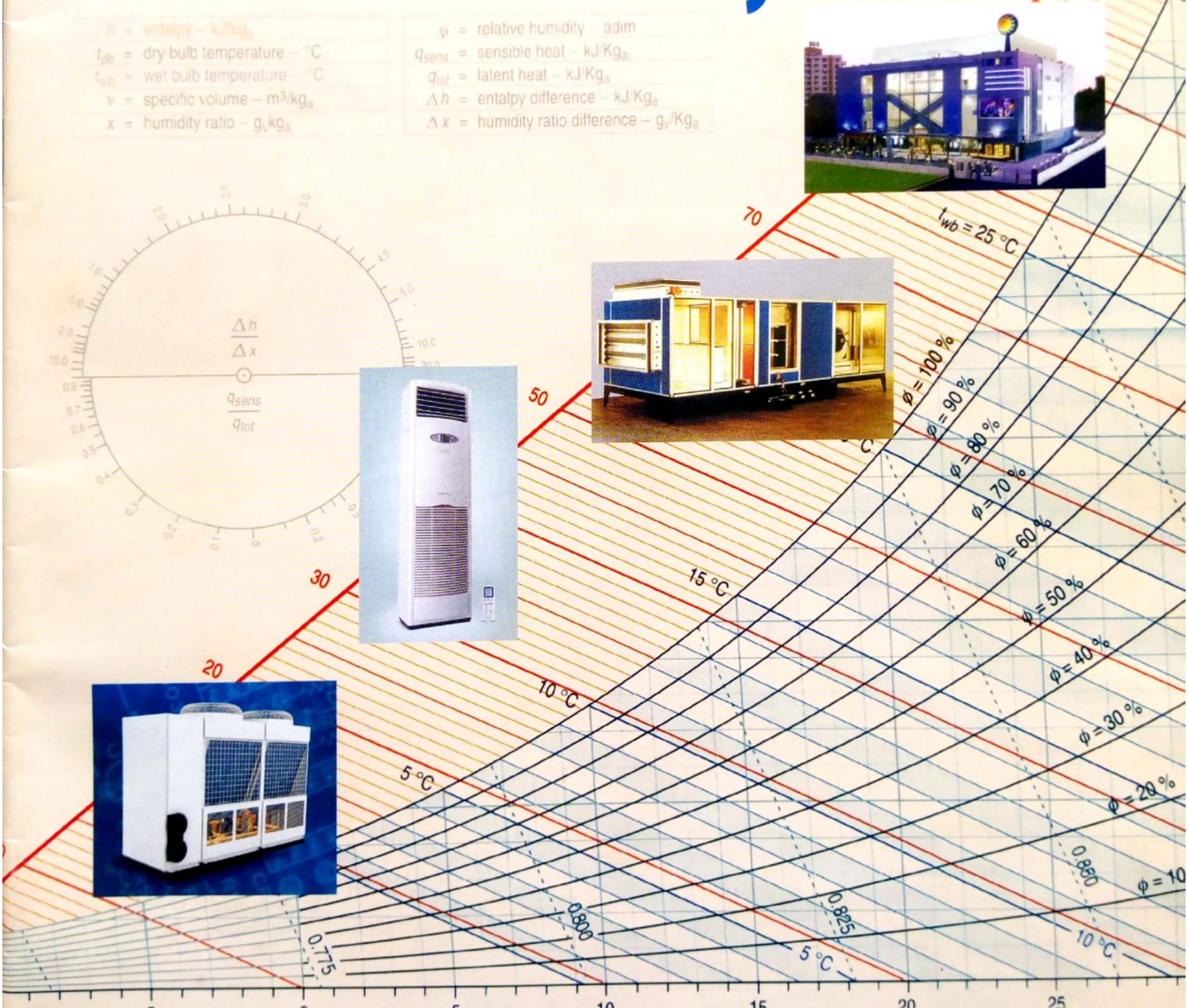
# Basics of Air Conditioning

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# Basics of Air Conditioning

Part 1

## Introduction to Air Conditioning and Human Comfort

Refrigeration is a process of producing cold or a process of taking away heat.

Whenever the temperature is to be lowered below the surrounding temperature, it has to be done through a process of cooling.

Heat always flows from a higher temperature region to a lower temperature region. If we have to lower the temperature below surrounding temperature, it requires external energy, as we would then be working against the laws of nature.

To understand this better, we can take the example of water. As we know, water flows from a higher level to a lower level, irrespective of the quantity involved. And if we want to raise water to the higher level, we require pumps which in turn require external energy.

Exactly similar is the case with heat energy. We therefore require external energy when using a refrigeration system with a compressor to pump out the heat from the place, where we do not want it to a place where it is not objectionable to discard this heat.

Air conditioning is a form of refrigeration. When we use air as a medium for cooling, we call it air conditioning.

## Definition

Air conditioning can be defined as the process of treating air so as to control simultaneously its temperature, humidity, cleanliness and distribution to meet the requirements of the conditioned space.

Although the primary objective of air conditioning is cooling, the term air conditioning has much wider meaning as described in its definition.

## Expectations

If we look at it closely, an air conditioning system must be able to treat air so as to :

- **Cool or Heat** to regulate temperature
- **Humidify/ Dehumidify** to regulate moisture content

**About the Author**

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- **Filter** to reduce contaminants
- **Ventilate** to regulate fresh air requirements air quality and odor
- **Circulate** to regulate air motion-no drafts/ no suffocation

The system, which takes care of all these aspects can truly be called a complete air conditioning process. Most of the products really perform a partial air conditioning process including the room air conditioner, as the standard room air conditioner does not have special components to control humidity or a heating device.

Air conditioning applications can be divided in two categories. Comfort air conditioning and process air conditioning.

## Comfort Air Conditioning

The comfort air conditioning applications relate to human comfort. Here the condition of surrounding air needs to be treated so that persons occupying the place feel comfortable. So the primary objective is human comfort for the people working in the premises.

## Process Air Conditioning

The objective of process air conditioning is to maintain the conditions which will help in carrying out the process effectively and satisfactorily. Process air conditioning therefore involves the objects or the medium to be cooled

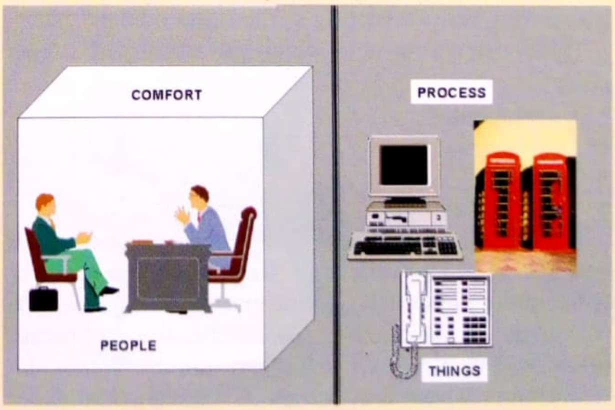


Figure 1 : Comfort and process air conditioning

as a focal point, and the comfort experienced by people is incidental and not the main design consideration.

The conditions to be maintained in process applications depend upon the nature of the process or material being handled and the controlled parameters therefore vary greatly from process to process.

Some of the major process air conditioning applications are :

1. Textile air conditioning - here the humidity to be maintained is the primary concern
2. Printing and paper manufacturing - requires close temperature and humidity control
3. Electronics/ computer rooms - air cleanliness, lower temperatures and close humidity control
4. Pharmaceuticals - IAQ, positive static pressure in the space
5. Hospitals/operation theatres - quality of air/ contamination control
6. Music and sound recording rooms - acoustics
7. CNC machine control panels/air conditioners for control rooms and switch rooms-requires unit design with very high sensible heat load capacity and controlled humidity to ensure no condensation of moisture, which could lead to short circuiting and rusting or very low humidity which could lead to dry contacts.

In this article and in future articles we shall deal with comfort air conditioning applications only.

We shall now look at the various parameters which make a person feel either comfortable or uncomfortable.

## Body Temperature Control

As the diesel engine needs fuel to generate power, the food we eat acts as a fuel and metabolism converts it into a form of energy which enables us to work. Work or any activity, generates heat. This excess heat needs to be rejected so as to maintain the body at the normal temperature of 37° C (98.6° F). When we are in a conditioned space, the body is, therefore, rejecting and transferring heat constantly to the surroundings.

There are four ways by which this heat transfer takes place :

- Conduction
- Convection
- Radiation
- Evaporation

**Conduction** – Whenever our body parts come in contact with any object or air which is at a lower temperature than the body temperature, the heat will be transferred by conduction. Cooler the surrounding air, faster will be the heat leaving the body. When the temperature is very low, occupants will complain about feeling too cold and when the temperature is very high, they will complain about feeling too hot.

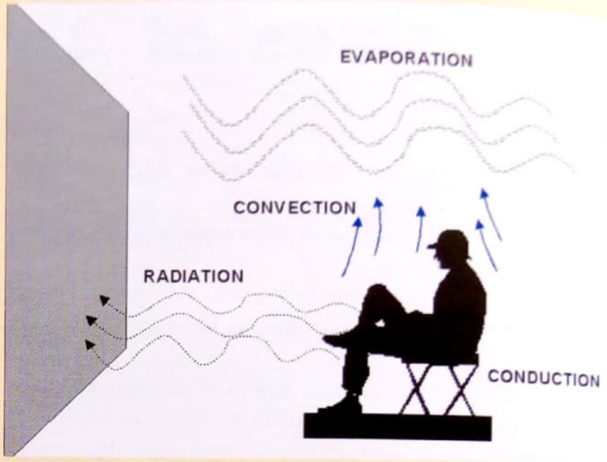


**Figure 2 : Four methods of heat transfer**

**Convection** – When the body is rejecting heat, the air closer to the body becomes warmer than the air which is away from the body. As warm air is lighter than cold air, it rises upward and is being replaced by the cold air. Convection currents are therefore set in motion and heat is thus rejected to the surrounding air. If the fans are circulating air then the heat transfer is faster as it would amount to forced convection which is faster than natural convection heat transfer.

**Radiation** – This heat transfer takes place due to temperature difference between the body and the objects in the premises which are at a lower temperature than the body. Radiant heat travels from a warmer body to colder objects or surfaces like walls, furniture etc. without heating the medium between the two.

**Evaporation** – It is the fourth way in which heat is transferred from our body to the space surrounding it. Moisture, what we call perspiration is released from the pores of our skin. As the moisture evaporates, the heat needed for such evaporation is provided by our body and in turn the body cools. This evaporation process is going on constantly whether we notice it or not. When we start sweating and the drops of perspiration appear on the skin, it means the rate at which the body is rejecting heat is faster than the rate at which the surrounding air can absorb it. This could be either due to excess moisture (or a higher relative humidity) already present in the air or due to a higher activity level, when the body is generating more heat than it can reject at a normal rate. Since most of the heat is rejected to the environment through the skin, it is often convenient to define heat production per unit area of skin. For a resting person it is 58.2 W/m². As the activity level increases, this rate also goes up. The average person has a skin area of about 1.8m².

### Respiration Losses

During respiration, the body looses both sensible and latent heat by convection and evaporation, since air inhaled is at the surrounding temperature, whereas exhaled air is warmer and nearly saturated.

All these four processes of heat rejection from the body to surroundings are taking place simultaneously. However, depending on the individual conditions, one form of heat transfer is more active compared to the other three.

Temperature levels considered as comfortable vary depending on many factors. Some of them are - level of activity, nature of activity, type of clothing, personal preferences, age, sex, geographical conditions, non-uniform temperatures within the space etc.

There is no rigid rule as to the best atmospheric conditions, when all people will feel comfortable. Most people can tolerate higher temperatures in summer and lower temperatures in winter without sacrificing comfort level to any noticeable amount. It is also impossible to satisfy all the people occupying the space for the reasons mentioned above. Extensive surveys have been conducted and charts have been published to show when majority of occupants feel comfortable. The ASHRAE Chart showing the summer and winter comfort zones when 80 percent of people feel comfortable is shown in Figure 3.

The defined comfort temperature range is 20-26° C, with 24° C(75° F) as a benchmark, when 80 percent of people find the environment thermally acceptable. Similarly humidity conditions are comfortable from 30 to 60% with 50 percent as a benchmark.

