	C10	0.667	1.0	140	0.2	0.667	93.4	18.68
12-in. h.w. concrete	C11	1.0	1.0	140	0.2	1.00	140.0	28.00
2-in. h.w. concrete	C12	0.167	1.0	140	0.2	0.167	23.4	4.68
6-in. h.w. concrete	C13	0.5	1.0	140	0.2	0.50	70.0	14.00
4-in. l.w. concrete	C14	0.333	0.1	40	0.2	3.33	13.3	2.66
6-in. l.w. concrete	C15	0.5	0.1	40	0.2	5.0	20.0	4.00
8-in. l.w. concrete	C16	0.667	0.1	40	0.2	6.67	26.7	5.34
Inside surface resistance	E0					0.685		
0.75-in. plaster; 0.75-in. gypsum or other similar finishing								

Figure 4-4 : overall heat transfer coefficient of wall

4.1.2 sun heat gain (**Qs**)

- the sun 3load is the result of the sun's rays transmitted by radiation
- The calculation is made only for the exhibition of the sunny and it is calculated through:

$$QS = Qs|\text{biggest wall} + Qs|\text{roof}$$

$$.\,Qs|\text{biggest wall} = (\text{wall}) + (\text{glass})$$

$$Qs|\text{biggest wall} = ((UAW\Delta Teq) + (I * SC * Ag))$$

Where:

⇒ U: overall heat transfer coefficient (W/m2 K)

⇒ ΔTeq: Additional temperature difference due to sun effect

⇒ AW: area of wall only

⇒ Ag: glass area

⇒ I: solar heat gain factor (w/m2)

⇒ SC: shading coefficient

→ And the roof load → $Qs|\text{roof} = UAroof\Delta Troof$

4.1.3 occupants load (person)(**QP**)

- The people who are in the room or the restaurant go out (sensible heat) as a result of the temperature difference between their body and the atmosphere
- As well as (late heat) as a result of sweat, the moisture content increases in the air of the room or restaurant and it is calculated through:

$$QP = QP, Sen + QP, Lat$$

$$QP, Sen = NP * q\ P, Sen$$

$$QP, lat = NP * q\ P, lat$$

Where:

⇒ NP: number of persons

⇒ q P, Sen /q P, lat: sensible / latent heat person (w/person)

4.1.4 light load

- it is calculated through

$$Qlight = N_{light} * \left(\frac{watt}{lamp}\right)$$

4.1.5 Appliances load

$$Qequip = \sum watt\ of\ all\ equipments$$

4.1.6 ventilation load

- it is calculated through

$$QVent = \dot{m_{vent}} * (ho - hi)$$

$$\dot{m_{vent}} = N_P * \frac{\left(\frac{\frac{l}{s}}{person}\right)}{V_O} * 10^{-3}$$

Where:

⇒ $\dot{m_{vent}}$: fresh air per person in liters per second ($\frac{\frac{l}{s}}{person}$)

⇒ NP : number of persons

⇒ h_o, h_i: enthalpy in Outside condition and inside condition

⇒ v^o: specific volume in outside condition

→ How to calculate the h_o, h_i, h_x

From psychrometric analysis cd shown in (Figure 4-5) by knowing:

- Outside condition → 42 °C DBT and 26.5 °C WBT
- Inside conditions → 24 °C and 50% RH

PSYCHROMETRIC ANALYSIS CD - PsyChart1

POINT	LABEL	AIR FLOW	UOM	PROCESS	GIVEN	DB	WB	RH
out	AR	2	STD L/S	Add State Point	POINT	42.0	26.5	29.9
in	AR	2	STD L/S	Add State Point	POINT	24.0	17.1	50.0
x	AR	2	STD L/S	Add State Point	POINT	42.0	22.6	18.2

PSYCHROMETRIC ANALYSIS CD - PsyChart1

W	V	H	DP	D	VP	AW	START POINT	SECOND POINT
15.5	0.91	82.26	20.8	1.1105	18.443	16.981		
9.3	0.85	47.89	12.9	1.1819	11.197	10.933		
9.3	0.91	66.29	13.0	1.1144	11.200	10.311		

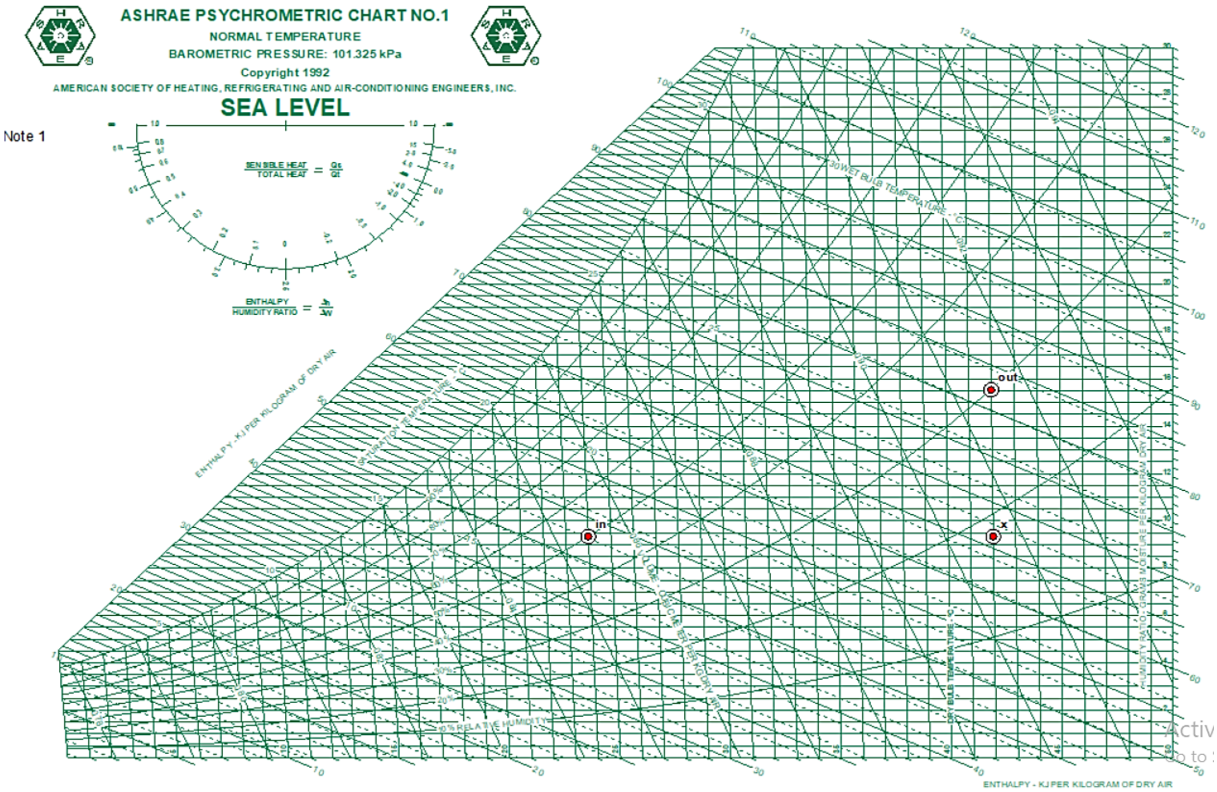


Figure 4-5 : psychrometric analysis cd

4.1.7 Infiltration load

- it is calculated through

$$Qinf = \dot{m}_{inf} * (ho - hi)$$

$$\dot{m}_{inf} = \frac{\forall * v_{space}}{v_o * 3600}$$

Where :

⇒ ∀: number of air change per/h

⇒ Vspace: room space (L*W*H) (m3)

4.2 load calculation Hourly Analysis Program (HAP)

→ in the Hourly Analysis Program, the program Dealing with loads as (Figure 4-6)

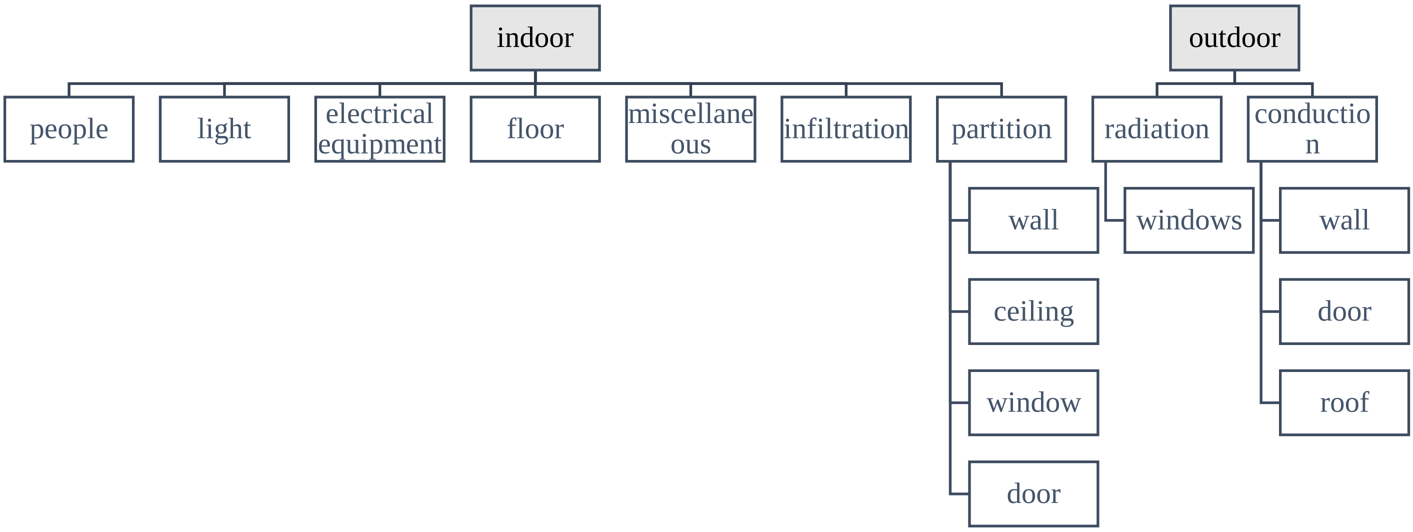


Figure 4-6 : Heat gains

4.2.1 design parameters

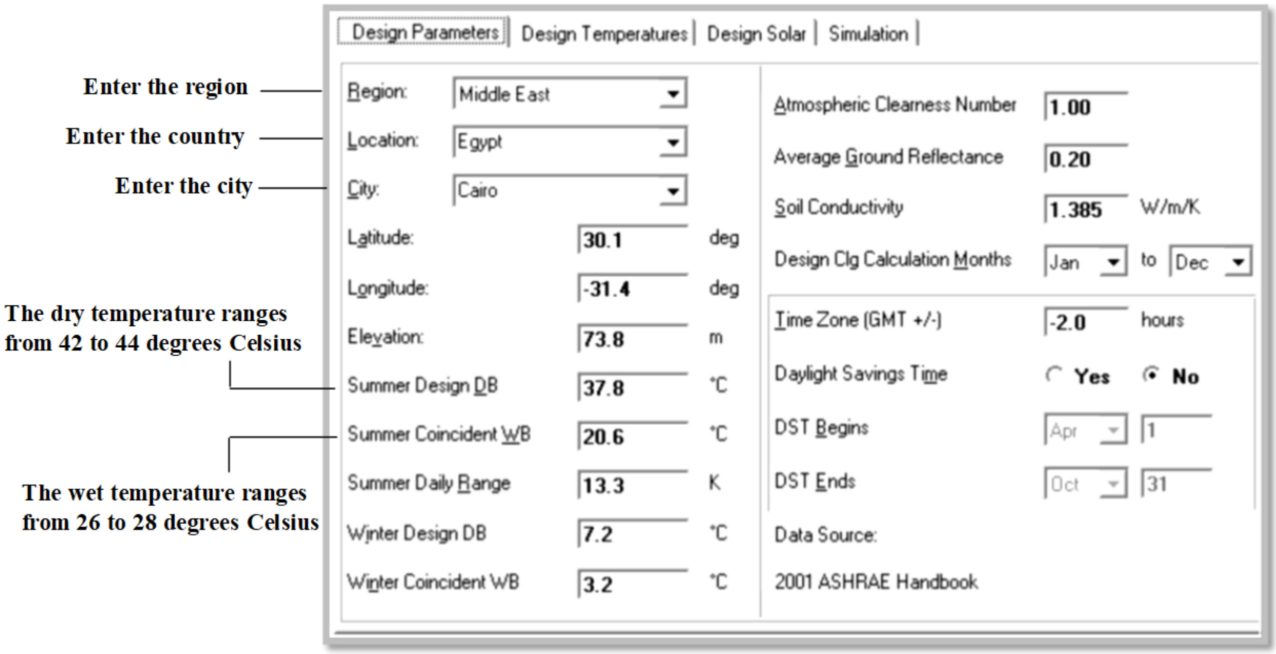


Figure 4-7 : design parameters

4.2.2.1 General

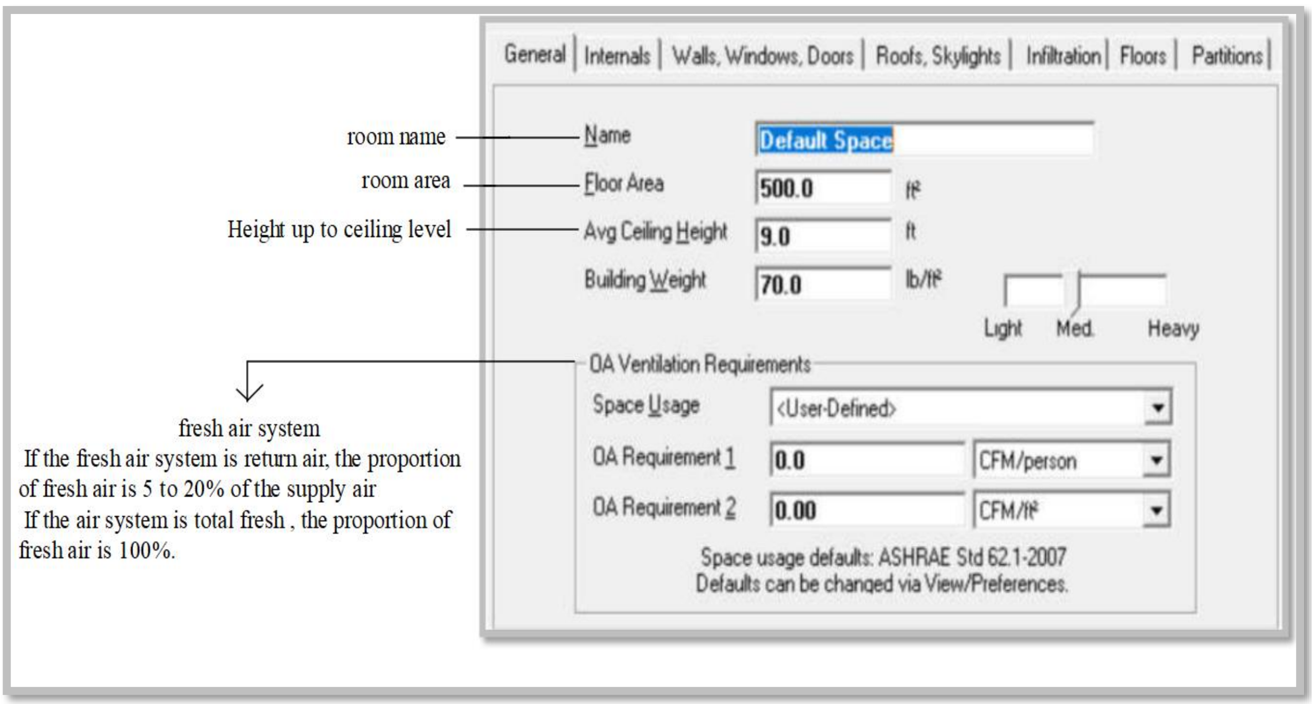


Figure 4-8 : general space input

4.2.2.2 Internal

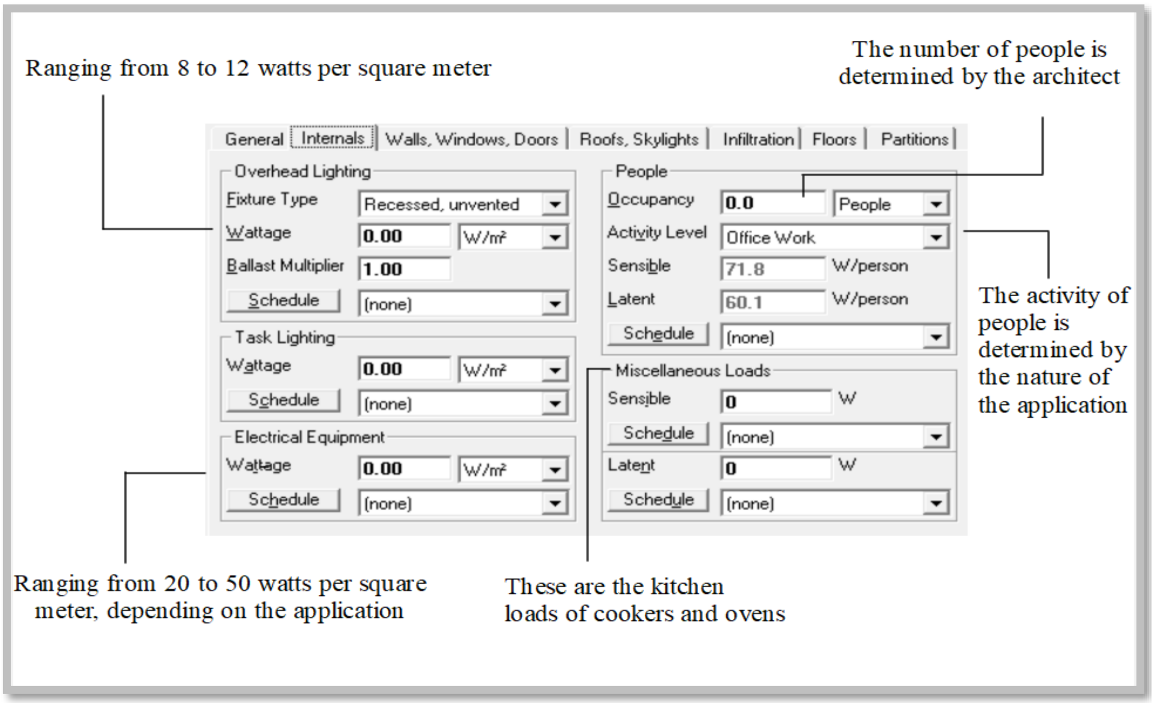


Figure 4-9: internal space input

4.2.2.3 wall, windows, door

.

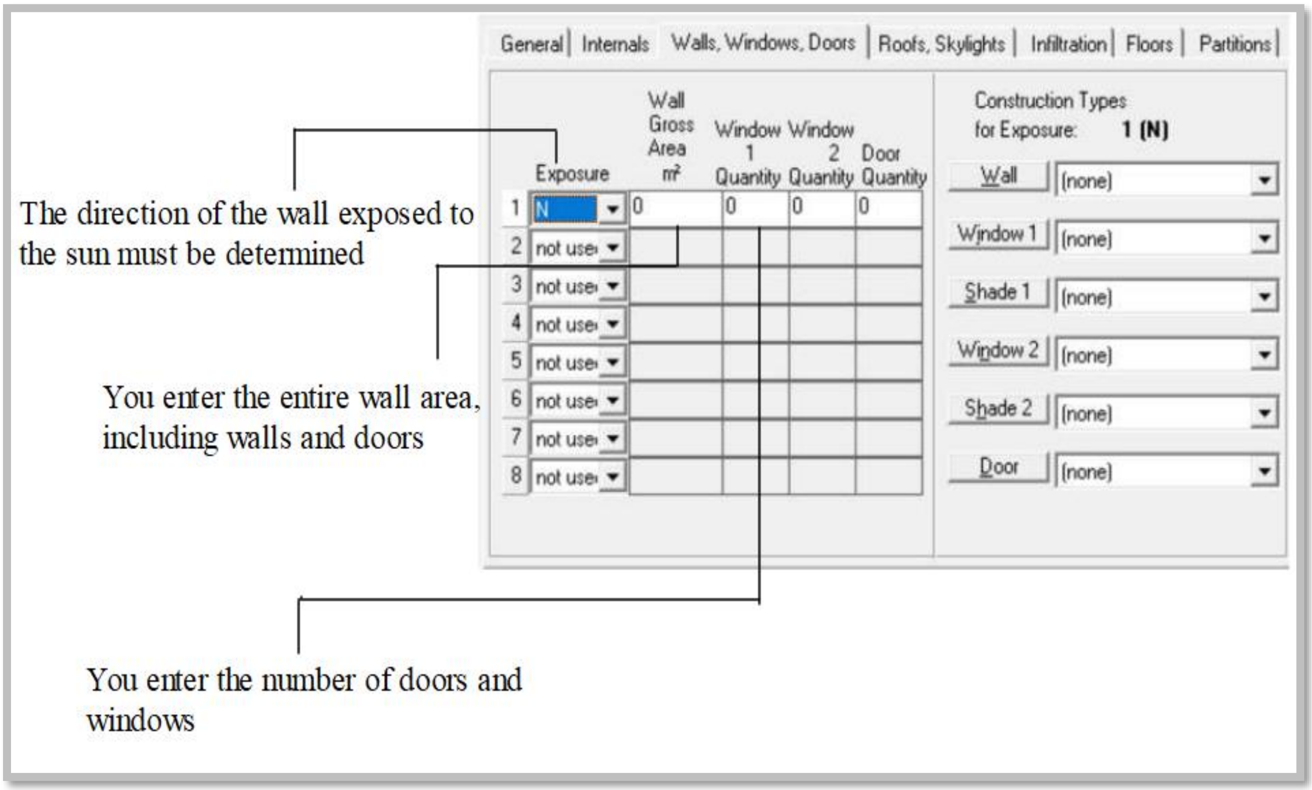


Figure 4-10 : wall, windows, door space input

4.2.2.4 roof

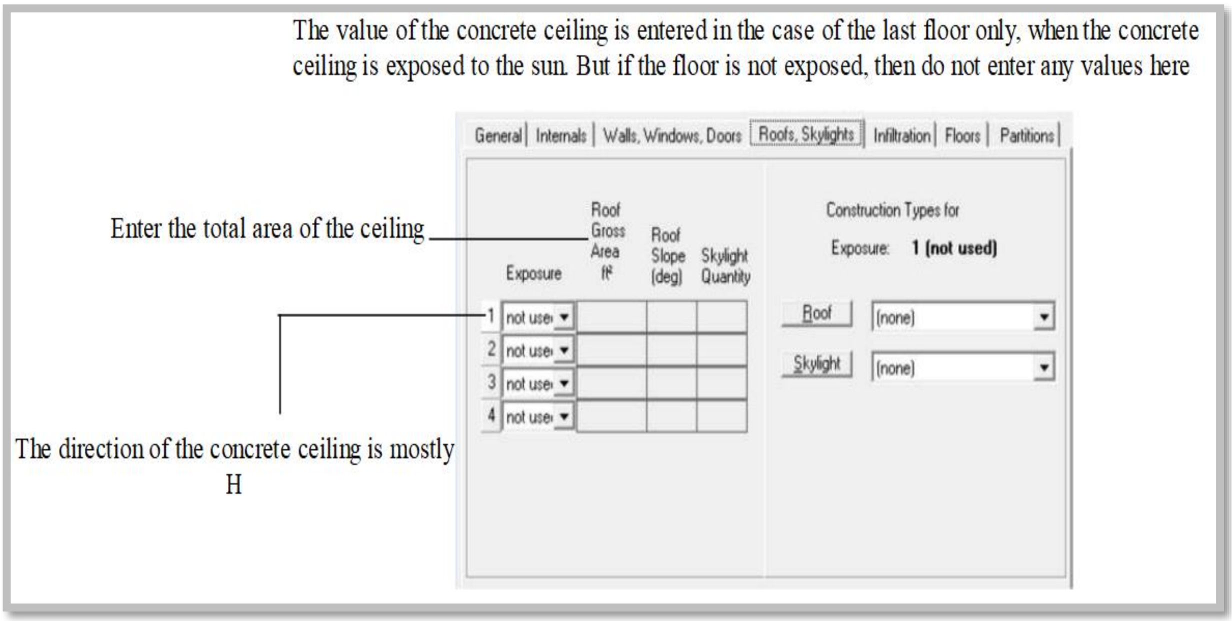


Figure 4-11: roof space input

4.2.2.5 Infiltration

Input of infiltration is effective if the room has a negative pressure, but if the room has a positive pressure, there is no infiltration

4.2.2.6 partitions

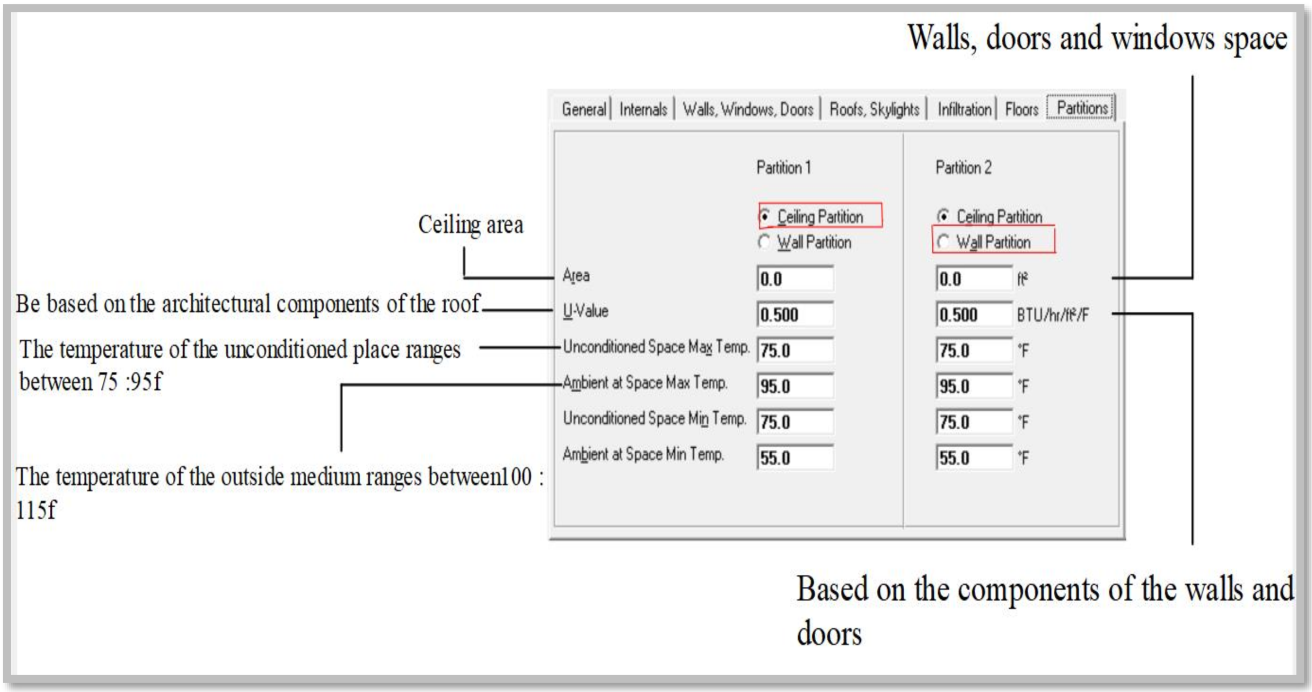


Figure 4-12 : partition space input

chapter (5): Air terminal outlets

→ Air outlets are used to distribute the air in the place well

5.1 Air Terminal Types

1) Square diffuser

It is in the form of a square and the tiles are installed in the ceiling, the most popular size of which is 60 * 60 cm, which is the same size as the ceiling tiles and is used in places whose height does not exceed 4 meters shown in (Figure 5-1)