The velocity of air over the skin is also important,

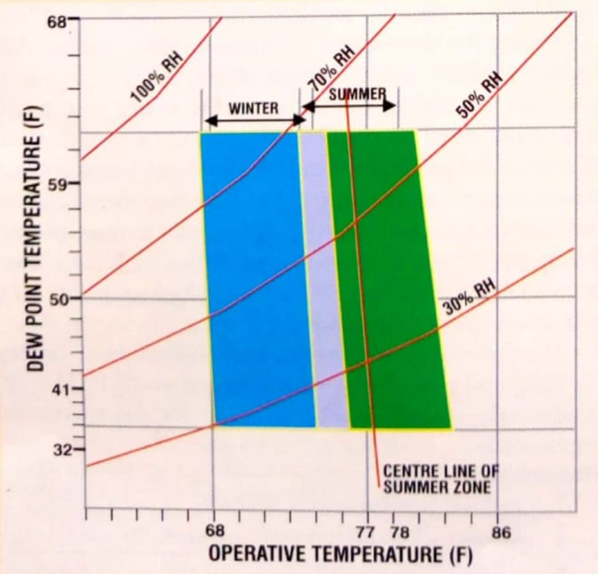


Figure 3 : ASHRAE Summer and Winter Comfort Zones

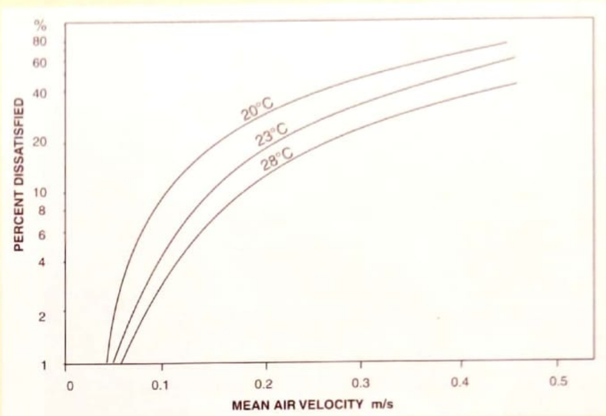


Figure 4 : Effect of air movement on comfort

influencing the feeling of comfort. Figure 4 indicates the percentage of people feeling discomfort as the mean air velocity changes from 0.08 to 0.25 meters per second.

As can be seen from these two figures if one condition is altered, the feeling of comfort requirements for other parameters also changes.

### Factors Affecting Human Comfort

Temperature, relative humidity and air motion are three factors which affect the body's ability to reject heat.

### Temperature

Greater the temperature difference, faster is the heat flow. The body may reject heat faster than required and we may feel cold. As the air temperature approaches the body temperature, the body looses heat at a slower rate and the convection mode becomes less dominant. We start feeling hot. Thus air temperature plays a major role for the body to be comfortable.

### Relative Humidity

Whenever the temperature is raised or lowered the relative humidity changes. Cool air has less capacity to hold moisture compared to warm air. When the surrounding air has lower humidity the body is able to

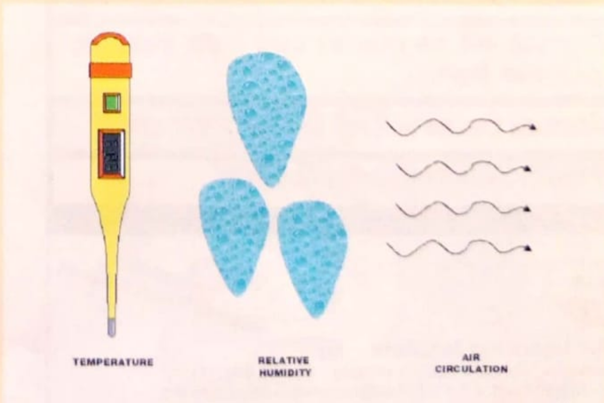


Figure 5 : Factors affecting human comfort

give away more heat through evaporation and conversely, when air is more humid evaporation diminishes. Evaporation takes away maximum heat since each kg of water evaporated removes nearly 2500 KJ (1000 Btu/lb).

It is often observed that conditions of slightly warmer temperature but lower humidity are more comfortable than one with lower temperature but higher relative humidity. Therefore lowering room RH enables use of higher dry bulb temperature in some cases.

Maintaining proper RH also depends upon correct sizing of the plant. Over designing is also detrimental as the plant operating time would then be shortened, leading to higher humidity conditions than designed. This is due to the fact that dehumidification takes place only when the plant and the compressor is in operation.

**Air Motion**

Air motion affects the heat rejection from the body. Forced air convection accelerates the rate of evaporation. If there are no air currents, air surrounding the body gets saturated. At this point evaporation would almost stop and result in discomfort and suffocation. Too high a velocity results in undesired local cooling and draft. Draft is considered as a most annoying factor in offices and people tend to demand higher temperatures in the room or stoppage of the ventilation system. Higher velocities also contribute to a higher noise level which becomes undesirable and irritating beyond a certain limit.

**Outdoor Air**

A certain amount of outdoor air needs to be delivered to a conditioned space in order to prevent noticeable odors. In addition to the odor problem, there is also the problem of keeping the atmosphere reasonably clean. Indoor Air Quality (IAQ) is receiving more attention these days as it has been established that if air quality is not monitored and maintained, it could lead to sickness and loss of efficiency. Instead of improving productivity because of air conditioned comfort, the findings show that the efficiency deteriorates and absenteeism increases due to more people falling sick if the IAQ is not proper. When such cases have been analyzed it has been found that indoor air quality is in fact much inferior to outside fresh air.

The designer's goal therefore is to study all these aspects carefully and provide the system which would be able to produce the desired level of comfort for the particular application.

**Reference**


1. Ashrae Fundamentals Volume
2. Carrier Corporation training material


**Next Issue** : Properties of air and the psychrometric chart ❖

# Basics of Air Conditioning

## Part 2

*This series of articles by Ramesh Paranjpey covers the basic principles of air conditioning. The articles will serve as a source of reference for newcomers joining our industry as well as for experienced engineers wanting to brush up on fundamentals*

### Properties Of Air

In the first article, we covered fundamentals, such as what is comfort, what does air conditioning mean, difference between process and comfort air conditioning, understanding heat and different ways in which heat transfer takes place, as well as factors affecting human comfort.

In order to air condition the space for human comfort, we supply cool, dehumidified air. This air absorbs heat and moisture from the space to be cooled, and helps in maintaining comfort temperatures between 23°C to 25°C with relative humidity of 40% to 60%.

Thus, we use air as a medium for absorbing heat. Air conditioning engineers therefore, have to deal with air and hence understanding the properties of air, while learning the basics of air conditioning is essential.

We shall, therefore now study the properties of dry and moist air and the basics of psychrometrics in this second part.

The air surrounding us and used for air conditioning is never absolutely dry and has some moisture in it. The moist air, comprises of two components i.e. dry air and water vapour.

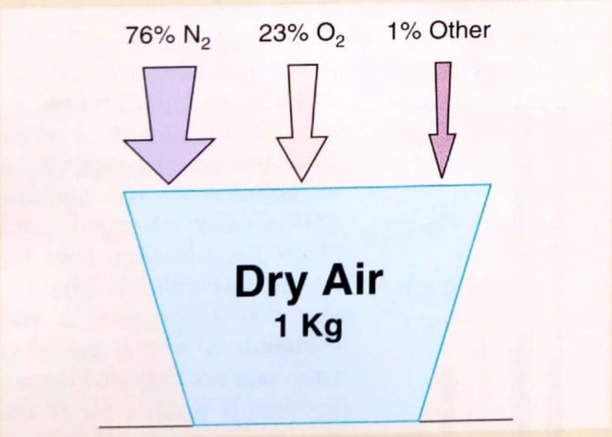


Figure 1: Composition of dry air

**About the Author**
Ramesh Paranjpey is a mechanical engineer with an M.Tech in refrigeration from IIT Bombay. He has 35 years' experience starting with Kirloskar Pneumatic in Pune in their ACR projects division. Later, he joined Carrier Transicold at Bangalore as their first M.D. and subsequently as Director Projects in Singapore. For the past 5 ½ years he has been working as CEO of Voltas-Air International Pune manufacturing car air conditioners and mobile defence equipment.
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Dry air itself is a homogeneous mixture of a number of gases and behaves as a pure substance.

The major constituents of dry air are Nitrogen and Oxygen gases.There are also traces of other gases present such as Argon, Carbon Dioxide, Hydrogen etc.

Dry air is a perfect gas and follows the gas laws and is considered as a fixed part of the moist air. The other constituent of moist air is water vapour, which is considered as a variable part.

Moist air is not a pure substance since, when air is cooled or heated condensation and evaporation of moisture occurs.

The amount of actual water vapour present in the air is very small and is measured in grams. Since 1000 grams is one kilogram, this water or so called humidity does not have much influence on the actual weight of air. The weight of moist air is predominantly that of dry air and a small portion of water vapour it contains.

The presence of water vapour in the air is due to many factors. Evaporation from oceans, rivers and lakes puts water in the air and in the clouds. In space, water is added

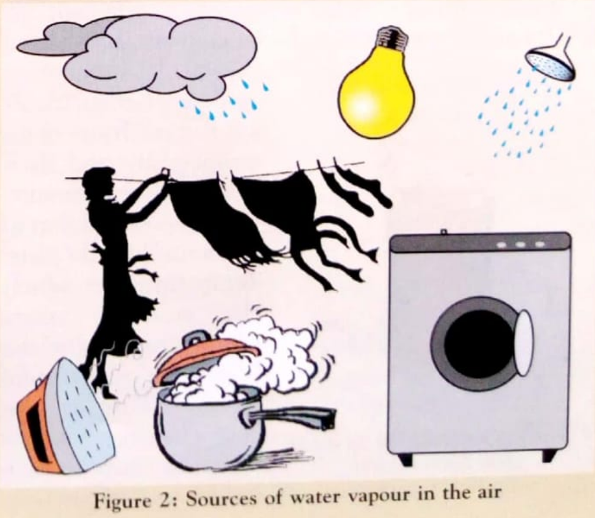

Figure 2: Sources of water vapour in the air

due to the presence of people, food , fruits, vegetables, cooking processes, bathroom showers, washing machines or any other process where moisture is involved.

### Dalton's law of Partial Pressures

If dry air and water vapour is mixed in a container, the pressure exerted by this mixture is the addition of the pressure exerted by each of the two gases.

In other words each constituent exerts the same pressure as if it alone was present in the space occupied by the mixture.

The volume occupied however, is the same as if the entire volume is occupied by each gas at its partial pressure.

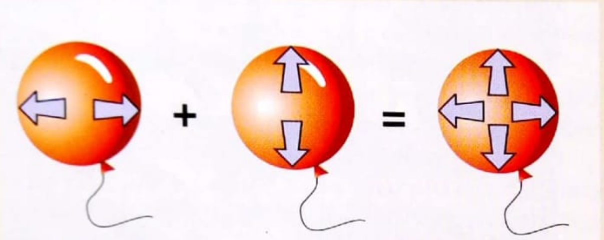


Figure 3: Dalton's law of partial pressures

The total enthalpy of the mixture is the sum of the enthalpies of each constituent at its partial pressure.

### Saturated Air

When air and saturated water vapour occupy the same volume, we say air is saturated. In reality what we mean is, only the water vapour is saturated. We shall however call it saturated air since this term is widely used and accepted in the air conditioning industry.

### Dew Point Temperature

When moist air is cooled at constant pressure, initially it will remove superheat, if the air contains superheated vapour. If the cooling continues further, the air will first reach saturation temperature and then below this temperature, water vapour will start to condense into droplets. Temperature at which the moisture starts condensing, is called the "dew point" temperature of air. This is the temperature below which air cannot simply hold any more moisture.

Figure 4: Condensation on a cold glass demonstrates 'dew point' temperature

### Dry Bulb Temperature

This is the temperature of air as indicated by an ordinary thermometer

### Wet Bulb Temperature

This is the temperature registered by the thermometer whose bulb is covered by a wetted wick over which air is moving at a velocity of 2.5-10.0 m/s .

If we take two thermometers showing the same dry bulb temperature, and then cover one thermometer bulb with a wetted wick, and swiftly rotate them in a stream of moving air, the temperature of the thermometer whose bulb is covered with the wetted wick will drop, until it reaches a stable point.

The air flowing past both the thermometers is at the same temperature, however as the water evaporates from the wick and the air surrounding the wick gives up the required heat for this evaporation , the temperature of the thermometer drops.

The latent heat required for the evaporation of water is provided by the sensible heat of the surrounding air .

It is therefore obvious that, if the air surrounding the wick is already wet, we would not see much wet bulb temperature depression, whereas, drier the air flowing past the wick of the wetted thermometer, the larger will be the amount of moisture evaporated into the air stream and lower will be the temperature reading of the wet bulb thermometer.

The difference between the readings of wet and dry bulb thermometer is called "wet bulb depression."

To enable movement of the air rapidly across the bulb of the thermometer, we use an instrument called a sling psychrometer. The wick of the wet bulb thermometer should be periodically changed, and cleaned as it becomes hard due to water minerals left behind or the dirt /dust settled and accumulated over a period of time, may not indicate a correct reading. It is advisable to use distilled water in the pot in which the wick is dipped and kept continuously wet.

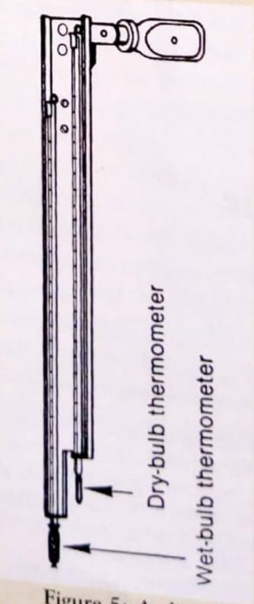


Figure 5: A sling psychrometer

Besides the dry bulb and wet bulb temperatures, we also need to understand a few more properties of air so that we can deal effectively with the air when we are treating it.

## Standard Air

ASHRAE defines standard air as dry air at sea level having an atmospheric pressure of 101.325 kPa, at 20°C. (with FPS units the standard conditions are 70°F,14.696psia)

## Atmospheric Pressure

The weight of air pushing down on the earth is referred to as atmospheric pressure, and a barometer is used to measure the same. We are living at the bottom of a sea of air that has weight. When we start climbing on mountains we have less weight of air column on top of us, which means the atmospheric pressure reduces as we go up from sea level.

The atmospheric pressure at sea level is 101.325 kPa or 760 mm of mercury column

## Specific Density

Dry air conforms closely to a perfect gas and follows the gas laws. This means that if the air is heated at constant pressure, the air would expand and therefore weigh less per unit volume. This property is defined by specific density.

Specific density means weight of air per unit volume. The unit of measure is kg /$m^3$

The specific density of air at standard conditions is 1.204 kg/$m^3$ (0.075lb/$ft^3$)

It is also important to remember, (contrary to our normal belief) that dry air weighs more than wet or moist air. Normally when it is cloudy or raining or the air is damp, we say that there is heaviness in the air, however in reality the air is actually lighter in such situations.

The reading of a barometer is maximum on sunny days indicating that such air has more weight.

As a storm approaches or the clouds gather, the barometer level drops, indicating that the air is exerting less pressure. This is due to increased presence of moisture in the air. As the water vapour displaces dry air to make up moist air it weighs less than it did when it was dry. This is how the weather reporting stations predict whether its going to rain or whether a storm is approaching.

Once we study the psychrometric chart this property of air will be easily understood.

## Specific Volume

Specific volume is the reciprocal of density, or specific volume means one over specific density. Specific volume indicates how much volume or space one kilogram of air will occupy. The unit of measure is $m^3$/kg.

As the air is heated its specific volume would increase, which means its density will decrease.

At higher temperatures, air weighs less per unit volume and therefore warm air rises, where as cold air tends to settle down as its density is more compared to hot air.

The specific volume of standard air is 1/1.204 or 0.830 $m^3$/kg (13.33$ft^3$/lb)

This property of air helps us in determining fan performance and selecting fan-motor sizes.

We have read earlier about dew point temperature. We now need to understand relative humidity and specific humidity.

## Relative Humidity

It is the ratio of the actual partial pressure of water vapour to its saturation pressure corresponding to the same dry bulb temperature.

This is the ratio of the amount of moisture present in the air to the amount, the same air can hold at saturation at the same temperature. It indicates the ability of air to absorb additional moisture and is expressed as a percentage.

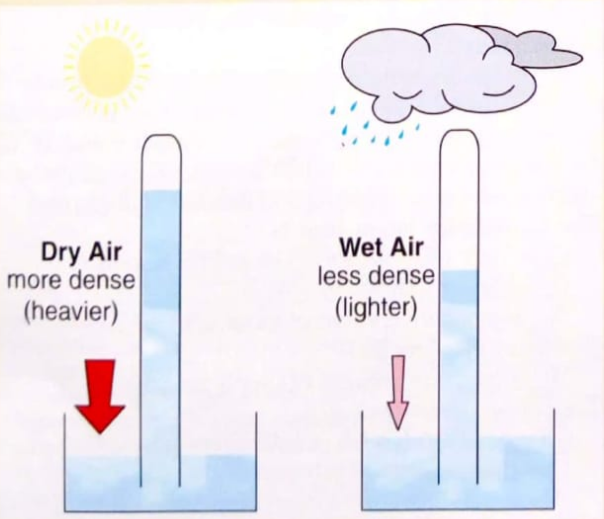


Figure 6: Dry air is heavier than moist air

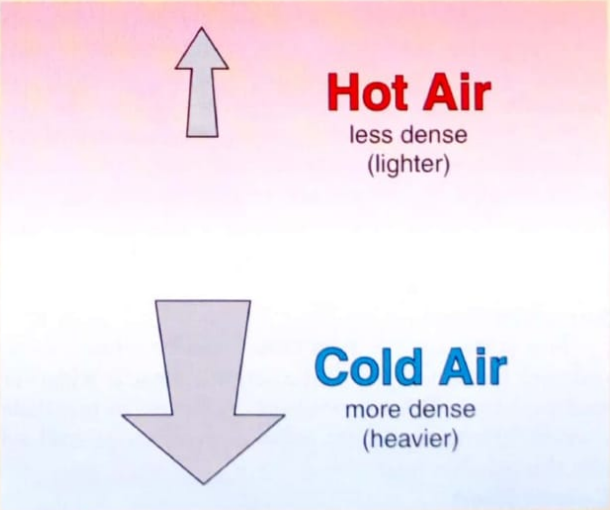


Figure 7: Cold air is heavier and warm air is lighter

For example, when we say air has a relative humidity of 50%, we are really indicating that the air at that specific temperature contains only half the amount of moisture compared to what it can actually hold. 100% relative humidity means air cannot hold any additional moisture at that temperature. We call this air "saturated air."

Since relative humidity is directly dependent on temperature, whenever the temperature is raised or lowered, the relative humidity will also change.

As the air is cooled, its ability to absorb additional moisture diminishes.

### Specific Humidity

Specific humidity indicates the amount of moisture present in the air and is expressed in terms of grams of moisture per kg of air.

Specific humidity is a more dependable indication of actual moisture present in the air. We are more used to relative humidity terminology in day to day life, but it really is not able to tell us how much moisture the air is actually holding per unit weight. Relative humidity can, many times, give us a misleading impression about the condition of air.

For example, cold air at 10°C may have a relative humidity of 80% , and air at 35°C may have a relative humidity of 40%, giving a wrong impression that 10°C air has more moisture content. In reality 10°C air is drier than 35°C air, as the specific humidity, or the actual moisture content of 10°C, 80% RH air, is much lower at 6g/kg, where as air at 35°C, 40% RH has a much higher moisture content of 14g/kg of moisture.

It is therefore suggested that we should use the specific humidity term more often than the relative humidity term if we want to describe the proper condition of the air.

We have now covered some of the properties of air, however, since air conditioning engineers have not only to deal with quantity of air handled but also the movement of heat, we should understand some terminologies related to heat.

### Specific Heat

Specific heat is another important property of air, indicating its ability to get hot or cold compared to water. It is measured by the amount of heat in kJ required to raise the temperature of one kg of a substance by one degree centigrade. Standard air has a specific heat of 1.006 kJ/kg.°C (0.24 Btu/lb.°C)

### Sensible Heat

This is the form of heat which can be sensed on an ordinary thermometer. When sensible heat is added or removed from the air, without a change in moisture content, the thermometer registers this change, and we call this sensible heat.

### Latent Heat

Whenever there is phase change , latent heat transfer is taking place. Latent heat is the amount of heat necessary to change a quantity of water to water vapour, without changing its temperature or pressure. When we boil water, the boiling point is 100°C at sea level. If we continue to apply further heat, the water will get converted to vapour and the heat required for vaporization is known as latent heat. When this water is getting converted into steam, the thermometer will still indicate 100°C. Latent heat is therefore not registered on an ordinary thermometer. Likewise when latent heat is removed, we call it condensation. Latent heat is always associated with moisture. Latent heat of vaporization of water at atmospheric pressure is 2256.28kJ/kg (970 Btu/lb)

### Total Heat (Enthalpy)

Total heat is the sum of sensible and latent heat. Changes in wet bulb thermometer readings are indicative of changes in total heat.

When cooling air, there is no change in its latent heat content as long as its dew point temperature remains constant. In such a case only sensible heat is removed. On the other hand, if the air is to be cooled to a point below its dew point, both sensible as well as latent heat must be removed. The total heat removed is, what needs to be considered while doing air conditioning cooling load calculations. The refrigerating system must have sufficient capacity to remove both the sensible heat of air and the latent heat, in order to condense water vapour.

We shall now look at some of the important formulae for cooling load calculations before we start plotting on a psychrometric chart.

**Sensible Heat :**

$Q_s = 1.23 \times \Delta T \times L/s$ or in FPS system, $Q_s = (1.1 \times \Delta T \times \text{cfm})$

$\Delta T$ is the dry bulb temperature difference between entering and leaving conditions of air in °C (°F).

L/s (cfm) is the volume flow rate of air handled.

### Latent Heat

In air conditioning applications, the industry standard for evaluating the latent heat capacity of air is based on the energy content of 50% relative humidity and 24°C dry bulb temperature, with 10°C condensate temperature which is normal for cooling and dehumidifying coils.

**The formula for latent heat is**

$Q_l = 3.0 \times (W_A - W_B) \times L/s$ or in FPS system

$Q_l = [0.69 \times \text{cfm} \times (W_A - W_B)]$

$W_A$ = moisture content of room, g/kg (lb/lb) – g(moisture)/kg (dry air)

$W_B$ = moisture content of supply air, g/kg

**Total heat (Enthalpy)**

$Q_t = 1.19 \times \Delta h \times L/s$ or in FPS system

$Q_t = (4.4 \times \text{cfm} \times \Delta h)$

$\Delta h$ is the difference in enthalpy between room and coil leaving condition.

# Basics of Air Conditioning

## Part 3

### Understanding the Psychrometric Chart

Having learned the properties of air, we shall now look at these properties as to how they are reflected on the psychrometric chart.

In the year 1911, Dr. Willis Carrier presented to the American Society of Mechanical Engineers, the "psychrometric" formulas and the psychrometric chart.

"Psychro" means cold and "metrics" means measurement. In reality, the chart helps us in more ways than just the measurement. It is a graphical representation of air properties we have so far learned.

We shall now look at various lines appearing on the chart such as, dry bulb temperature, wet bulb temperature, enthalpy, specific humidity, relative humidity, dew point and specific volume.

The charts are available for normal, low and high temperature ranges at sea level as well as at levels of 750m, 1500m and for other elevations. We shall, however, refer to a normal temperature and sea level condition chart, which gives fairly accurate results as needed by most practicing engineers.