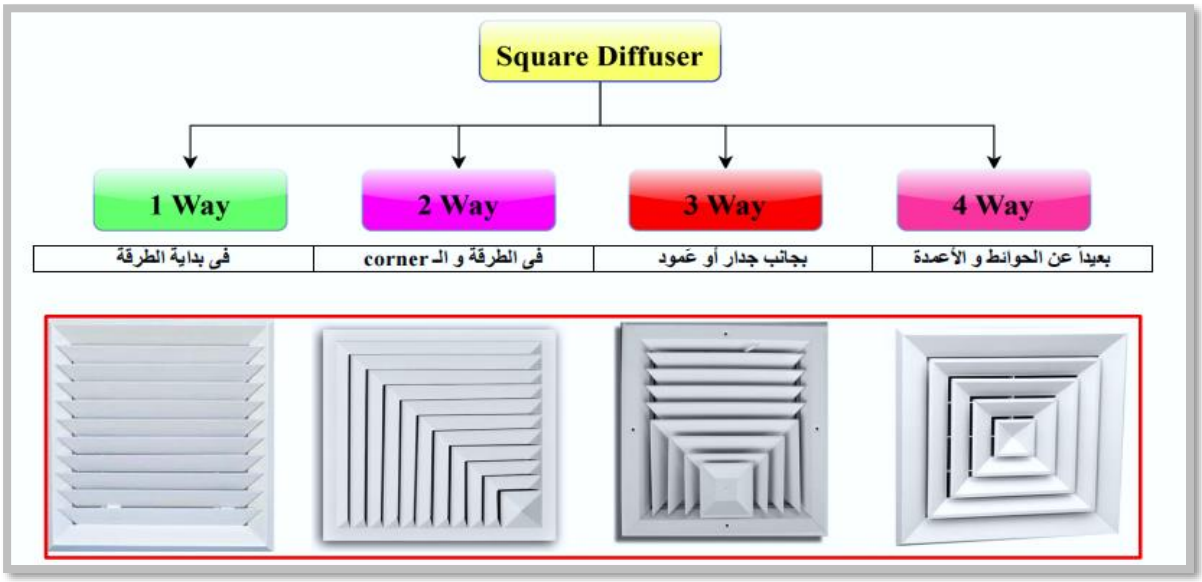


Figure 5-1 : Square diffuser

2) Round diffuser

This type is used if the shape of the false ceiling in the place is circular and is used in places whose height does not exceed 4 meters shown in (Figure 5-2)

Figure 5-2 : Round diffuser

3) Slot diffuser

Very similar to the grill, but its decorative shape is better shown in (Figure 5-3)

Figure 5-3 : Slot diffuser

4) Swirl diffuser

It is very similar to the Round, from which the air comes out in the form of swirls and is used up to a height of 5 meters shown in (Figure 5-4)

Figure 5-4 : Swirl diffuser

5) Grill and register diffuser

- They are installed in the wall in the absence of a ceiling in the place and there are longitudinal holes and other types of transverse openings shown in (Figure 5-5)
- The word Register is a regular grill with a volume damper in addition to the volume damper found in the AHU

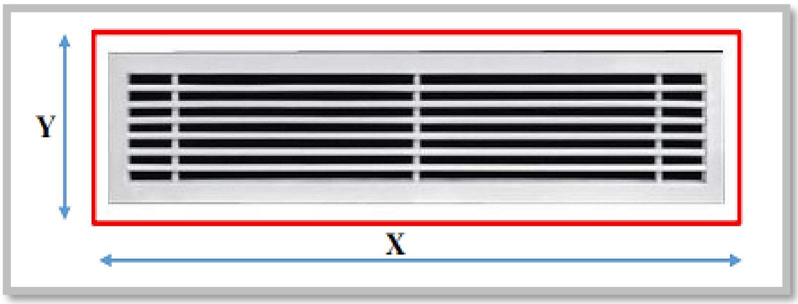


Figure 5-5 : Grill and register diffuser

6) Jet diffuser

- Used in places with high altitudes up to 30 meters shown in (Figure 5-6)

Figure 5-6 : Jet diffuser

7) Door grill and undercut diffuser

- A grill is installed under the doors in order to pass air from one space to another shown in (Figure 5-7)

Figure 5-7 : Door grill and undercut diffuser

note

- When distributing the diffuser, try to make the distance between the supply and return not less than 1.8m in order to avoid a short cut

5.2 Factors affecting the choice of air outlet type

- The Factors affecting the choice of air outlet type shown in (Figure 5-8)

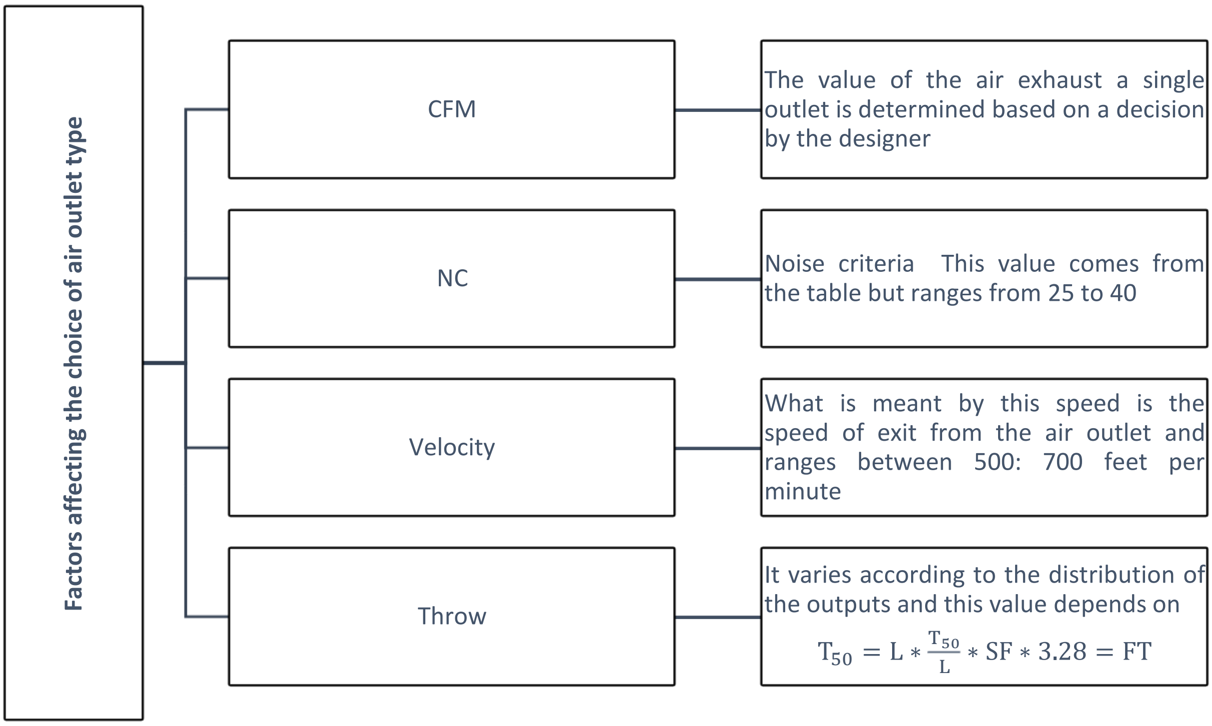
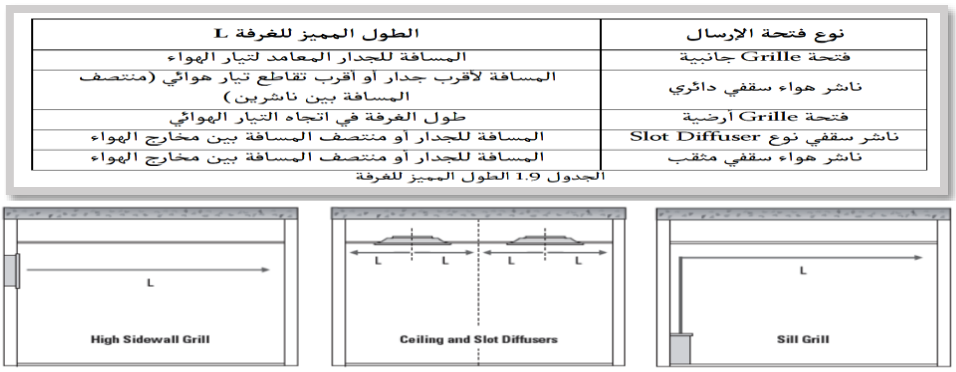


Figure 5-8 : Factors affecting the choice of air outlet type

- To determine the value of room characteristic length (L), which is the horizontal distance between the middle of the transmitter air outlet and the wall, or the mid-distance between two transmission air outlets, and this value varies according to the distribution of
- T50 →The speed will be outside from the air outlet for example500 fpm I will need speed when the average person in the room high which is 1.8m high. It will be 50 fpm. This value meaning how far distance which the speed will be 50 feet per minute.
- The following (Figure 5-9) shows the L for each type of air outlet



الطول المميز للغرفة L	نوع فتحة الإرسال
المسافة للجدار المعامد لتيار الهواء	فتحة Grille جانبية
المسافة لأقرب جدار أو أقرب تقاطع تيار هوائي (منتصف المسافة بين ناشرين)	ناشر هواء سقفي دائري
طول الغرفة في اتجاه التيار الهوائي	فتحة Grille أرضية
المسافة للجدار أو منتصف المسافة بين مخارج الهواء	ناشر سقفي نوع Slot Diffuser
المسافة للجدار أو منتصف المسافة بين مخارج الهواء	ناشر هواء سقفي مثقب

الجدول 1.9 الطول المميز للغرفة

Figure 5-9 : characteristic length of room L

- For T50/L, it varies according to the type of each outlet and also comes through (Figure 5-10)

مدى القيمة T_{50}/L	T_{50}/L	نوع فتحة الإرسال
1.0 — 2.2	1.7	فتحة Grille جانبية
0.5 — 1.5	0.8	ناشر هواء سقفي دائري
0.8 — 1.8	1.6	فتحة Grille أرضية شفرات بزاوية 0
0.6 — 1.7	0.7	فتحة Grille أرضية شفرات بزاوية غير 0
0.3 — 1.5	0.3	ناشر سقفي نوع Slot Diffuser — T_{100}/L
1.4 — 2.7	2.0	ناشر هواء سقفي مثقب

الجدول 2.9 القيمة الوسطية ل T_V/L

Figure 5-10 : mean value of T50/L

5.3 Case study on Air outlets distribution

- Room (4.6 * 8 m) → cfm =1600
- Assume 4 diffuser supply and 4 diffuser Return

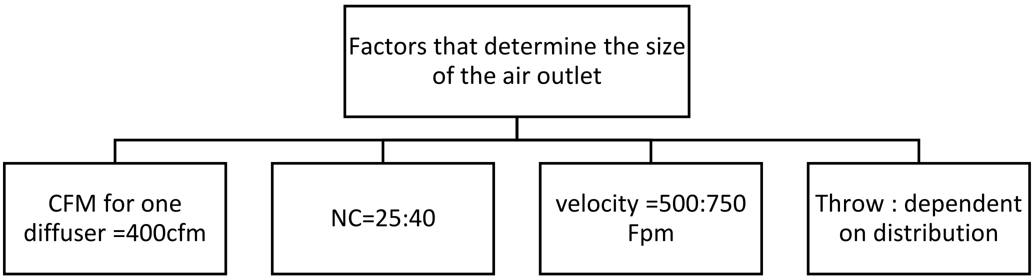


Figure 5-11 : Factors that determine the size of the air outlet

→ The distribution of exits depends on the designer, and the most important thing is to take into account the aesthetic form

First distribution

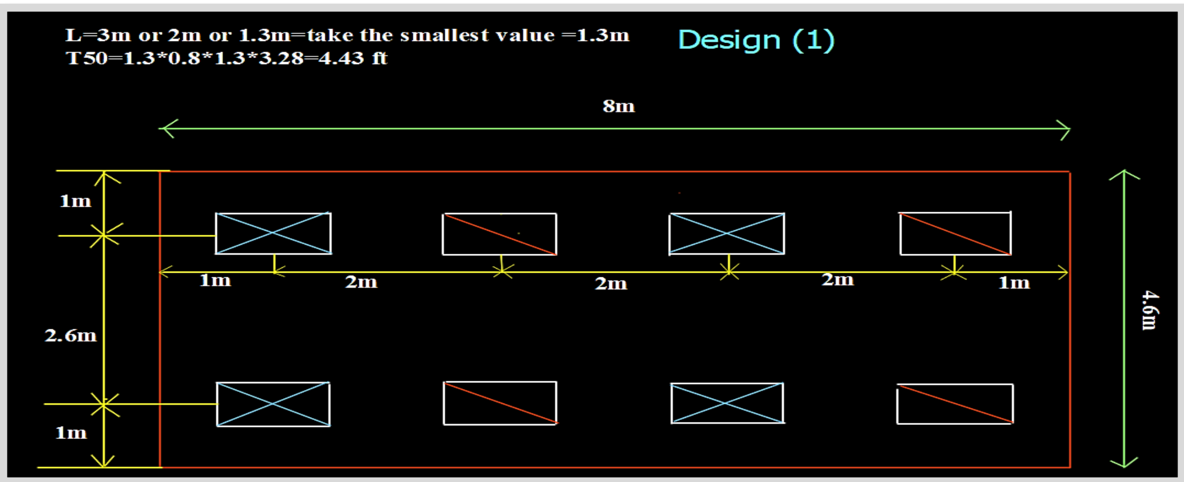


Figure 5-12 : first distribution of airoutlets

Second distribution

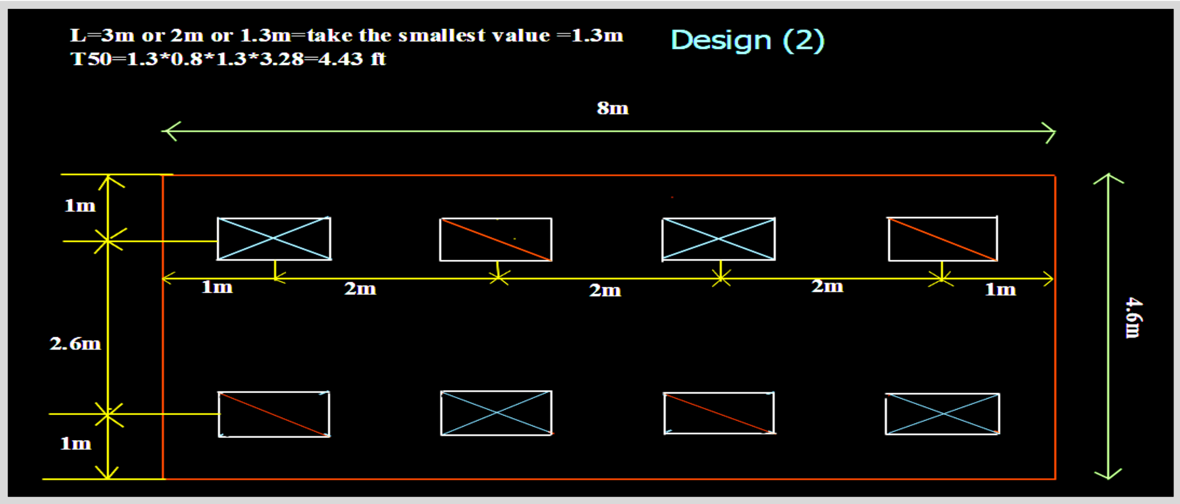


Figure 5-13 : second distribution of airoutlets

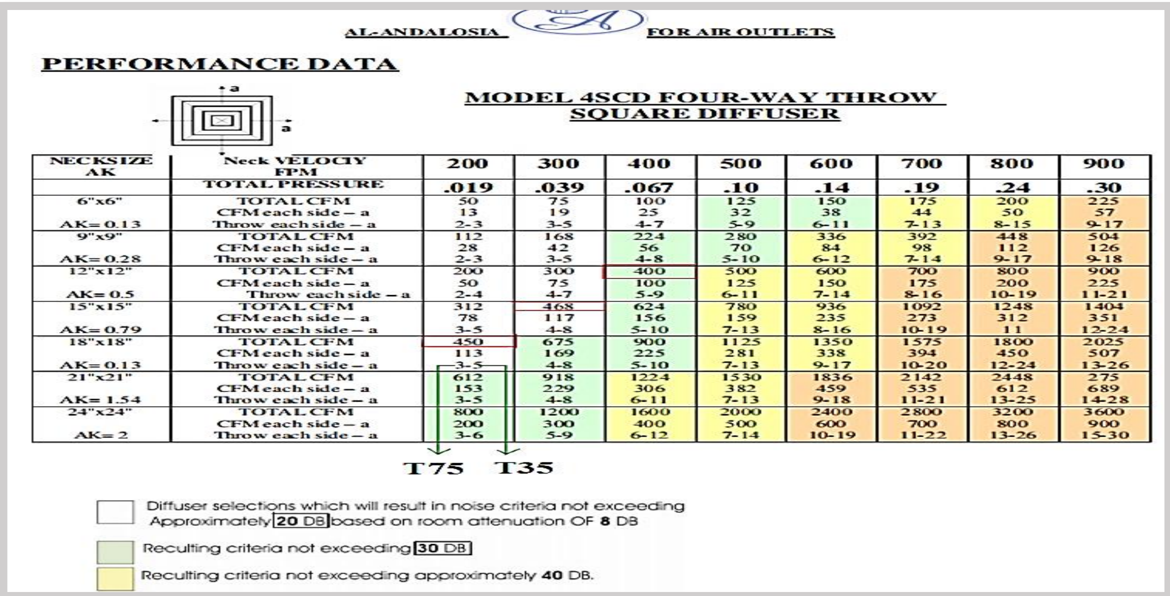
→ $T_{50} = 4.43$ ft

→ 400 cfm

→ NC = 25: 40

→ Velocity = 500: 750 fpm

- Now you can enter the catalog and choose the appropriate outlets through this set of variables, for example, and let it be from Alandalosia
- The design is on the neck and for the face it is face=neck+6"



AL-ANDALOSIA FOR AIR OUTLETS

PERFORMANCE DATA

MODEL 4SCD FOUR-WAY THROW SQUARE DIFFUSER

NECKSIZE AK	Neck VELOCIY FPM	200	300	400	500	600	700	800	900
	TOTAL PRESSURE	.019	.039	.067	.10	.14	.19	.24	.30
6"x6"	TOTAL CFM	50	75	100	125	150	175	200	225
	CFM each side – a	13	19	25	32	38	44	50	57
AK= 0.13	Throw each side – a	2-3	3-5	4-7	5-9	6-11	7-13	8-15	9-17
9"x9"	TOTAL CFM	112	168	224	280	336	392	448	504
	CFM each side – a	28	42	56	70	84	98	112	126
AK= 0.28	Throw each side – a	2-3	3-5	4-8	5-10	6-12	7-14	9-17	9-18
12"x12"	TOTAL CFM	200	300	400	500	600	700	800	900
	CFM each side – a	50	75	100	125	150	175	200	225
AK= 0.5	Throw each side – a	2-4	4-7	5-9	6-11	7-14	8-16	10-19	11-21
15"x15"	TOTAL CFM	312	468	624	780	936	1092	1248	1404
	CFM each side – a	78	117	156	159	235	273	312	351
AK= 0.79	Throw each side – a	3-5	4-8	5-10	7-13	8-16	10-19	11	12-24
18"x18"	TOTAL CFM	450	675	900	1125	1350	1575	1800	2025
	CFM each side – a	113	169	225	281	338	394	450	507
AK= 0.13	Throw each side – a	3-5	4-8	5-10	7-13	9-17	10-20	12-24	13-26
21"x21"	TOTAL CFM	612	918	1224	1530	1836	2142	2448	275
	CFM each side – a	153	229	306	382	459	535	612	689
AK= 1.54	Throw each side – a	3-5	4-8	6-11	7-13	9-18	11-21	13-25	14-28
24"x24"	TOTAL CFM	800	1200	1600	2000	2400	2800	3200	3600
	CFM each side – a	200	300	400	500	600	700	800	900
AK= 2	Throw each side – a	3-6	5-9	6-12	7-14	10-19	11-22	13-26	15-30

T75 T35

Diffuser selections which will result in noise criteria not exceeding Approximately 20 DB based on room attenuation OF 8 DB

Reculting criteria not exceeding 30 DB

Reculting criteria not exceeding approximately 40 DB.

Figure 5-14 : Alandalosia catalog

→ From Alandalosia catalog shown in (Figure 5-14)

- Neck size = 18 " * 18"
- face size = 24" * 24"

chapter (6): operation room

6.1 Types of supply outlets in operation room

→ There are two types of outlets used in operation room shown in (Figure 6-1)

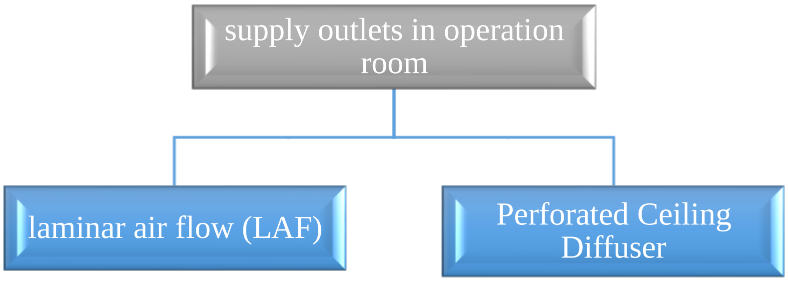


Figure 6-1 : diffusers in operation room

6.1.1 Laminar air flow (LAF)

- All of it is one piece and all of this space has a filter with a very high purity and low speeds and reaches at a height of 1.8m to 25fpm shown in (Figure 6-2)

Figure 6-2 : Laminar air flow (LAF)

- The number of connections is low compared to the perforated shown in (Figure 6-3)

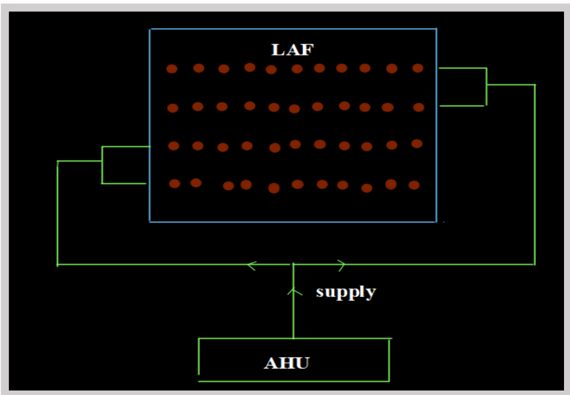


Figure 6-3 : Laminar air flow (LAF) connections

6.1.2 Perforated ceiling diffuser

- It is a group of outlets next to each other, unlike the LAF, which was one piece and there is no filter, but it is installed shown in (Figure 6-4)
- The air also comes out laminar and pure, but less than LAF

Figure 6-4 : Perforated ceiling diffuser

- The number of connections is large compared to the LAF, each output with a connection shown in (Figure 6-5)

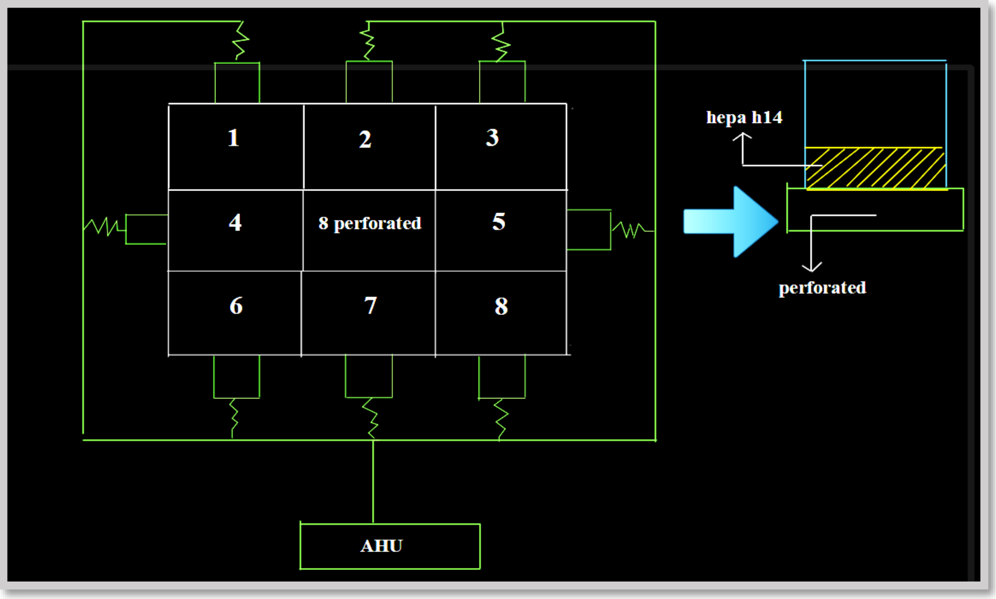


Figure 6-5 : Perforated ceiling diffuser connection

6.2 Exhaust system in operation room

- As for the exhaust, it is done through a grill. It is possible from 2way or 3war or 4way. It is preferable if the 2way are in each other and the intake is done from below to ensure that the turbulent flow does not occur.
- In the (Figure 6-6) below, I will draw 2/3 from the bottom and 1/3 from the top

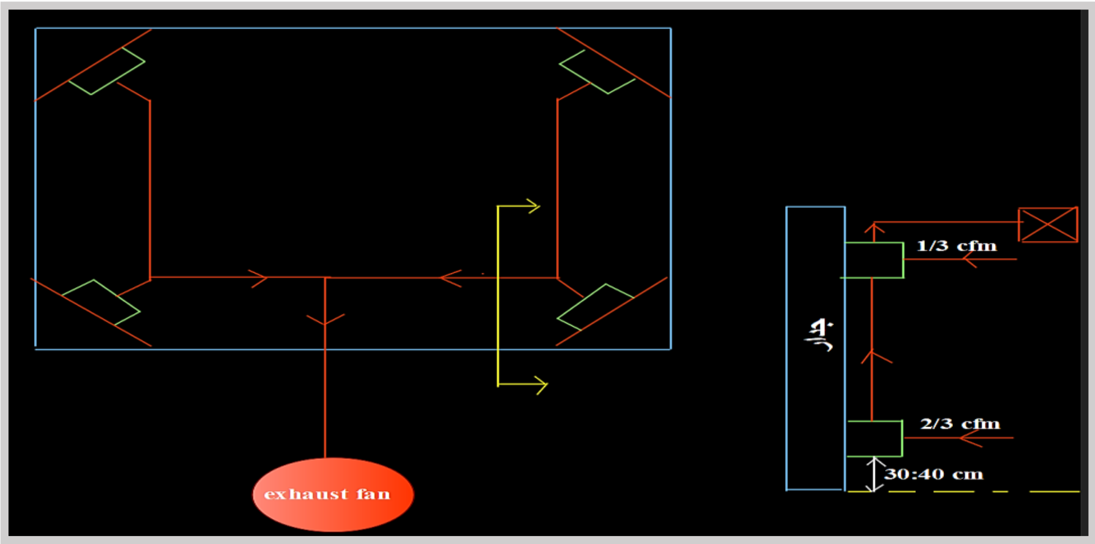


Figure 6-6 : Exhaust system in operation room

- It can come ready and be connected to a flexible duct shown in (Figure 6-7)