### Dry Bulb Temperature (Figure 1)

The vertical lines parallel to 'Y' axis are dry bulb lines and the 'X' axis represents dry bulb temperature scale ranging from –10°C to +55°C

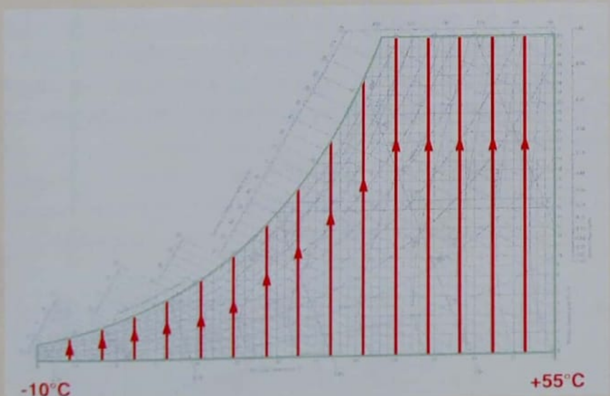


Figure 1 : Dry bulb temperature lines (°C)

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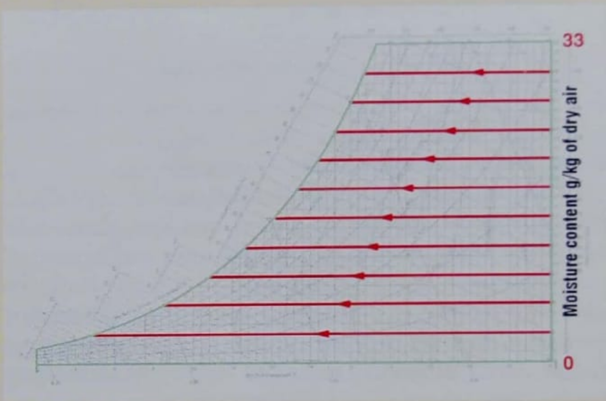


Figure 2 : Specific humidity lines

### Specific Humidity (Figure 2)

The 'Y' axis represents moisture content in gms per kg of dry air starting from 0 at the lower end and extending to 33 gms at the top. These are horizontal lines running from right to left.

### Relative Humidity (Figure 3)

These lines look very much like saturation lines on the chart. They are in increments of 10% starting from the right hand corner to the left. The extreme left line is 100% saturation line and any condition of air beyond

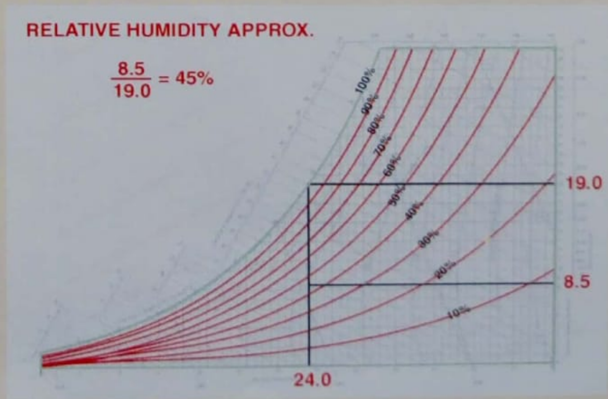


Figure 3 : Relative humidity lines

**About the Author**

Ramesh Paranjpey is a mechanical engineer with an M.Tech in refrigeration from IIT Bombay. He has 35 years' experience starting with Kirloskar Pneumatic in Pune in their ACR projects division. Later, he joined Carrier Transicold at Bangalore as their first M.D. and subsequently as Director Projects in Singapore. For the past 6 years he has been working as CEO of Voltas-Air International Pune manufacturing car air conditioners and mobile defence equipment.
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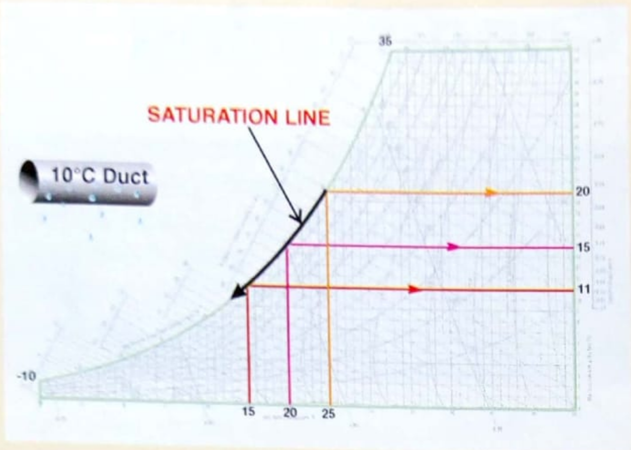


Figure 4 : Dew point temperatures

this line means there is free moisture present in the air, either in the form of fog or mist.

**Dew Point Temperature** (Figure 4)

The dew point temperature lines run horizontally to be read from left to right and coincide with specific humidity lines, but the scale is different and is marked on the saturation line from –10°C to +35°C. It can be observed that any point lying on the saturation line means its dry bulb, wet bulb and dew point temperature is the same. When air is cooled below the dew point temperature, there would be condensation.

**Wet Bulb Temperature** (Figure 5)

These lines run at about 30° angle to 'X' axis and the scale ranges from –10°C to +35°C. Both the dew point and wet bulb scales are marked on the 100% saturation line.

**Enthalpy** (Figure 6)

Wet bulb temperature is a measurement of the total heat and when the wet bulb lines, with reasonable degree of accuracy, are extended beyond saturation lines and are divided as per enthalpy scale, they represent total enthalpy of 1 kg of dry air in kJ/kg of dry air. (For the purpose of this article we shall overlook the fact that specific enthalpy lines are different

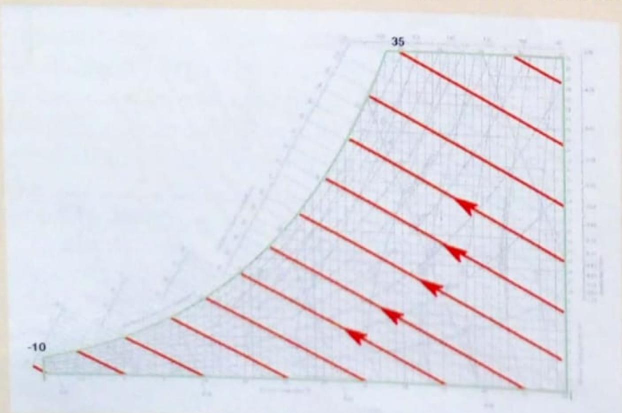


Figure 5: Wet bulb Lines

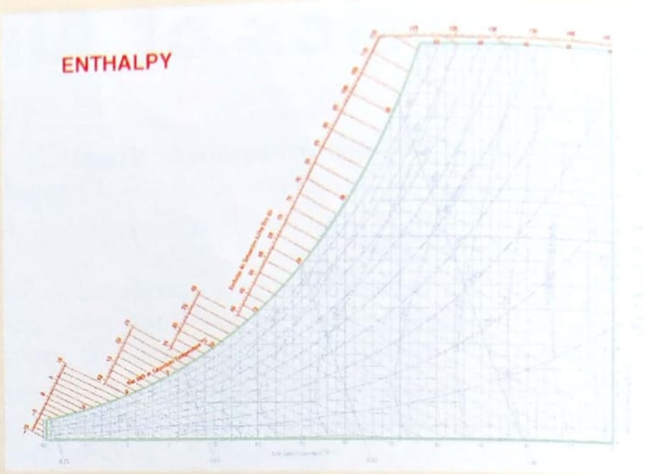


Figure 6 : Enthalpy scale

from wet bulb lines and therefore do not coincide).

**Specific Volume** (Figure 7)

These lines are similar to wet bulb lines but at about 70° angle to 'X' axis. They represent space occupied in cubic meter per kg of dry air. The specific density of air is the reciprocal of specific volume.

**Sensible Heat Factor** (Figure 8)

On the right extreme, beyond 'Y' axis one will find a scale. This scale starts from top at 0.36 to 1.00. This scale indicates sensible heat factor and helps in determining percentage of sensible heat and latent heat contribution in total cooling load. On the chart there is also a point marked at 24°C and 50% R.H. which is used as a reference point.

ASHRAE psychrometric chart uses a protractor to plot the slope of the line representing the sensible heat ratio.

The chart shown in Figure 9 is the complete chart combining most of the lines and other parameters so far discussed:

1. Represents sensible heating.
2. Sensible heating and humidification.
3. Chemical dehydration.

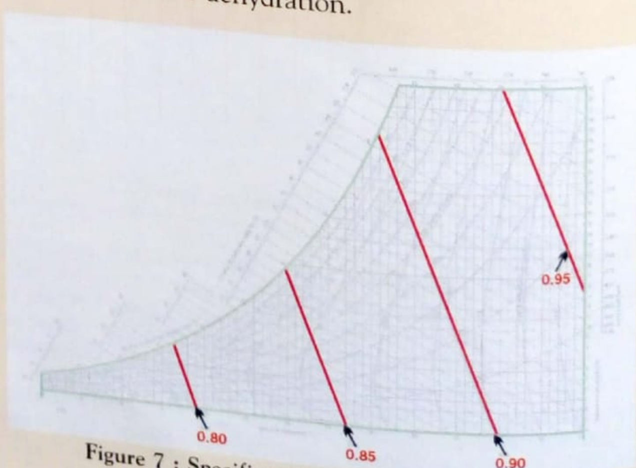


Figure 7 : Specific volume lines $m^3/kg$ of dry air

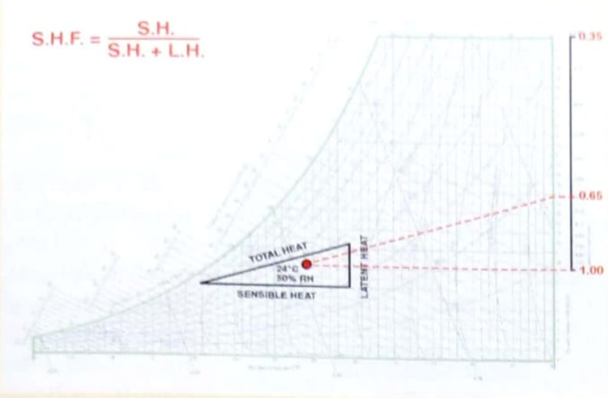


Figure 8 : Sensible heat factor

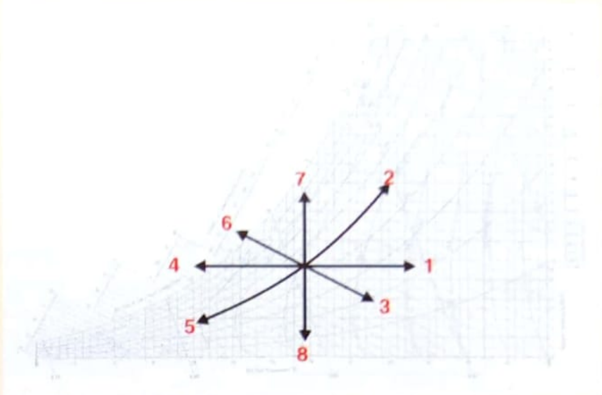


Figure 9 : Combination

4. Sensible cooling.
5. Cooling and dehumidification.
6. Evaporative Cooling
7. Latent heat addition-humidification.
8. Latent heat removal-dehumidification.

## Applied Psychrometrics

The psychrometric chart is the most useful tool for understanding the properties of air. The chart is also very useful in understanding the various processes in which we normally treat air and in designing systems which are cost-effective while giving the desired performance. The chart is also useful in troubleshooting in case the system performance is at variance with design parameters.

We shall now discuss various day-to-day processes which a practicing engineer has to deal with.

## Sensible Heating Process (Figure 10)

This is the process where the temperature of the air is increased without any change in moisture content or specific humidity. This process uses an electric heater and the process reverses when the heater is switched off and air gets cooled to its original temperature. Let us say, air is heated from 19°C dry bulb and 13°C wet bulb to 28°C dry bulb temperature. The points to be noted are:

1. The dew point temperature 8.4°C remains unaltered.
2. The specific humidity 7g/kg remains unaltered, as no moisture has been added or removed.
3. The wet bulb temperature increases to 16.36°C. This is quite natural, since when the air temperature goes up, the temperature of moisture in the air will also go up.
4. The relative humidity decreases from 50% to 29%. This point is important. It means warm air has greater capacity to absorb moisture than cold air.

It also means that if a humidifier is to be used for a warm air furnace it should be placed on the supply air side and not in the return air path, to get better results.

This also explains why relative humidity is higher during early morning hours and decreases as the day progresses.