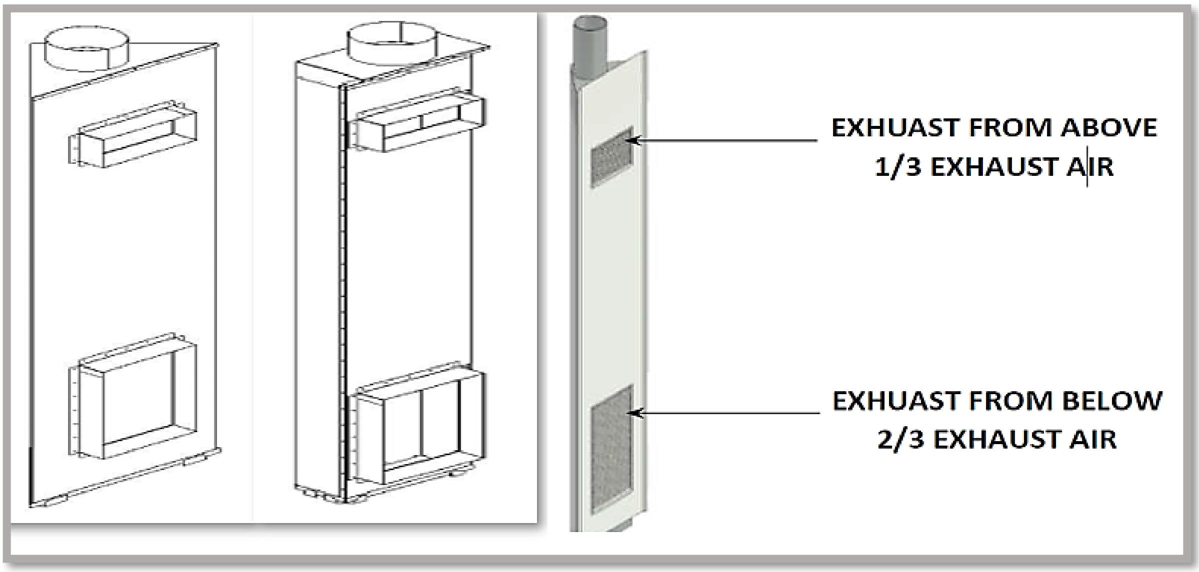


Figure 6-7 : exhaust grill

6.3 operating room in Revit

- the Perforated ceiling diffuser and Exhaust system in operation room shown in (Figure 6-8)

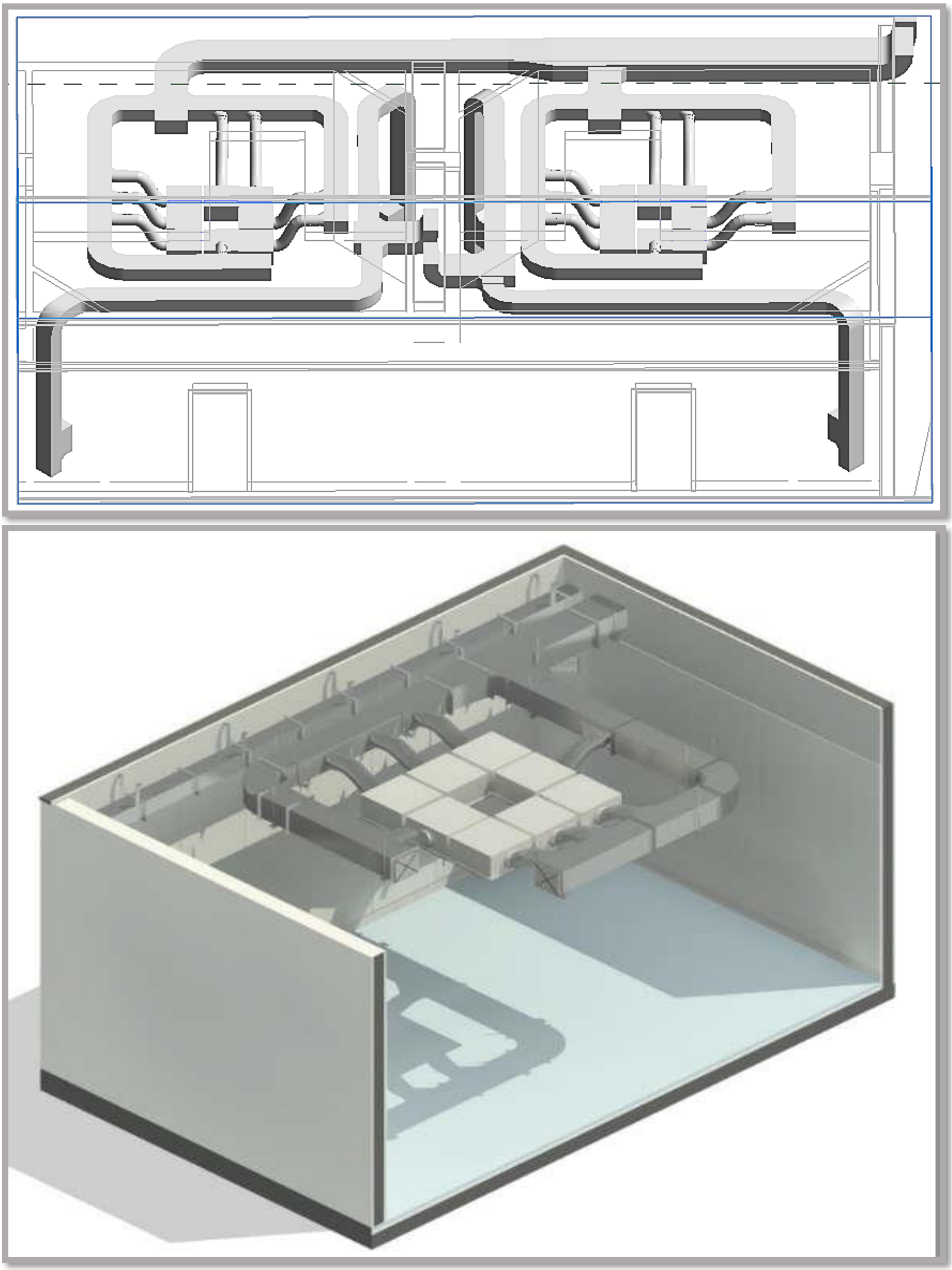


Figure 6-8 : Perforated ceiling diffuser and Exhaust system in operation room

6.4 How to control the pressure of each room

- In case you want the room positive

$$CFM_{fresh} = \frac{ACH * Area * Height}{1.7}$$
CFM Exhaust = (0.85 : 0.90) x CFM Fresh

- Is it really, I made the room pressure positive, but how much?
    - Of course, you don't know, so we need to know how I can know the pressure of each room

6.4.1 Case study on control the pressure of each room

Three room shown in (Figure 6-9) has the cfm as the following

$\rightarrow CFM_{fresh\ or\ supply}(1) = 600cfm$

$\rightarrow CFM_{fresh\ or\ supply}(2) = 550cfm$

$\rightarrow CFM_{fresh\ or\ supply}(3) = 500cfm$

- The value of cfm fresh is calculated by this law, given the rate of air change for each room, as well as the volume of the room

$$CFM_{fresh} = \frac{ACH * Area * Height}{1.7}$$

- The rooms are total fresh air

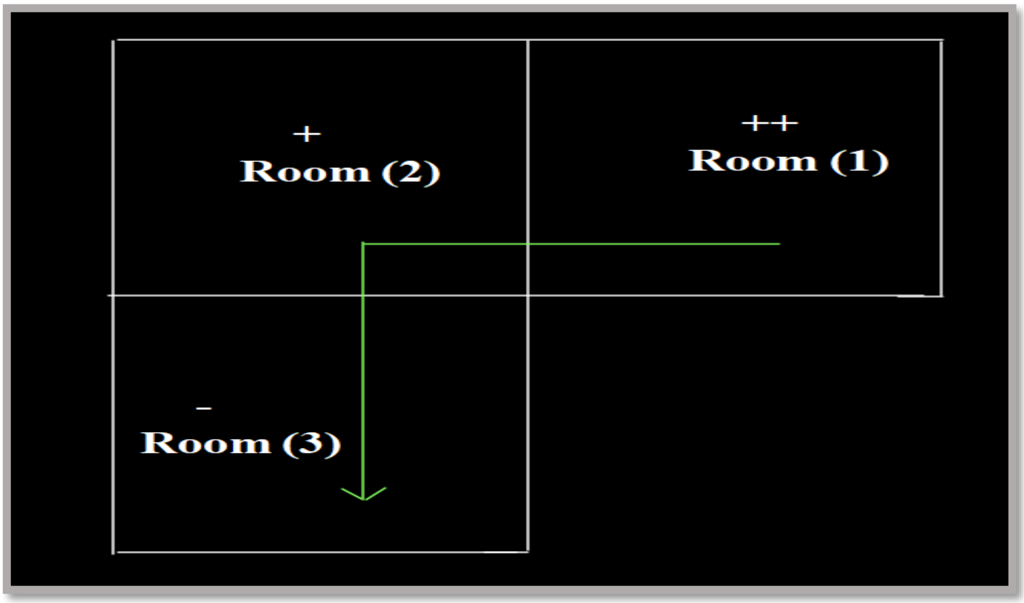


Figure 6-9 : pressure in three room

Solution

- From these rules can control the pressure of each room

$CFM\ Exhaust = CFM_{fresh\ or\ supply} \pm \Delta cfm$
$+ve\Delta cfm \rightarrow$ when negative room
$-ve\Delta cfm \rightarrow$ when postive room
$\Delta cfm = 2610 * A_C * \sqrt{\Delta P}$
$A_C \rightarrow$ infiltration are $(0.2{:}\,0.3)ft^2$
$\sqrt{\Delta P} \rightarrow$ pressure of room (inwg)

- This means that + or – one equals 2.5pa = 0.01 inwg

$\Delta cfm = 2610 * A_C * \sqrt{\Delta P} = 2610 * 0.2 * \sqrt{0.01} \cong 50CFM$

Room (1)

$CFM_{fresh\ or\ supply}(1) = 600cfm$

$$\Delta cfm = 100$$

$$CFM_{Exhaust} = CFM_{fresh\ or\ supply} - \Delta cfm = 600 - 100 = 500CFM$$

Room (2)

$CFM_{fresh\ or\ supply}(2) = 550cfm$

$$\Delta cfm = 50$$

$$CFM_{Exhaust} = CFM_{fresh\ or\ supply} - \Delta cfm = 550 - 50 = 500CFM$$

Room (3)

$CFM_{fresh\ or\ supply}(3) = 500\ cfm$

$$\Delta cfm = 50$$

$$CFM_{Exhaust} = CFM_{fresh\ or\ supply} + \Delta cfm = 500 + 50 = 550\ cmf$$

6.5 Case study in design of operation room

Operation room:

→ area $= 40m^2$

→ high $= 3m$

determine the number of perforated and number of exhaust grill

Note

- These rooms are designed with total fresh air although the code did not mention that, but this is for the sake of safety like
  - Operations room → (ACH=20:25)
  - (Incubations - isolation - sterilization - rays) → (ACH=15)

Solution

→ $CFM_{\text{fresh or supply}} = \frac{ACH*Area*Height}{1.7} = \frac{20*40*3}{1.7} = 1450cfm$

- Pressure in operation room =10pa

→ $\Delta cfm = 2610 * A_C * \sqrt{\Delta P} = 0610 * 0.3 * \sqrt{0.04} = 150cfm$

→ $CFM_{\text{Exhaust}} = CFM_{\text{fresh or supply}} - \Delta cfm = 1450 - 150 = 1300\ cfm$

Fresh

→ no of filter = no of porferated $= \frac{CFM_{\text{fresh or supply}}}{\text{rated air flow of filter}} = \frac{1450}{200} = 7.2$ filter $\cong$ 8filte

Part Number	Style Code	Nominal Size (Feet)	Actual Size Inches (H x W x D)	Rated Airflow (SCFM) 100 FPM	Std. Pkg. Qty	Ship. Wt. Lbs/Box (± 7%)	Cubic Ft.	Cell Sides	Media Pack
99.99% Scanned (H)									
577-890-004	43A89A2T2H0	2 x 1	24 x 12 x 2¾	165	1	18.0	0.7	2¾ (89)	2" (A)
577-890-005	14A89A2T2H0	2 x 2	24 x 24 x 2¾	350	1	21.0	1.4	2¾ (89)	2" (A)
577-890-007	46A89A2T2H0	2 x 3	24 x 36 x 2¾	540	1	24.0	2.0	2¾ (89)	2" (A)
577-890-008	17A89A2T2H0	2 x 4	24 x 48 x 2¾	725	1	28.0	2.7	2¾ (89)	2" (A)
99.9995% on 0.1 to 0.2 micron - Laser Tested and Scan Tested (M)									

Figure 6-10 : filter selection

→ The filter used can get 350, but I chose 200. We thought that after a while it would start to get dirty. The pressure drop will increase shown in (Figure 6-10)

→ number of Perforated ceiling diffuser shown in (Figure 6-11)

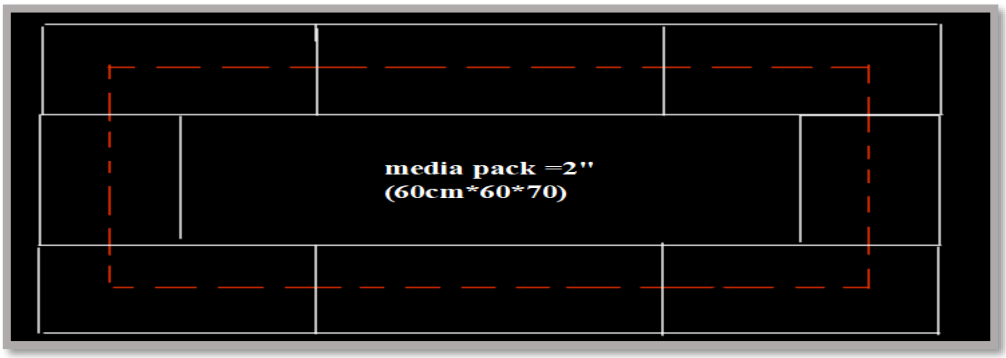


Figure 6-11 : number of Perforated ceiling diffuser

→ Calculating the pressure drop for one filter using the media pack -100fpm shown in (Figure 6-10), using the flowing (Figure 6-12)

$$\Delta P = 0.45 \text{ inwg}$$

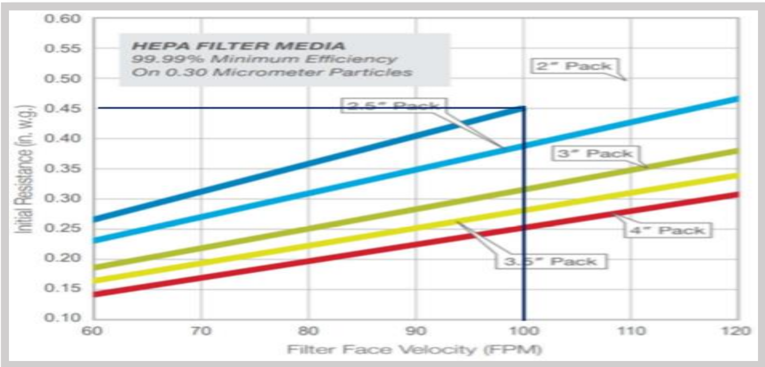


Figure 6-12 : pressure drop for one filter

Exhaust

$$\text{cfm for one grill} = \frac{1300}{2} = 650 \text{ cfm}$$

$$A = \frac{Q}{V} = \frac{650 * 144}{(300:400)} = 234\text{ft}^2$$

→ Velocity is the speed of air entering the grill

$$A = y * x$$

$Assume\ x = 8"$

$then\ y = 30"$

→ the exhaust system in operation room shown in (Figure 6-13)

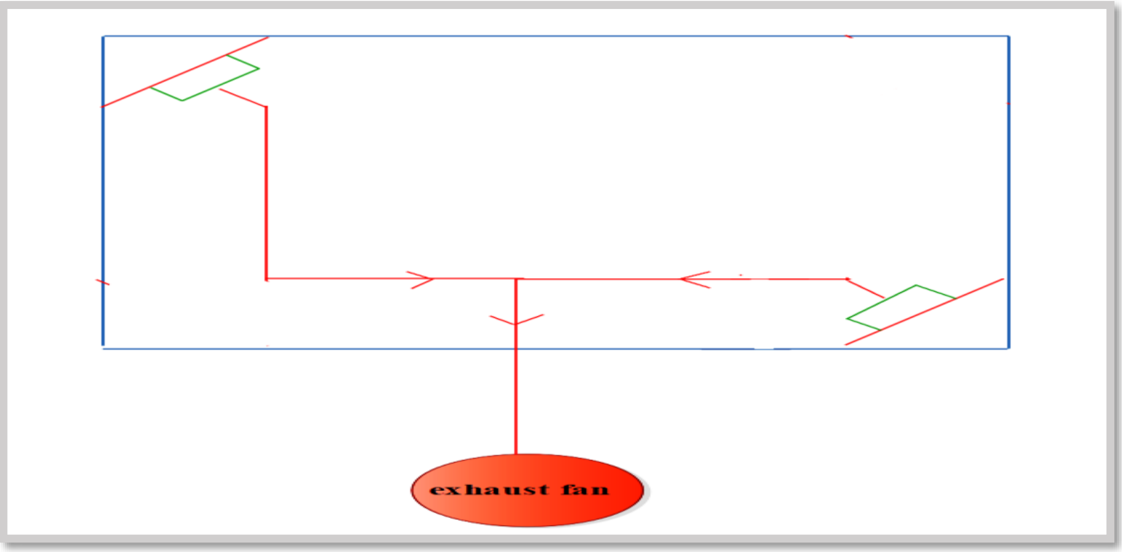


Figure 6-13 : exhaust system of operation room

chapter (7): Air distribution system

→ The last step in designing of air conditioning for any place is the distribution of air in the place

7.1 Main component of air distribution system

1) Fan → which is responsible for moving the air in the duct until it reaches the air outlet and goes into the room
2) Duct →The pipes through which the air flows until it reaches the air outlet
3) Air outlet → Air outlets are used to distribute the air in the place well

7.2 Ducts

→ Ducts are used in heating, ventilation, and air conditioning (HVAC) to deliver and remove air

7.2.1 Duct materials

1) Galvanized duct

It is characterized by durability, rust resistance and fire resistances shown in (Figure 7-1), and it can be manufactured in several forms:

- Rectangle
- Round
- Ovel

Figure 7-1 : Galvanized steel duct

2) Fabric duct

It is a duct of tissue that transports and distributes air through perforations located on the duct itself shown in (Figure 7-2)

Fabric duct features include:

- simpler design
- not commonly create condensation on its surface and can therefore be used where air is supplied below the dew point. Material that eliminates moisture may be healthier for the occupants
- Much lighter weight than sheet metal
- lower price or cheaper
- Ease of installation

Figure 7-2 : Fabric ducts

3) Flexible ducting

- Flexible ducts (also known as flex) are typically made of flexible plastic over a metal wire coil to shape a tube
- the insulation is usually, polyester fiber and glass wool

Figure 7-3 : Flexible ducting

7.2.2 Advantages & Disadvantages of Various Ducts

1) Round Duct

- Advantages:
- Lowest friction
- Less material required for fabrication

2) Square Duct

- Advantages:
- Less Friction as compares to rectangular duct
- Less material required for fabrication as compared to rectangular duct

3) Rectangular Duct:

- Advantages:
- Easy to fabricate at site
- Disadvantages:
- Friction is more in rectangular duct as compared to round & square duct

7.2.3 Dampers

1) Volume control dampers

It is used to calibrate the flow of air passing through the duct, which is in several forms shown in (Figure 7-4)

- Opposed Blade damper
  - The result of the shape of the opposite blades
- Parallel Blade damper
  - The result of the shape of the parallel blades

Figure 7-4 : Opposite blades and parallel blades

2) fire dampers

- The fire valve separates the rooms and its normal state is the normal open, when sensing a fire, the valve closes to prevent the spread of fire between rooms
- The valve has two types shown in (Figure 7-5):
  - curtain
  - multi-blade

Figure 7-5 : fire dampers

3) Smoke damper

- Its normal condition is the open state, and when it senses the presence of smoke, the valve closes to prevent the spread of smoke between areas shown in (Figure 7-6)
- It takes a signal from the smoke detector

Figure 7-6 : Smoke damper

7.2.4 Factors to be taken into consideration when designing

1) ceiling height
   - It is necessary to know the height above the ceiling, I mean how many centimeters are available above the ceiling, so that it is not possible to have 40 centimeters available, for example, and I design the height of the duct 45 centimeters
2) Aspect ratio
   - It is the ratio between the width of the duct to the depth of the duct
     - $\text{AR} = \frac{\text{W}}{\text{H}} \leq (1:4)$
     - The higher the AR, the higher the loss
3) Air velocity inside the duct
   - We try to keep the air speed inside the duct equal to the speed of the machine used in order to avoid losses shown in (Table 7-1)

Table 7-1 : speed of the machine

Ahu→1200 :1400 fpm
Fcu→ 800: 900 fpm
Fresh fan →1400: 1600 fpm
Exhaust fan→1400: 1800 fpm

4) The height of the branch must be less than the height of the duct in order to know how to enter the branch in the air of the main
5) The width of the branch must be greater than the width of the diffuser in order to know how to open it
6) When changing the dimensions, it is preferable to constant one of the dimensions and change the other dimension (and it is preferable to specify the height) because that is better when manufacturing

7.3 Equal Fraction Method by Using Duct Sizer

7.3.1 The factors which affected on design of ducts

- The variables which affected on design of ducts shown in (Figure 7-7)

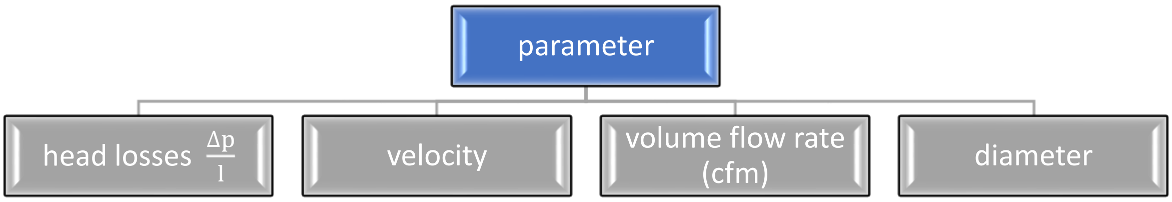


Figure 7-7 : factors affected on design of ducts

Very important notes

- In order to be able to deal with the program, you must have two characteristics
- We will have the cfm quantity as well as the speed. We can get Δp/l and diameter, if the duct is circular, or we get a length in width if the duct is rectangular.
- Try to keep the value of Δp/l not exceeding 0.09 inwg/100 ft
- And the value of Δp/l = 0.088” wg/100ft. We try to fix it with the cfm value and we get the dimensions of the duct. Keep in mind the value of Δp/l is not a code. Shown in (Figure 7-8)
- When changing the dimensions, it is preferable to constant one side and change the other dimension.

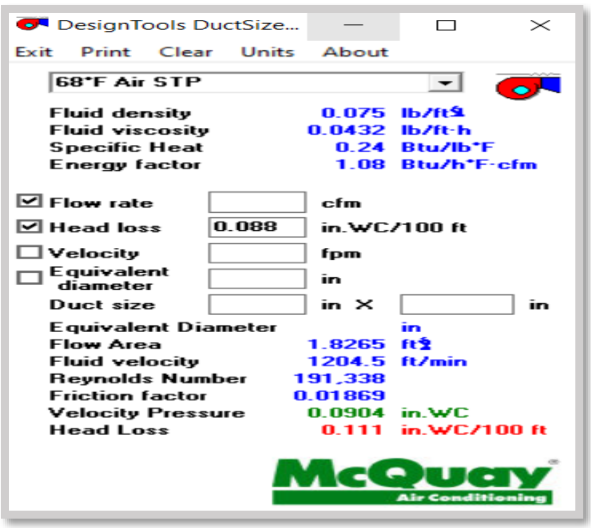


Figure 7-8 : duct size with Knowing two characteristics (pressure loss and CFM)

7.3.2 Case study on duct design

Size the following duct system → Given: The shown data:

- The initial velocity of air at plenum = 8 m/s.
- The maximum allowable height of Duct equals = 150mm.

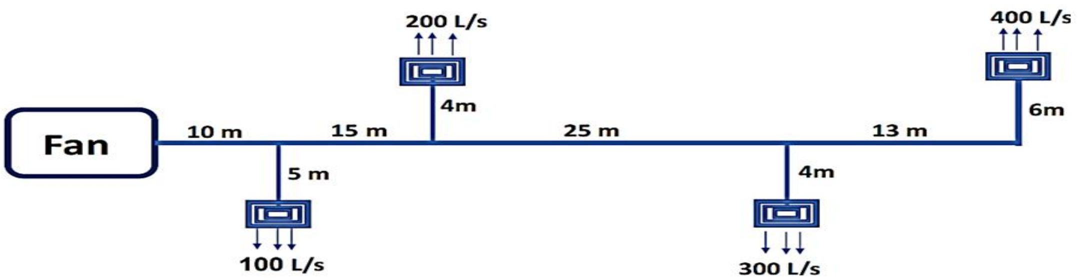


Figure 7-9 : Case study on duct design

solution

→ The first step is to know the amount of flow in each path shown in (Figure 7-10)

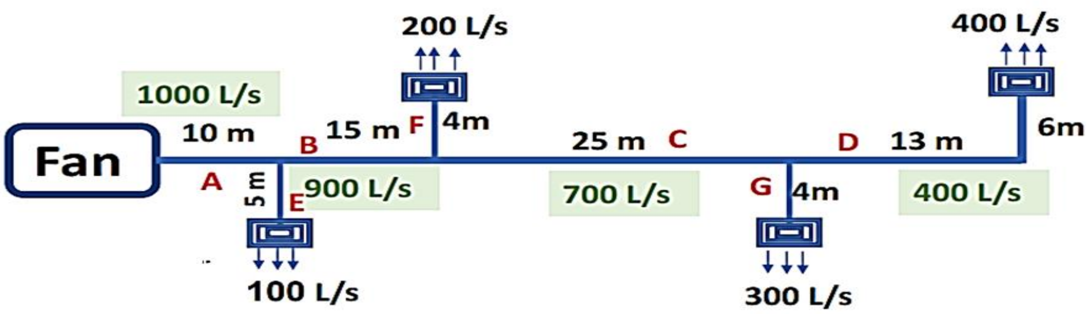


Figure 7-10 : amount of flow in each path

→ The second step by using two characteristics (velocity = 8 m/s -flow at plenum=1000L/s) We can get Δp/L from duct sizer shown in (Figure 7-11)

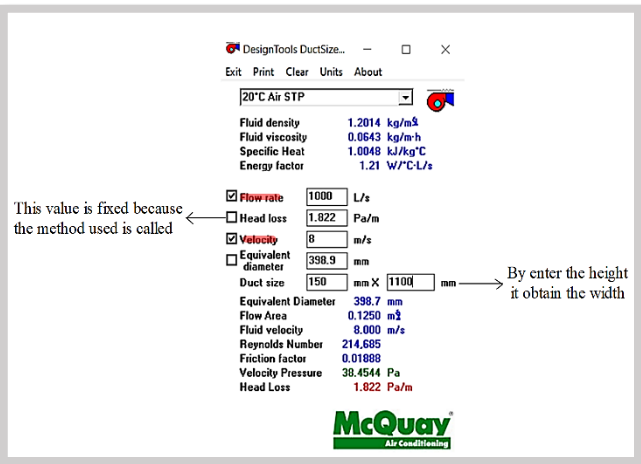


Figure 7-11 : duct sizer

→ From duct sizer, with the aid of (ΔPa /m and Flow rate)

- get the dimension of other duct paths labelled by (B,C,D,E,F,G) shown in below

Section	Velocity (m/s)	$\Delta p/L\ (\frac{pa}{m})$	L/s	W (mm)	H(mm)
A	8	**1.822**	1000	**1100**	150
B		**1.822**	900	**1000**	150
C		**1.822**	700	**800**	150
D		**1.822**	400	**475**	150
E		**1.822**	100	**150**	150
F		**1.822**	200	**275**	150
G		**1.822**	300	**375**	150

7.4 External static pressure

→ Because low static pressure leads to large static losses, and variable losses are low in terms of losses

- Total losses = static losses + variable losses
- variable losses = 20% from total losses
- the static losses = internal losses + external loss
- internal losses = Δpcoil, filter, fan body

→ shown in the (Figure 7-12)