5. The specific volume of air increases from 0.84m$^3$/kg to 0.86 m$^3$/kg, indicating that as air is heated its density decreases, and cold air, therefore, is heavier/denser than hot air.
6. The enthalpy of air has increased from 36.58 kJ/kg to 45.74 kJ/kg , since we have added heat with the help of a heating coil.
7. If we draw a line parallel to our process line from reference point of 24°C and 50% R.H. to intersect the sensible heat factor scale, we will notice that it intersects at 1.0, indicating once again that it is only sensible heating and zero latent heat, as no moisture has been added or removed.

The sensible

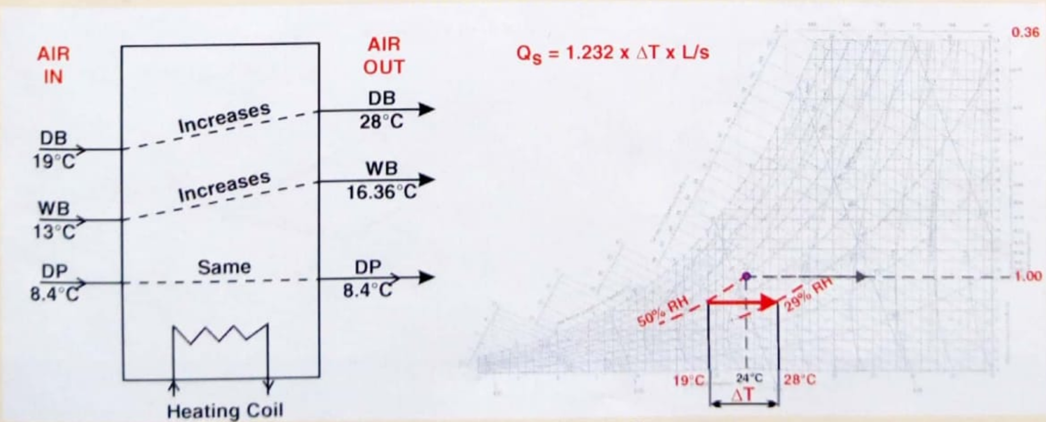


Figure 10 : Sensible Heating Process

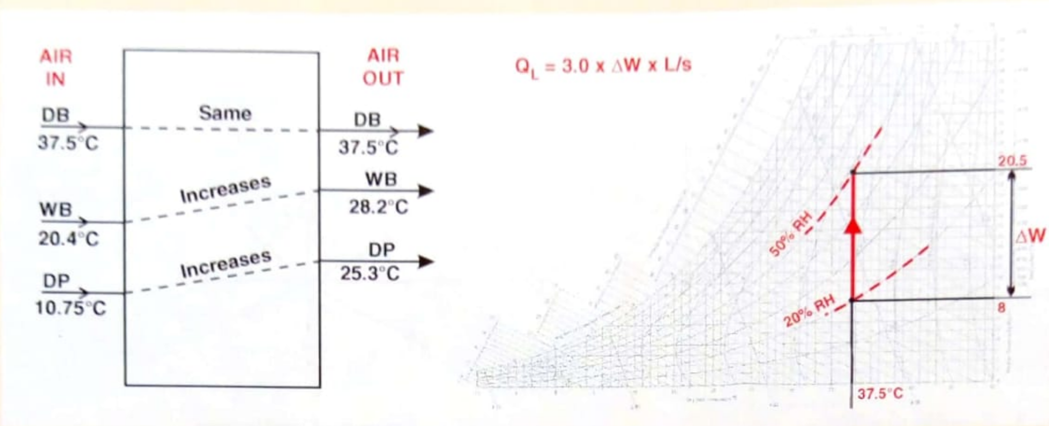


Figure 11 : Latent Heating Process

cooling process is the reverse of this process.

**Latent Heating Process** (Figure 11)

A latent heat change occurs when water is evaporated or condensed without changing dry bulb temperature. This is shown by a vertical line on the chart.

**Cooling and Dehumidification** (Figure 12)

In standard air conditioning applications this is the most widely used process one comes across.

Let us assume that air is cooled from 35°C dry bulb and 24°C wet bulb temperature to 20°C dry bulb and 17.6°C wet bulb temperature. If we join these two points and draw a parallel line from the reference point to intersect the sensible heat factor line, we will notice that it intersects at 0.74 indicating that there is 26% latent heat removal and 74% sensible heat removal.

Let us now look at the major changes taking place in this process of cooling and dehumidifying.

1. The relative humidity of leaving air increases, indicating that as air is cooled its ability to hold water diminishes and the air passing over the cooling coil releases this extra moisture.

2. Dew point temperature of air decreases. This is due to decrease in specific humidity or weight of moisture per kilo of dry air, indicating latent heat removal.

3. Specific volume has decreased, again indicating that cold air is denser than warm air.

4. Enthalpy of air leaving is less than entering air, indicating removal of total heat.

In applications, where there are a large number of people, like theaters and restaurants, the line will be steeper, indicating a larger percentage of latent heat load in the total load.

In applications, where there is a heavy electrical load and low occupancy, there is more sensible load and less latent heat load. In such cases the line on the psychrometric chart will be flatter with less slope. Such applications include computer rooms, studios, control rooms etc.

If this cooling and dehumidification process is reversed then it is known as a heating and humidification process.

**Evaporative Cooling** (Figure 13)

This process of cooling the air is comparatively more effective in areas with drier climate. It is essentially the same as a wet bulb process. When air is passed through a series of water sprays, it looses its sensible heat and picks up latent heat. This means decreasing the dry bulb temperature and increasing the specific humidity. When no heat is added or removed from the recirculated water, it is called an adiabatic process and is represented along the wet bulb temperature line or constant enthalpy line. The efficiency of the adiabatic saturator determines how close the outlet air condition would approach the saturation line. Higher the efficiency, closer would be

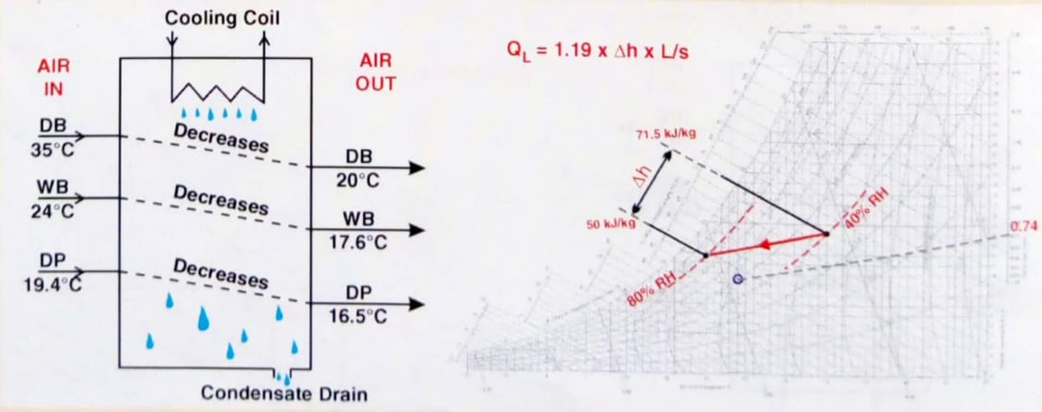


Figure 12 : Cooling and Dehumidification

the outlet air condition to the saturation line, and lower the dry bulb temperature.

The ASHRAE chart uses a protractor, outside the psychrometric chart on the left side, which has a scale of $\Delta h/\Delta w$ printed around its perimeter. This helps us to demonstrate that in evaporative cooling, the ratio of enthalpy change to humidity change is constant and is equal to the enthalpy of the water being sprayed into the air stream.

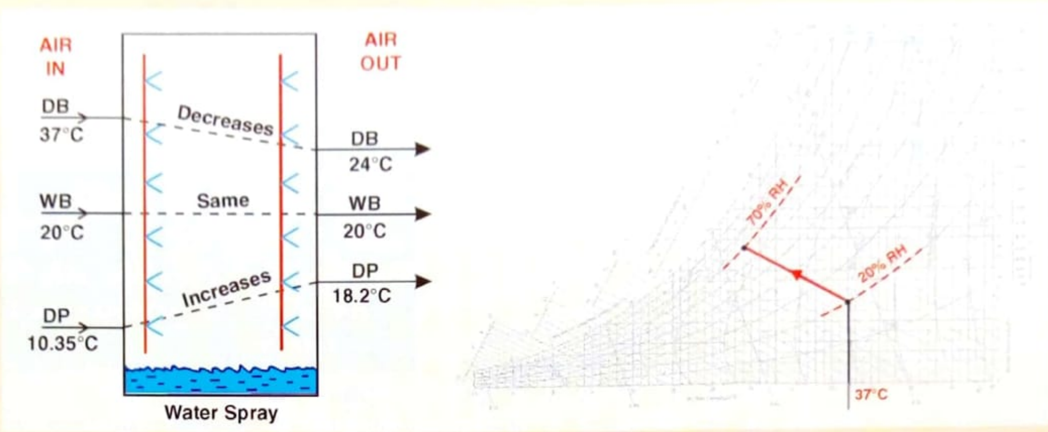


Figure 13 : Evaporative Cooling

The effect on air passing through such a saturator is as under:

1. Adds latent cooling load to space
2. Removes sensible cooling load from space
3. Used where high humidity is required like textile mills
4. Used as a low cost and simple alternative to air conditioners and is more effective in areas of dry climate, since the wet bulb depression is high and air can be cooled to a lower dry bulb temperature. It can be used for space cooling provided the final state is not too humid.

## References

Carrier Corporation Training Material.
Trane Air Conditioning Handbook.
ASHRAE Fundamentals Handbook.

**Next Issue :** Will cover some more applications using a psychrometric chart, such as mixing of air streams, apparatus dew point, bypass factor, RSHF, GSHF, chemical dehumidification, cooling tower process and will conclude our study on psychrometrics. ❖

# Basics of Air Conditioning

## Part 4

### Applied Psychrometrics

In the last article (Part-3), we covered understanding of 1. sensible heating/cooling 2. latent heating/cooling 3. evaporative cooling and 4. cooling and dehumidification processes using the psychrometric chart.

For practicing air conditioning engineers in India, a commonly used application is comfort air conditioning, mostly involving cooling and dehumidification process and we shall discuss in this article some more terminologies associated with the same using a psychrometric chart.

In any air conditioning application, air is used as a medium for picking up heat from the space to be cooled and larger the air quantity circulated, faster will be the heat removal (transfer) process.

The basic heat transfer equation, ($Q=m.Cp^{*}.\Delta t$) holds good also for air and it is therefore essential to calculate the mass of air required for a particular job under consideration, that would be adequate and sufficient to transfer heat efficiently from the space to be cooled, so as to maintain the designed space conditions. (*Sp. Ht.)

In the previous article we have seen how the psychrometric chart enables us to determine values of sensible heat and latent heat, as well as total heat or enthalpy. It also helps us to determine the sensible heat ratio.

In this article we shall discuss, how to make use of the psychrometric chart to arrive at the required air quantity and quality, which will be adequate to do the job.

### Outside Air Mixing

Barring applications involving total sensible cooling loads only, in most of the other applications of comfort air conditioning, outside air needs to be introduced in the space for ventilation and needs to be cooled by passing it over the dehumidifying equipment i.e. the cooling coil.

The air quantity required for ventilation depends on the number of people occupying the space, and the extent of air contamination on account of food, smoking or other processes.

**About the Author**

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Since the outside air dry bulb temperature and humidity is changing constantly, when outside air is mixed with return air, the mixture condition would also have varying conditions. The final condition of mixed air would depend upon the quantity as well as the temperature and moisture content present in the air at a given time.

Mixing outside air with return air (room air) can be plotted on a psychrometric chart by connecting the room conditions with outside conditions by a straight line. The air mixture conditions would then fall on this straight line and can be determined by knowing the quantities of outside air and return air. The mixture point would be nearer to the larger quantity in the proportion to the air quantities.

For example, if we mix 1500 L/s of outside air at 35°C db/28°C wb (point B) with 4500 L/s of recirculated air at 25°C and 50% R.H.(point A), the mixture point C would be closer to the point A because of the greater amount of recirculated air.