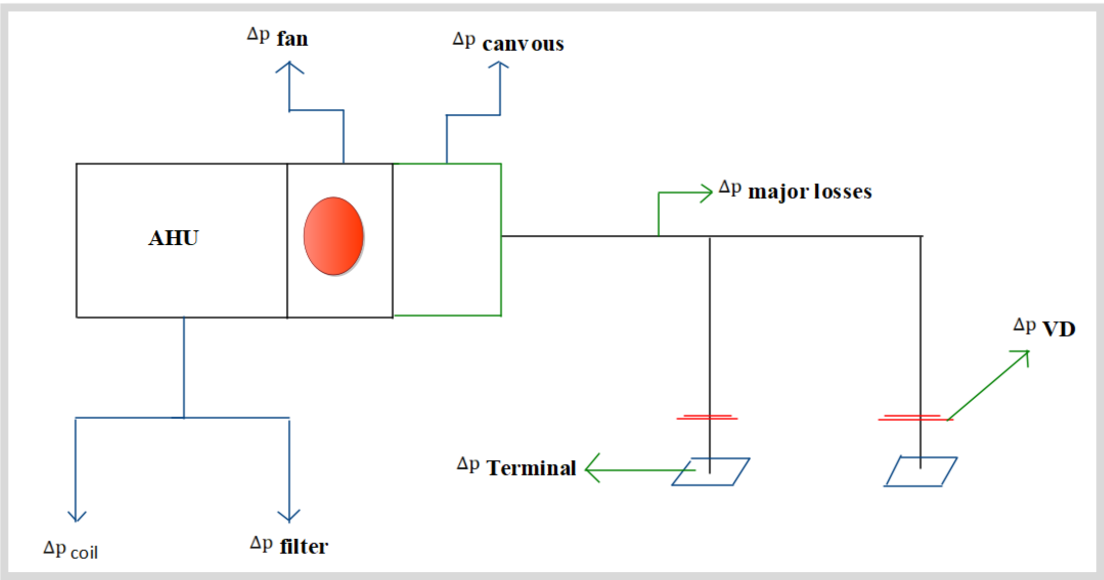


Figure 7-12 : static losses

Very important notes

- The internal losses are coming from the supplier. I am only responsible for calculating the external losses
- to calculate the external losses, we must determine the most difficult path, not the longest path, then the compensation in the following law:

$$ESP = \left(\frac{\Delta p}{l}\left(L + \sum L_{equ}\right)\right) + \Delta p_T + \Delta p_{VD} + \Delta p_{Flexible} + \Delta p_{Canvous}$$

- We express the secondary losses from, elbows, and take offs in what is known as the equivalent length shown in (Table 7-2)

Table 7-2: equivalent length of secondary losses

Type	Equivalent Length
Branch	10 ft
Elbow	20 ft
Reducer	15 ft
Take off	25 ft

- These numbers are approximate for ease and accuracy of using the program shown in (Table 7-3)

Table 7-3 : equivalent length of volume damper and other component

Type	Δp(inwg)
$\Delta p_T \rightarrow$ air terminal	0.01:0.09
$\Delta p_{VD} \rightarrow$ volume damper	0.04:0.1
$\Delta p_{Flexible}$	0.1
$\Delta p_{Canvous}$	0.02:0.04

- Different branch and main path shown in (Figure 7-13)

Figure 7-13 : branch and main path

7.4.1 Case study on external static losses

From the pervious case study shown in (Figure 7-9) determine the external static pressure

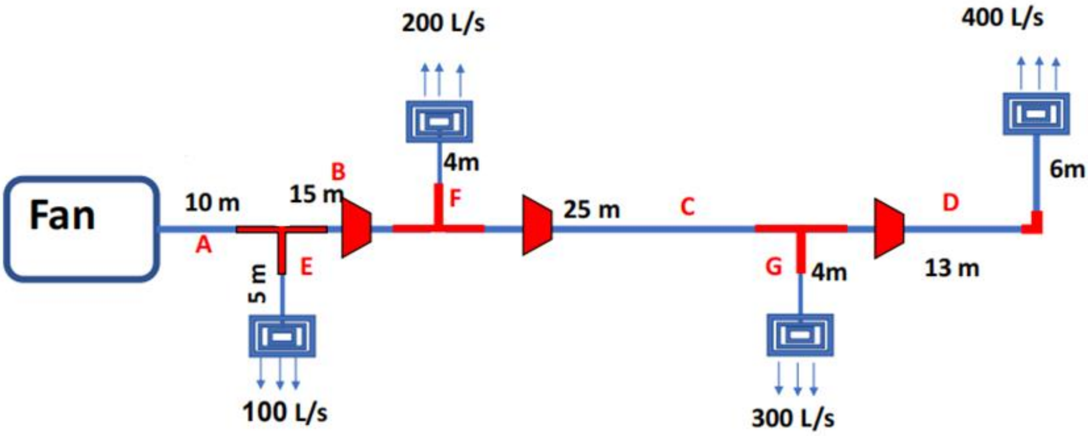


Figure 7-14 : Case study on external static losses

Solution

determine the most difficult path → ABCD

branch	$\Delta p/L$	$\sum L$	$\sum L_{equ}$	Δp_T	Δp_{VD}	$\Delta p_{Flexible}$	$\Delta p_{Canvous}$
ABCD	1.822	10+15+25+13+6=69m	T_B+R+T_B+R+T_B+R+E=8+4+8+4+8+4+6=42m	2 Pa	5 Pa	10 Pa	15 Pa

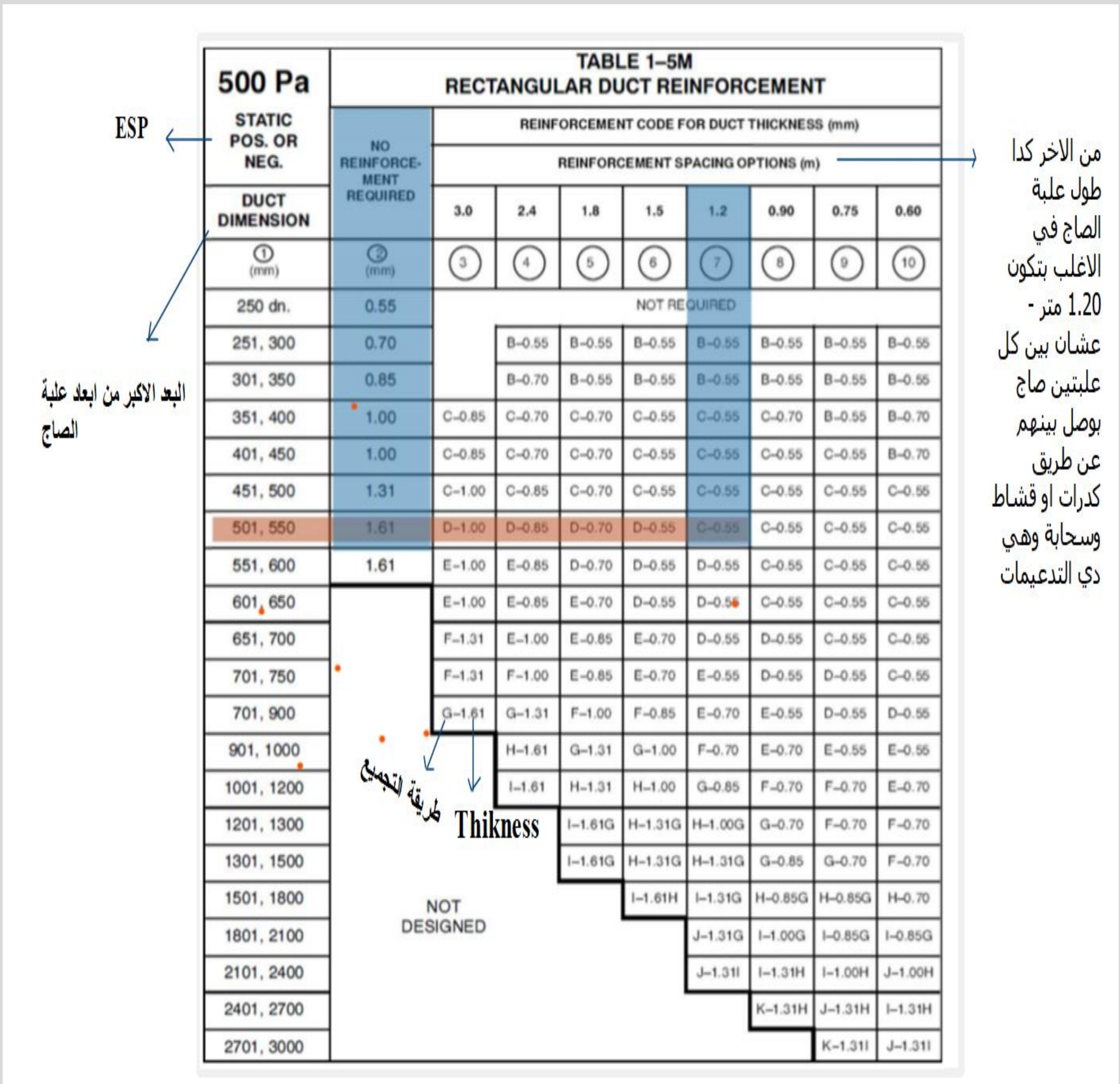
Then:

$$ESP = \left(\frac{\Delta p}{l}\left(L + \sum L_{equ}\right)\right) + \Delta p_T + \Delta p_{VD} + \Delta p_{Flexible} + \Delta p_{Canvous} = 235 \text{ pa}$$

7.5 Thickness of duct

- In order to reach the thickness stage, you must go through the two stages that you missed, which are (Duct sizing-ESP) because with the two information, I will be able to get a thickness

- Keep in mind most of the duct packet when they come to be manufactured, they are often 1.20 meters
- I fear this table for each packet individually, because the dimensions of the duct can change and therefore the thickness changes shown in (Figure 7-15)



500 Pa STATIC POS. OR NEG.	TABLE 1–5M RECTANGULAR DUCT REINFORCEMENT								
	NO REINFORCEMENT REQUIRED	REINFORCEMENT CODE FOR DUCT THICKNESS (mm)							
		REINFORCEMENT SPACING OPTIONS (m)							
DUCT DIMENSION		3.0	2.4	1.8	1.5	1.2	0.90	0.75	0.60
① (mm)	② (mm)	③	④	⑤	⑥	⑦	⑧	⑨	⑩
250 dn.	0.55	NOT REQUIRED							
251, 300	0.70		B–0.55	B–0.55	B–0.55	B–0.55	B–0.55	B–0.55	B–0.55
301, 350	0.85		B–0.70	B–0.55	B–0.55	B–0.55	B–0.55	B–0.55	B–0.55
351, 400	1.00	C–0.85	C–0.70	C–0.70	C–0.55	C–0.55	C–0.70	B–0.55	B–0.70
401, 450	1.00	C–0.85	C–0.70	C–0.70	C–0.55	C–0.55	C–0.55	C–0.55	B–0.70
451, 500	1.31	C–1.00	C–0.85	C–0.70	C–0.55	C–0.55	C–0.55	C–0.55	C–0.55
501, 550	1.61	D–1.00	D–0.85	D–0.70	D–0.55	C–0.55	C–0.55	C–0.55	C–0.55
551, 600	1.61	E–1.00	E–0.85	D–0.70	D–0.55	D–0.55	C–0.55	C–0.55	C–0.55
601, 650		E–1.00	E–0.85	E–0.70	D–0.55	D–0.55	C–0.55	C–0.55	C–0.55
651, 700		F–1.31	E–1.00	E–0.85	E–0.70	D–0.55	D–0.55	C–0.55	C–0.55
701, 750		F–1.31	F–1.00	E–0.85	E–0.70	E–0.55	D–0.55	D–0.55	C–0.55
701, 900		G–1.61	G–1.31	F–1.00	F–0.85	E–0.70	E–0.55	D–0.55	D–0.55
901, 1000			H–1.61	G–1.31	G–1.00	F–0.70	E–0.70	E–0.55	E–0.55
1001, 1200			I–1.61	H–1.31	H–1.00	G–0.85	F–0.70	F–0.70	E–0.70
1201, 1300				I–1.61G	H–1.31G	H–1.00G	G–0.70	F–0.70	F–0.70
1301, 1500				I–1.61G	H–1.31G	H–1.31G	G–0.85	G–0.70	F–0.70
1501, 1800	NOT DESIGNED				I–1.61H	I–1.31G	H–0.85G	H–0.85G	H–0.70
1801, 2100						J–1.31G	I–1.00G	I–0.85G	I–0.85G
2101, 2400						J–1.31I	I–1.31H	I–1.00H	J–1.00H
2401, 2700							K–1.31H	J–1.31H	I–1.31H
2701, 3000								K–1.31I	J–1.31I

Figure 7-15 : Thickness of duct

7.6 Mass of duct

- The mass of the duct is calculated from the following equation:

$$\text{Mass} = 2(W + H) * LxTx\rho$$

Where:

$\rightarrow$ W, H, L, T $\rightarrow$ dimension of duct (m)

$\rightarrow \rho \rightarrow \frac{Kg}{m^3}$ (Stainless = 8000 , Galvanized = 7850 , Aluminum = 2700

7.6.1 case study on mass of duct

- required is to calculate the mass of the duct, knowing that it is made of Galvanized shown in (Figure 7-16)

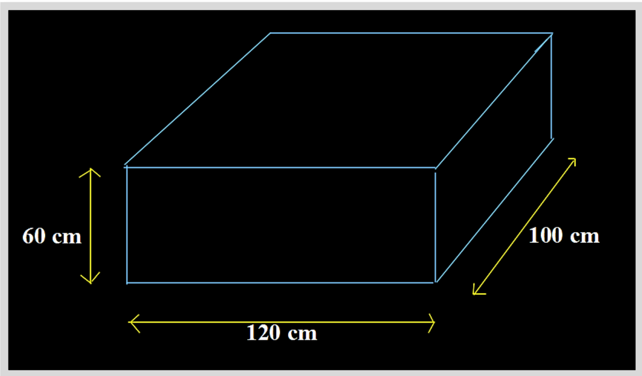


Figure 7-16 : Galvanized steel duct

Solution

- The thickness of the duct from the (Figure 7-15) by ESP and the largest dimension is 1200mm, the thickness is 0.85mm

$$\text{Mass} = 2(W + H) * LxTx\rho$$

$$\text{mass} = 2(1.2 + 0.6) * 1 * 0.00085 * 7850 = 24 \text{ kg}$$

7.7 duct hangers

- The rectangular iron duct is suspended on the concrete ceiling by means of L-shaped galvanized iron supports that are under the duct, and then tapers shown in the (Figure 7-17)

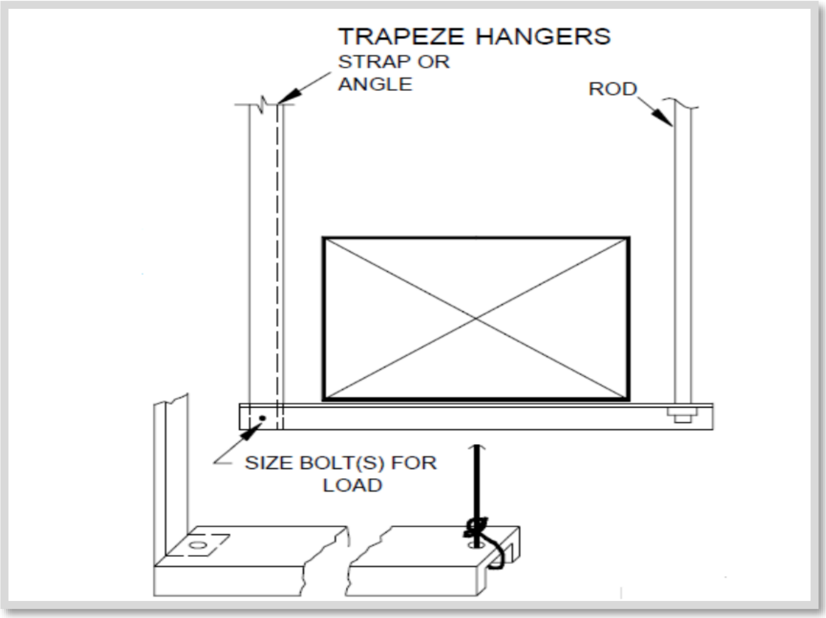


Figure 7-17 : trapeze hanger

- The supports have certain dimensions, as well as the distance between each support and another, and this is determined according to the dimensions of the duct according to the Egyptian code shown in (Figure 7-18)

المسافة ما بين التحاميل (متر)	أبعاد الزاوية (مم)	قطر التيش	طول البعد الأكبر (بوصة)
٣	٢×٢٥×٢٥	٨	حتى ٢٠
٢.٥	٢.٥×٣٥×٣٥	١٠	٢١ إلى ٣٢
٢.٥	٣×٣٥×٣٥	١٠	٣٣ إلى ٤٠
٢	٣×٤٠×٤٠	١٢	٤١ × ٦٠
١.٥	٥×٥٠×٥٠	١٢	أكبر من ٦١

Figure 7-18 : dimensions of the duct according to the Egyptian code

chapter (8): Design of chilled water system

→ Requirements

1) Gpm (flow rate)
2) Pipe size
3) Pressure drops of pump

8.1 Gpm (chiller flow rate)

→ It is the amount of chilled water passing through the pipes needed to remove the load

- ECWT → entering chilled water temperature =54 Fahrenheit =12celsius (in Egypt)
- LCWT → leaving chilled water temperature = 44 Fahrenheit =6 Celsius (in Egypt)
- ΔT → temperature difference = ECWT- LCWT= 10 Fahrenheit or 6 Celsius

→ Form this law → $Q = \dot{m} c_p \Delta T$

Where:

Q: load

Cp : specific heat at constant pressure

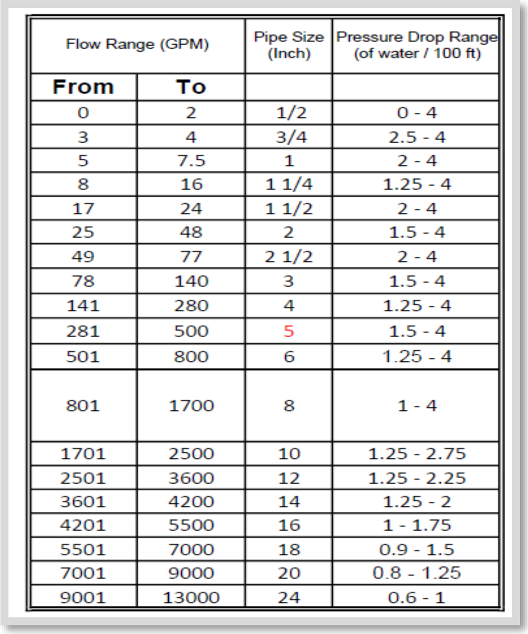
$\dot{m}$: water flow rate

→ The gpm of water

$$GPM = \frac{TR * 24}{\Delta T\ (F)}$$

8.2 Pipe size

- After calculating the load for each space in the project in TR
- GPM is specified inside all pipes
- The diameters of all pipes are determined, but there are several factors that must be taken into account
  - The velocity of water inside the pipes ranges between (2:10 f/s)
  - Pressure loss through the pipes that no more than (10ft/100ft)
- The diameter of the pipes can be determined by the (Figure 8-1) shown below by knowing the chilled flow rate



Flow Range (GPM)		Pipe Size (Inch)	Pressure Drop Range (of water / 100 ft)
From	To		
0	2	1/2	0 - 4
3	4	3/4	2.5 - 4
5	7.5	1	2 - 4
8	16	1 1/4	1.25 - 4
17	24	1 1/2	2 - 4
25	48	2	1.5 - 4
49	77	2 1/2	2 - 4
78	140	3	1.5 - 4
141	280	4	1.25 - 4
281	500	5	1.5 - 4
501	800	6	1.25 - 4
801	1700	8	1 - 4
1701	2500	10	1.25 - 2.75
2501	3600	12	1.25 - 2.25
3601	4200	14	1.25 - 2
4201	5500	16	1 - 1.75
5501	7000	18	0.9 - 1.5
7001	9000	20	0.8 - 1.25
9001	13000	24	0.6 - 1

Figure 8-1: Pipe size by the table

8.3 Pressure drops of pump

- After calculating the load for each space in the project in TR
- The gpm and frication loss are determined at each part of the network
- diameter of the pipes is determined at each part of the network
    - The longest path in the piping network is determined

$$\text{pressure drop} = \text{static head} + \text{residual head} + \text{friction losses} + \Delta p\ \text{chiller} + \Delta p\ \text{fcu} + \Delta p\ \text{AHU}$$

where:

static head → in closed loop is equal zero

residual head→ equal zero

Δp chiller→from supplier (10-20 ft)

Δp AHU → from supplier (8-12 ft)

Δp fcu→ from supplier (4-6 ft)

$$\text{friction losses} = (\text{pipe length} + \text{fitting and valves equavaint length}\ ) * \frac{\Delta p}{l}$$

where:

pipe length→ longest path in the piping network is determined in (ft)

$\frac{\Delta p}{l}$ → equal (4ft/100ft) : (6ft/100ft)

→ fitting and valves equaviant length→based on diameter from the (Figure 8-2)

Equivalent length in meters of pipe for common CHW BS fittings									
			Elbows			Smooth bend tees			
			90° Elb Std.	90° long radius	45° Elb Std.	Tee-branch	Tee-straight	Tee-Reduced 1/4	Tee-Reduced 1/2
	Pipe size (mm)	Pipe size (Inch)						¼d	½d
Threaded	15	1/2"	0.48	0.30	0.24	0.91	0.30	0.42	0.48
Threaded	20	3/4"	0.6	0.42	0.27	1.21	0.42	0.57	0.60
Threaded	25	1"	0.8	0.51	0.39	1.52	0.51	0.70	0.79
Threaded	32	1 1/4"	1.0	0.70	0.51	2.13	0.70	0.94	1.0
Threaded	40	1 1/2"	1.2	0.79	0.64	2.43	0.79	1.12	1.21
Threaded	50	2"	1.52	1.0	0.79	3.04	1.0	1.43	1.52
Welded	65	2 1/2"	1.82	1.25	0.97	3.65	1.25	1.7	1.82
Welded	80	3"	2.28	1.52	1.21	4.57	1.52	2.13	2.28
Welded	100	4"	3.0	2.04	1.58	6.4	2.04	2.74	3.04
Welded	125	5"	3.96	2.5	1.98	7.62	2.5	3.65	3.96
Welded	150	6"	4.87	3.04	2.40	9.14	3.04	4.26	4.87
Welded	200	8"	6.09	3.96	3.0	12.19	3.96	5.48	6.09
Welded	250	10	7.62	4.87	3.96	15.24	4.87	7.01	7.62
Welded	300	12"	9.14	5.79	4.87	18.29	5.79	7.92	9.14
Welded	350	14"	10.36	7.01	5.48	20.73	7.0	9.14	10.36

Equivalent length in meters of pipe for common valves used in chilled water system							
			Globe valve	DRV	Gate valve	Swing check valve	Strainer
	Pipe size (mm)	Pipe size (Inch)					
Threaded	15	1/2"	5.48	2.13	0.21	1.82	0.91
Threaded	20	3/4"	6.70	2.74	0.27	2.43	1.21
Threaded	25	1"	8.84	3.65	0.30	3.04	1.52
Threaded	32	1 1/4"	11.58	4.57	0.45	4.26	2.74
Threaded	40	1 1/2"	13.10	5.48	0.54	4.87	3.0
Threaded	50	2"	16.76	7.31	0.70	6.09	4.26
Flanged	65	2 1/2"	21.03	8.84	0.85	7.62	8.53
Flanged	80	3"	25.60	10.67	0.97	9.14	12.80
Flanged	100	4"	36.58	14.32	1.37	12.19	18.29
Flanged	125	5"	42.68	17.68	1.82	15.24	24.39
Flanged	150	6"	51.82	21.34	2.13	18.29	33.53
Flanged	200	8"	67.07	25.91	2.74	24.39	45.73
Flanged	250	10"	85.36	32.01	3.63	30.48	57.92
Flanged	300	12"	97.56	39.63	3.96	36.58	76.20
Flanged	350	14"	109.75	47.25	4.57	41.15	91.0
Flanged	400	16"	125.0	54.87	5.18	45.73	102.13
Flanged	450	18"	140.24	60.97	5.79	50.30	112.0
Flanged	500	20"	158.53	71.64	6.70	60.90	-
Flanged	600	24"	185.97	80.79	7.62	73.17	-

Figure 8-2 : fitting and valves equivalent length

8.4 Hook up of AHU in project

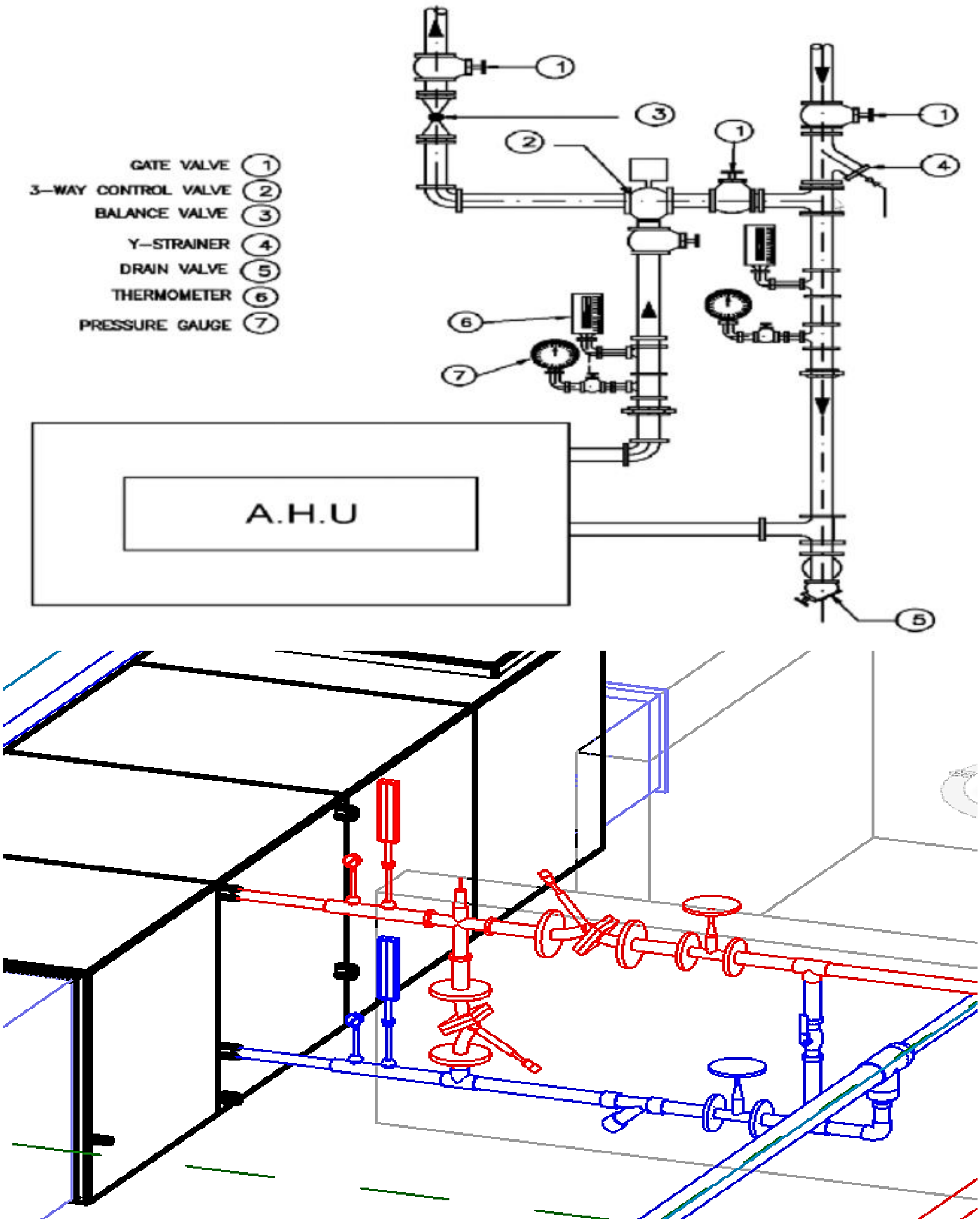


Figure 8-3 :Hook up of AHU

conclusions

At the end of this report, I was able to know (Air Conditioning Systems and Ventilation systems and load calculation manual, by using program, design of operation room, air distribution system, design of chilled water and distribution Air terminal outlets)

In conclusion, a HVAC (Heating, ventilation, and air conditioning) system is a very important component of a healthy, comfortable, and energy-efficient building. There are various types of HVAC installations depending on the location and size of a building. The three common types are centralized, packaged and individual or decentralized.

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