Since the outside air quantity is ¼ th the total quantity (6000 L/s), if we divide the straight line AB in 4 parts, the mixture point would end up at ¼ th distance from the recirculated air point. The mixture temperature at

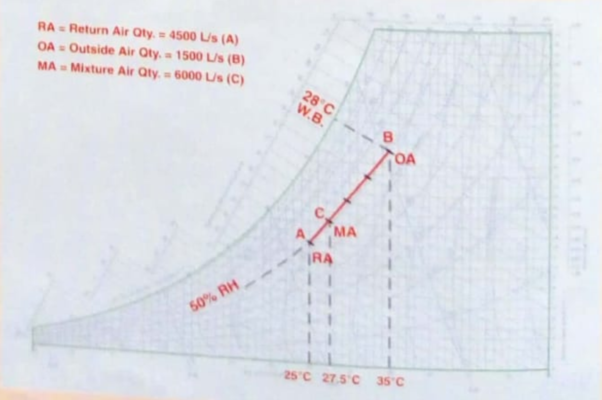


Figure 1 : Mixing of air

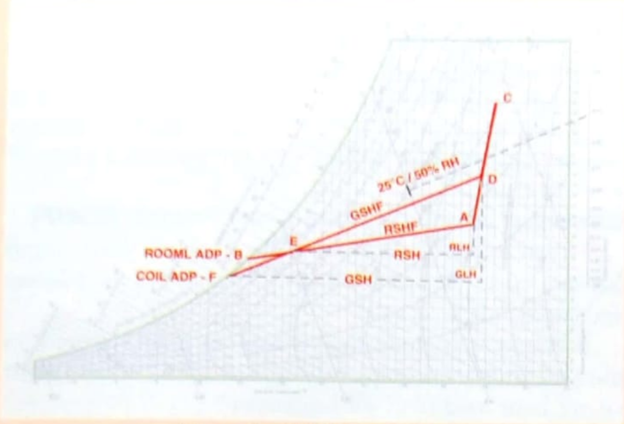


Figure 2 : RSHF & GSHF

C can then be read as 27.5°C and all other conditions of mixture air like relative humidity, wet bulb temperature and other properties can be found easily once the mixture point is located on the chart.

The mixture condition can also be calculated as under. Use of the psychrometric chart, however, gives fairly accurate values for all practical purposes.

$$T_{mix} = \frac{(1500 \times 35) + (4500 \times 25)}{6000} = 27.5°C$$

### Establishing Required Air Quantity (RSHF & GSHF)

We have now seen how to arrive at conditions of air entering the apparatus, which is a mixture of outside air and return air. It is now essential to determine the air quantity required to offset, not only the room load comprising of Room Sensible Heat(RSH) plus Room Latent Heat (RLH), but also it should be able to meet total load including load imposed on the apparatus by outdoor air.

We need to, therefore, differentiate between Room Sensible Heat Factor (RSHF) and Gross Sensible Heat Factor (GSHF). These factors can be calculated once we calculate room sensible heat (RSH) and room latent heat (RLH) load as also Gross sensible (GSH) and Gross latent heat (GLH) load.

$$RSHF = \frac{RSH}{RSH+RLH} = \frac{RSH}{RTH} \qquad GSHF = \frac{GSH}{GSH+GLH} = \frac{TSH}{GTH}$$

Once the factors are calculated we can then draw a line parallel to the line connecting standard condition point of 25°C, 50% R.H. and the sensible factor (RSHF) calculated as above, from room condition A to intersect saturation line at B and from mixture condition D to intersect the saturation line at F. The points at which these lines would intersect the saturation line are termed as room Apparatus Dew Point ADP (B) and coil ADP(F).

It is essential that the supply air condition should always lie on line AB representing RSHF, if it is to satisfy room design conditions and in Figure-2, the point E represents supply air condition.

Figure-2 does not show the effect of heat gain due to fan and duct heat gain and duct losses. These also have to be taken into account while establishing the cooling load and for determining supply air quantity. However to simplify understanding, we have not considered these in this article.

The air quantity ($L/s_{room}$) required to meet room load would then be calculated as :

$$L/s_{room} = \frac{RSH}{1.23(T_D - T_E)} \quad \text{or in FPS units as } cfm_{room} = \frac{RSH}{1.1(T_D - T_E)}$$

Similarly the air quantity required can also be calculated, knowing the latent heat or from the total heat for which we have already discussed the applicable formulas in previous articles.

In order to arrive at point E one has to go through a trial and error method, since the quantity of air needed to satisfy both room as well as total load has to be the same. If the air was to leave fully dehumidified, i.e. at point B, the matter would have been simpler, but the coil has its own inefficiency which is an unknown factor. Another unknown factor is the percentage of outside air to total air which does not get established, till the amount of air to be handled by the apparatus is arrived at.

Normally, the room load and RSHF ratio at full load under a given set of conditions will remain constant. However the GSHF ratio, either increases or decreases as the outdoor conditions and the quantities change.

It can be seen that, if the point E is nearer to point B (saturated condition), the quantity of air required to be circulated to meet the room cooling load decreases and as the point E moves away from B, on line BA, the

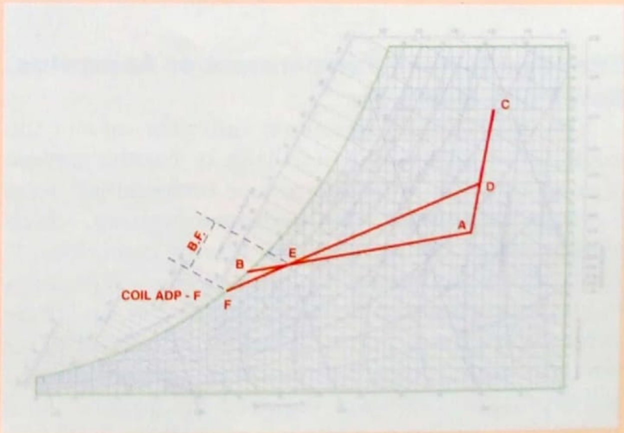


Figure 3 : Bypass Factor

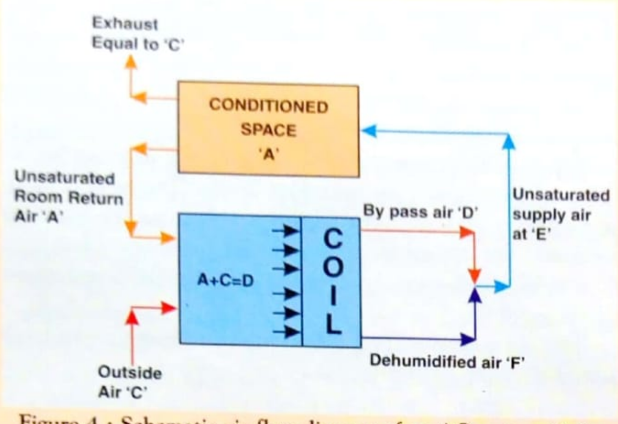


Figure 4 : Schematic air flow diagram of an AC system. Refer Figure 2 for various conditions

quantity of air required increases.

## Bypass Factor

Since the air is cooled by the cooling coil, the coil efficiency needs to be taken into account and it leads us to another terminology, called Bypass Factor (BF).

Air leaving the cooling coil will not always be saturated and will leave at a condition where some moisture will condense over the coil and thus both sensible and latent heat removal takes place. It is difficult to calculate mathematically the exact coil leaving conditions of air and hence for simplifying calculations and understanding the subject, it is assumed that a certain percentage of air while passing over the coil is not affected at all and leaves the coil at the same conditions at which it has entered the coil. This quantity of air to the total quantity circulated is termed as coil bypass factor (BF). It is also assumed that the remaining quantity of air (1–BF) reaches saturation point F and is therefore called dehumidified air and many times, (1–BF) factor is also referred as the "contact factor."

$$\text{BF (Coil Bypass Factor)} = \frac{(T_E - T_F)}{(T_D - T_F)} \qquad \text{Contact Factor} = \frac{(T_D - T_E)}{(T_D - T_F)}$$

## Effective Surface Temperature or Apparatus Dew Point (ADP)

The coil surface temperature varies throughout the surface as air comes in contact with it. For the purpose of simplicity, the "effective surface temperature" term has been established. This is the temperature, which would produce the identical air leaving conditions. It can be seen from Figure 5, that the temperature condition of air drops when it is flowing over the coil, and some common reference point, therefore, is needed. This reference point is the effective surface temperature and it helps us to calculate required air quantity and temperature reference while selecting the coil. This is the temperature at which the GSHF line crosses the saturation line on the psychrometric chart and thus is also known as Apparatus Dew Point or ADP.

Having understood the terms RSHF, GSHF, BF and ADP, we shall now look at the method used for determining air quantity without resorting to a trial and error method.

## Effective Sensible Heat Load Factor (ESHF)

Effective Sensible Heat takes into account, room sensible heat, fan motor heat gain, supply duct losses and supply air heat gain

It does not include the outside air load imposed on the coil, that gets completely dehumidified at dew point i.e. air heat load of (1-BF) quantity

Similarly Effective Latent Heat factor would be termed as(ELHF).

$$\text{ESHF} = \frac{\text{ERSH}}{\text{ERSH+ERLH}} = \frac{\text{ERSH}}{\text{ERTH}}$$

The Effective Sensible Heat Factor (ESHF) is the ratio of Effective Room Sensible Heat (ERSH) to Effective Room Sensible Heat plus Effective Room Latent Heat (ERLH) or Effective Room Total Heat (ERTH)

The effective sensible heat factor is (ESHF) represented on the psychrometric chart by connecting points F and A.

This is a simplified method of establishing required air quantity (L/s), and then it is not necessary to calculate RSHF or GSHF, since the air quantity using ESHF, BF and ADP results in the same air quantity.

$$\text{L/s} = \frac{\text{ERSH}}{1.23(\text{TA}-\text{TF})\text{x}(1-\text{BF})}$$

This calculated dehumidified air quantity, simultaneously offsets both room heat load as well as

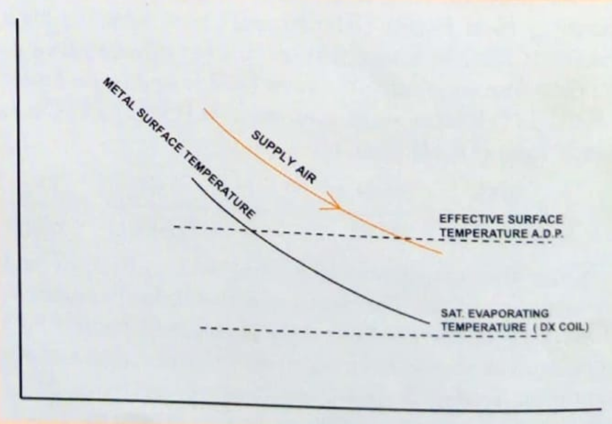


Figure 5 : Effective surface temperature or Apparatus Dew Point (ADP)

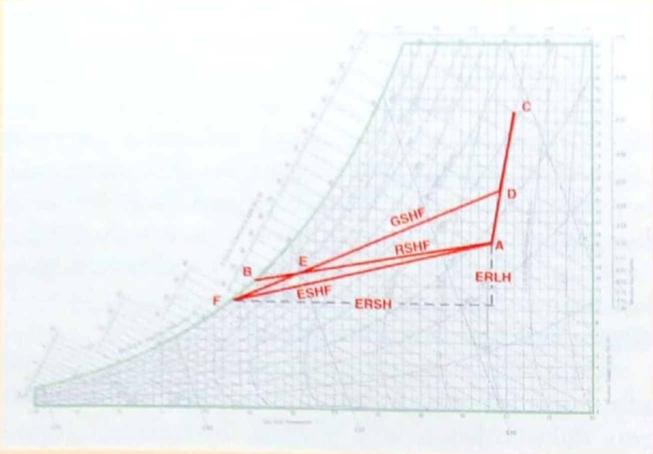


Figure 6 : Effective Sensible Heat Factor (ESHF)

total heat load, including outdoor air loads.

The quantity of air thus established is called *effective* since its coil leaving temperature and humidity level is *effectively* lower than the room conditions required to absorb losses in supply air path, as well as to take care of room load, and the factor is therefore called Effective Sensible Heat Factor (ESHF).

The quantity of air can also be calculated from ERLH or from total heat load ERTH, by using applicable formulas already discussed.

## Psychrometrics in Cooling Tower Operation

We have gone into considerable details of cooling and dehumidification application for determining the quantity of air required to offset the various loads and one more area where the psychrometric process plays an important role that remains to be looked at, is the cooling tower.

Our discussions on psychrometrics would not be complete unless we touch upon this apparatus as well.

If the question is asked, whether water can be cooled from 30°C to 21°C when the ambient temperature is 26°C with 50% R.H., without knowing the psychrometric principles, one has to think hard as to what would be the right answer. Now that we have learned the principles, the counter question that would immediately be raised is "what is the wet bulb temperature of ambient air?"

Besides the use of evaporative cooling for coolers in dry climates, evaporation as a means of cooling water is utilized to its fullest extend in cooling towers. When compression equipment is not working, cooling tower acts as an evaporative cooler. Since no heat is added to the condenser water loop, the condenser water temperature will tend to approach the entering wet bulb temperature with a deviation to some extent due to the condenser pump heat addition. The tower will cool and saturate the air flowing through it like a desert/ evaporative cooler.

## Heat and Mass Balance

We shall see how the heat exchange takes place in a cooling tower operation.

When the compression starts, heat is rejected to condenser water raising its temperatur, say from 21°C to 30°C. The cooling tower working on the evaporation of water principle, will cool the water to 21°C. This is possible even with dry bulb temperature higher, at 26°C.

Since the incoming air temperature(26°C), is higher than the desired cold water temperature (21°C) sensible cooling of water by air is not possible. We must know the wet bulb temperature of outside air. Assuming, wet bulb temperature to be 18.7°C corresponding to 50% R.H., it is possible now to cool water to 21°C by evaporation. We know very well that it is the large amount of latent heat required for evaporation, which plays a dominant role in transferring heat from water and thereby cooling it, compared to sensible heat transfer.

As the moisture is evaporating, air will also get humidified, increasing its absolute (specific) humidity from 10.54 g/kg to 24.229g/kg.

The enthalpy of air at 26°C db/18.7°C wb is 52.997 kJ/kg and assuming air leaves in an almost saturated condition at 28°C db/28°C wb, the enthalpy is 89.976 kJ/kg. Thus the heat added in the air stream is 89.976–52.997 = 36.979kJ/kg. The wet bulb temperature rise is approximately 9°C (28°C –18.7°C). We can thus see that an exactly equivalent temperature drop takes place in water from 30°C to 21°C.

The heat added for the evaporation of water can also be established with the help of the chart. The moisture content of air at inlet is 10.54 g/kg, whereas at the outlet it is 24.229 g/kg. The increase in moisture content of air is 24.229–10.54 =13.689 g/kg. The latent heat of evaporation at this temperature is 2434.85kJ/kg and multiplying these we get the total heat used in evaporating water as 33.3kJ/kg, which gets reflected in

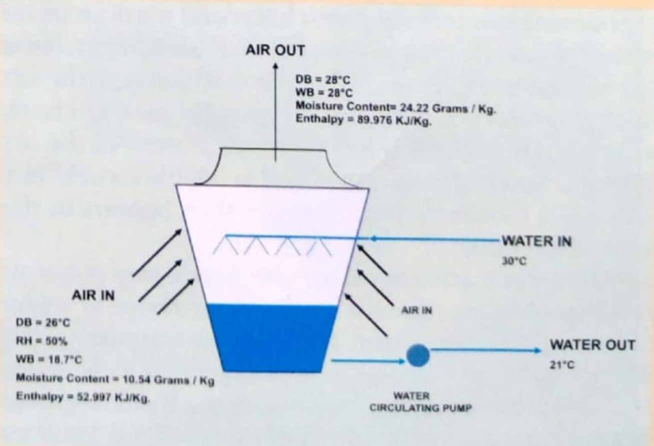


Figure 7 : Cooling tower – peak load

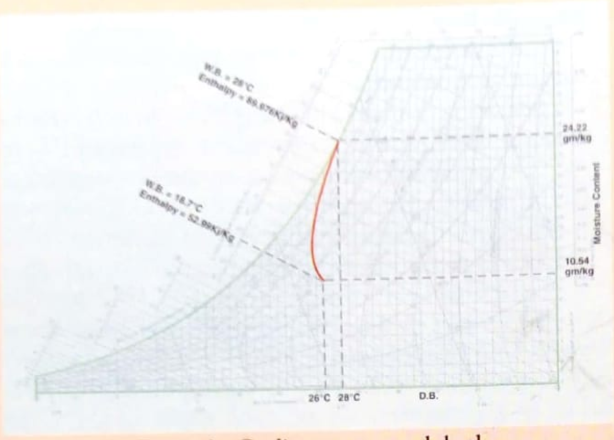


Figure 8 : Cooling tower – peak load

enthalpy increase of air as calculated earlier at 36.979kJ/kg. Out of 36.979 kJ/kg, 33.3 kJ/kg is latent heat due to moisture added in air from evaporation and the remaining 3.679 kJ/kg is reflected in dry bulb temperature increase from 26°C to 28°C.

While selecting the cooling tower, the temperature drop in water from 30°C to 21°C is called "range" whereas the difference between water outlet temperature of 21°C to wet bulb temperature of 18.7°C is termed as "approach."

The size of tower increases as the "approach" is reduced. We should therefore select a cooling tower for monsoon conditions when the wet bulb temperature is high and the load on cooling tower may be higher than in summer, when, although the dry bulb temperature is more, the air is drier and the wet bulb temperature is much lower providing a greater "approach". Cooling towers may therefore perform better in summer with drier climate than in monsoon with humid air.

## Eliminating Fog/Mist and Condensation

On many occasions we notice that there is fog or mist formation at the air outlet of cooling towers. When the outside ambient temperature is lower and warm air mixes with this stream, on some occasions the mixture joining line will be going through the region which is to the left of the saturation line on the psychrometric chart, indicating that there is free moisture present in the air, after the air is fully saturated and is unable to hold any additional moisture. This moisture then appears in the form of fog/mist.

On some installations, we also notice that if our air conditioning equipment is working, there is water dripping/ condensation forming on carpets, walls developing wet patches, curtains appearing to be soggy etc. The most common observations are window glasses getting fogged either from inside or outside, depending upon outside weather conditions or inside conditions in an air-conditioned space. The most irritating experience is while driving the car during monsoon, when visibility gets obscured due to misting of glass.

If the inside conditions in the air conditioned space has higher specific humidity than outside ambient air, one would notice window glasses getting fogged, as additional moisture beyond the dew point will condense on surfaces that are at lower temperature than dew point like inside wall, curtains, uninsulated ducts etc. It is now easy to address all these problems, having learned the psychrometrics.

## Chemical Dehumidification

When unsaturated air is passed through solid adsorbents like Silica Gel, or Activated Alumina, air gets dehumidified and heated. Sometimes liquid absorbents like Calcium Chloride are also used.

As the moisture is condensed from air, latent heat is released while condensation results in heating of air. In theory, it can be termed as an adiabatic process, if no external heat is added or removed and then such a process would take place along the wet bulb line AB. In actual practice, however, the final state of chemically dehumidified air would be more likely to fall at point C instead of B.

On many occasions chemical dehumidification is used in addition to an air conditioning plant, like gauge/tool rooms, as this combination becomes more economical than designing the air conditioning plant alone to maintain the required low humidity.

We have now covered most of the processes with the help of the psychrometric chart. There are, many more combinations, which we come across in actual day-to-day work that can be addressed using a psychrometric chart.

It is, however, amazing to note that very few practicing engineers/consultants make use of this most powerful and valuable tool.

Most of us feel that taking help of the psychrometric chart may be wastage of time and would lead to a more complicated solution. On the contrary, plotting the process

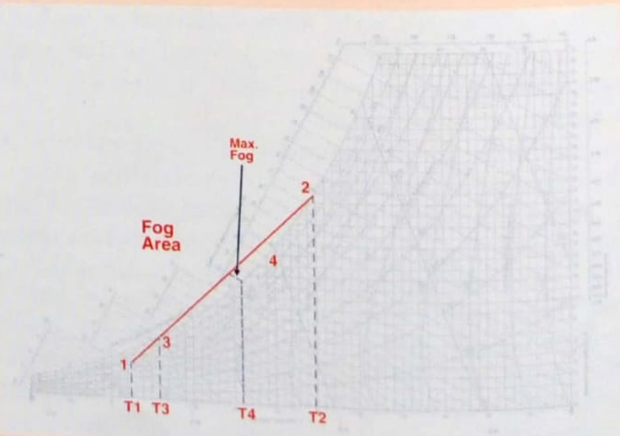


Figure 9 : Cooling tower fog formation



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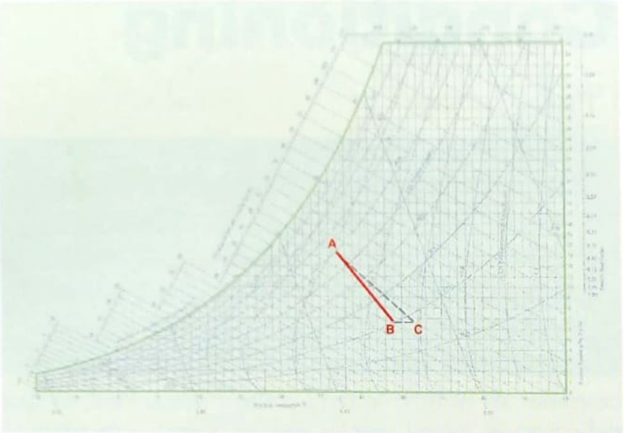


Figure 10 : Chemical Dehumidification

on the chart leads to a simpler and cost effective solution.

Many times we estimate cooling loads on the basis of general guide lines and select plant capacities, which look adequate for performing the required duty. The problem arises when the plant fails to maintain design conditions, as sharing of load pattern between sensible and latent heat is very different than normal and although total capacity of plant is adequate, the plant fails to satisfy the design requirements.

My personal experience is that the process designed with the help of a psychrometric chart, leads to better understanding and also gives confidence that the plant would definitely achieve the expected designed conditions.

ASHRAE USA has recently introduced and is marketing an excellent CD on applied psychrometrics, which allows the application engineer to know all properties of air with accuracy, and also plot the processes. This CD is available in the ISHRAE Mumbai office and can be studied, if any one is interested. I have made use of this software in my current presentations

### References

1. Carrier Training Manual
2. Paharpur Cooling Tower-Fundamentals
3. Tested Solutions to Design Problems by Melvin A. Ramsey

**Next Issue :** With this remark we conclude this article on psychrometrics and in the next issue we shall discuss factors influencing cooling load, and possible available options to reduce the same. ❖

# Basics of Air Conditioning

## Part 5

### Estimating Cooling Loads

While designing air conditioning systems, the main objective is to maintain design conditions in the specified space. If air conditioning calls for reducing temperature and humidity, we need cooling. In order to do so we have to pump out heat from this space, to a place where we are able to reject this heat.

In order to pump out heat we need external energy to operate an air conditioning system. In an air conditioning system, the air, which is circulating in the space to be cooled, picks up this heat and in turn gets heated. The warm air is then mixed with a certain quantity of fresh, outside air and the mixture is then cooled in an air handler housing a cooling coil, where it gets cooled, dried, filtered and is then supplied to the space for picking up heat again. The cycle thus continues.

In the last article, we have seen that estimating the correct quantity of air which will offset the heat load and maintain design space conditions is the primary objective while estimating the cooling load.

In order to arrive at this quantity of air, we need to first estimate the cooling load imposed on the space that needs to be neutralized.

There are many independent/dependant variables which influence cooling load and it is necessary to understand these, if cooling load is to be estimated accurately.

The purpose of this article of the *Classroom* on cooling load estimating is not aimed at indicating various factors and formulae with calculations. There are enough reference books available to provide all this information, and it would be a voluminous collection of different tables and factors, if one would like to include it in this article. The suggested reference books are *Carrier Air Conditioning Manual* or *ASHRAE Handbook*.

We shall try to cover the basic understanding of factors influencing and contributing to the cooling load so that an optimum and accurate cooling load is estimated. The designer should also be able to guide the customer about how to reduce the cooling load, so that reduction in initial cost as well as a saving in running cost is possible.

*This series of articles by Ramesh Paranjpey covers the basic principles of air conditioning. The articles will serve as a source of reference for newcomers joining our industry as well as for experienced engineers wanting to brush up on fundamentals*

It is a fact that an air conditioning plant/system operates at peak load very rarely since the peak load occurs less than 5% of the time. A peak load on the system is by exception and part load is the norm.

The designer must, however design and select the system so that it satisfies requirements and maintains design conditions, both during peak load hours as well as during partial load operation.

While estimating loads, the designer has to estimate both heating as well as cooling loads. For estimating heating loads, winter design temperatures are used. These occur mostly during early morning hours when there is very little human activity within the space. Also, the heat generating sources within the space help in offsetting heat capacity requirements.

As against this, the maximum cooling load occurs in the afternoon and all the energy producing sources, within the space, contribute to the cooling load and therefore need more careful consideration and effort in understanding and devising methods to reduce their impact.

We shall now look at the various sources that contribute to the cooling load. These can be grouped in three categories :

- External load
- Internal load
- Other loads

**External loads** – Originating from heat sources outside or external to the conditioned spaces. The main contributing sources are :

- Heat gain by conduction through walls, roofs, windows and through internal partitions, ceilings and floors.
- Solar heat gain through glass radiation and conduction.
- Outside air load through ventilation and infiltration.

**Internal loads** – These include:

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He has conducted training programs in ACR for several corporates and is a visiting faculty for the Government College of Engineering for graduate and post-graduate courses. He can be contacted at *pramesh@vsnl.com* or on 020-5436142

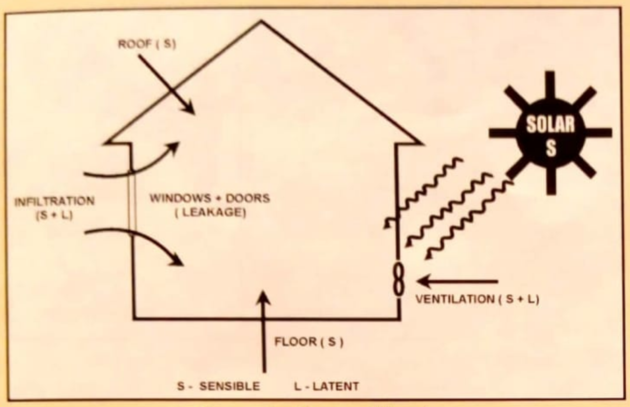

Figure 1 : External Loads

- People
- Lighting
- Appliances and equipment

**Other loads** – These are system generated heat sources and may comprise of:

- Supply/return air duct leakage and heat gain
- Heat gain from the fan motor in the air handler or the supply/return air booster fan motor.

There is another way to look at the distribution of the heat contributing sources. The cooling load components can be divided into:

- **Sensible heat load components.** These components would always tend to cause an increase in the dry bulb temperature of the space. They are from walls, windows, roofs, lights, solar heat gain as well as from people, appliances, equipment and ventilation/infiltration air.
- **Latent heat load components.** Latent heat results when moisture is entering the space and causes humidity to increase. The factors contributing to latent heat load are people, appliances as well as infiltration/ventilation air.

As can be seen from Fig. 1 & 2, a load component may be totally sensible, or latent or a combination of the two, such as people, appliances, and air which contributes to both sensible and latent heat load.

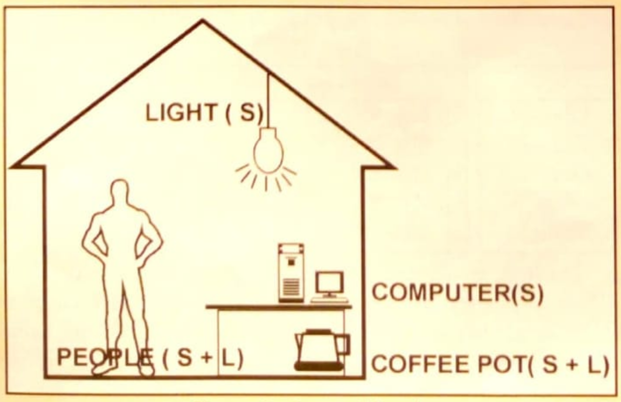

Figure 2 : Internal Loads

The air conditioning plant design should also ensure that proper humidity levels are maintained inside the space, besides temperature. Higher humidity decreases the body's ability to loose heat due to evaporation and therefore has a major effect on human comfort. The cooling plant must, therefore remove moisture at the rate it is being added to the conditioned space.

There are many factors besides the ones stated earlier which influence the cooling load. These are mostly beyond the control of a system designer. He can however advise the builder/architect before the project commences, so that some design aspects can be built–in to reduce the heat load. These could be :

1. Latitude, altitude, azimuth angles and location/ orientation of the building. The angle at which solar rays impinge, plays an important role in a heat load due to solar radiation and transmission.

2. The design ambient conditions during Summer/ Winter/Monsoon

3. The type of construction of the building – heavier the construction, less is the heat transmission. It also shifts the timing favorably i.e. when heat gets released to the space from the structure. The lighter construction (such as wood frame or insulated metal walls) will react more quickly to temperature changes than heavy structures made of concrete or bricks.

As the material of construction changes and the thickness increases, it takes longer for a temperature front to pass through massive walls, and therefore the timing of the peak cooling load for the space shifts. The inertia of a building and its ability to retain heat plays an important role.

4. The colour and the texture of the outside surface of the building has a strong influence in determining how much solar heat is absorbed or reflected from the surface. The lighter colours absorb less heat whereas darker shades absorb a higher percentage of solar energy, leading to a higher outside surface temperature of the building.

5. The location of insulation, if provided, also affects the cooling load. In a standard construction of a building, concrete or brick walls are insulated from inside to protect insulation. If exterior insulation with vapour barrier is provided, it reduces both sensible and latent load, especially if the temperature gradient between outside temperature and inside temperature is very high. Many times a sandwich wall, with insulation in between or ventilated space is provided.

These factors not only change the cooling load but also affect the thermal load profiles.

We shall now briefly cover various load contributing factors, their influence, and suggested methods to reduce their impact.

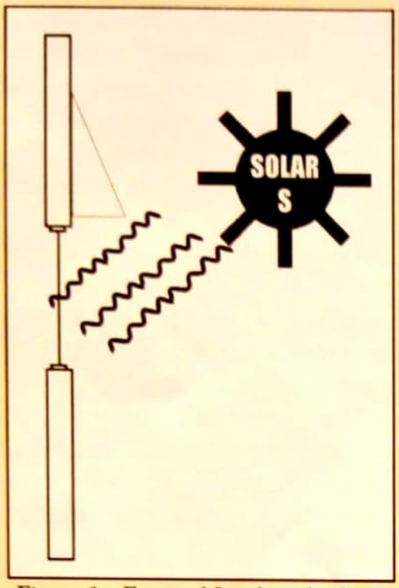


Figure 3 : External Loads, solar heat gain through glass

## External Loads

**Heat gain through glass areas.** The greatest external source of heat load is due to the sun.

The solar heat through glass is absorbed instantaneously in the room. The radiant solar wave energy that passes through glass strikes the objects within the room and heats the surface where it strikes. More the glass area, more would be the solar heat load due to radiation heat. For example, in passenger cars, the glass area is very large, comprising of windshield, rear view and four windows. The solar load, therefore, is the largest constituent of a passenger car heat load, nearly 55% of the total load.

- One of the ways to reduce the heat load due to solar radiation is to use heat absorbing glass. This can reduce solar heat gain by 25%
- Multiple panes with non-convecting dead air spaces decreases heat transfer. Double or triple panes are commercially available. Using only double pane glass can reduce the load by 10 to 20%. We see application of such an arrangement in air conditioned rail coaches, or air conditioned luxury buses. In critical applications even an inert gas can be filled in between the space to enhance the insulation effect.
- Another method is to use stained glass. Depending on the colour shade, and the density or using chemical coating can reduce the heat transfer to the extent of 30-65%. The use of glazing films covering the glass area is also a popular method used to reduce solar energy impact.

| | SOLAR HEAT REDUCTION |
|---|---|
| HEAT ABSORBING GLASS | 25% |
| DOUBLE PANE | 10-20% |
| STAINED GLASS | 30-65% |
| U-FACTOR | % REDUCTION IN HEAT LOSS |
| WINDOW 6.42 | — |
| STORM WINDOW 2.56 | 60% |
| DOUBLE WINDOW 3.69 | 43% |

Figure 4

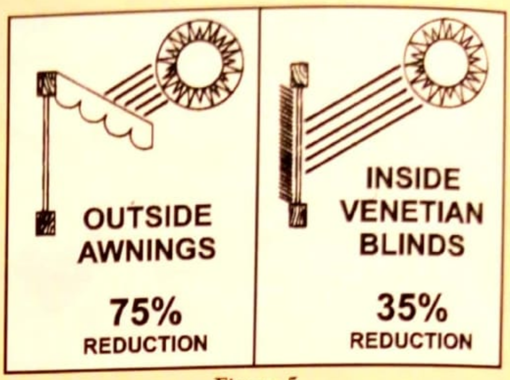


Figure 5

- The glass fixing frames normally made of metal can serve as a heat transfer bridge. Adding a thermal break within the frame wall decreases the rate of heat transfer or alternate materials such as wood, or vinyl can also reduce heat ingress.

As can be seen from Fig. 4, the heat transfer coefficients reduce significantly if a double window or such additional device is used, leading to substantial reduction in heat gain.

- As complete outside shading, which is the most effective way to reduce solar heat is not practicable, an outside awning will turn away nearly 75% of the solar heat, whereas an inside roller shade or venetian blind will give about 35% reduction in solar heat gain through glass.

The cooling load calculation format therefore has separate identity for this solar heat load through glass. Radiation heat entering through all the glass areas facing East, West, North, South as well as skylight glass is included in this part. The various factors to be taken for calculation are available in reference books. Table 1 shows this format.

Table 1

| Item | Area | Sun Gain or Temp. Diff °C | Factor | Watts |
|---|---|---|---|---|
| | | Solar Gain – Glass | | |
| Glass-N | Sq. Mtr. × | × | | |
| Glass-E | Sq. Mtr. × | × | | |
| Glass-S | Sq. Mtr. × | × | | |
| Glass-W | Sq. Mtr. × | × | | |
| Skylight | Sq. Mtr. × | × | | |

**Heat gain through walls and roof due to solar effect.** The second part of the cooling load estimate form deals with the heat transmitted by solar rays through all the walls and shaded/exposed roofs. The effect is in two ways:

- Sun's rays striking the walls and therefore resulting

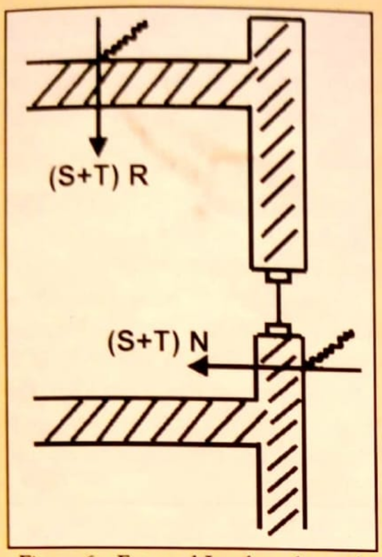


Figure 6 : External Loads, solar and transmission

in heat gain and temperature rise.

- Since outside temperature is higher than the inside temperature, it results in transmission heat gain through the walls.

Many times the outside temperature of walls and roof is higher than the ambient temperature, and during this period all the heat does not get transmitted inside and some of the heat is reflected back in the air.

Also, since the walls have mass, the storage effect makes the heat flow time related. It is very complex to determine the actual amount of heat transmitted. The Equivalent Temperature Difference (ETD) concept was therefore developed and reference books or tables give this value, which takes into consideration the time related storage capability of the walls and roofs.

Table 2

| Item | Net Area | ETD | Factor "U" | Watts |
|---|---|---|---|---|
| Solar & Trans. Gain - Wall & Roofs | | | | |
| Wall-N | Sq. Mtr. × | × | | |
| Wall-E | Sq. Mtr. × | × | | |
| Wall-S | Sq. Mtr. × | × | | |
| Wall-W | Sq. Mtr. × | × | | |
| Roof-Sun | Sq. Mtr. × | × | | |
| Roof-Shaded | Sq. Mtr. × | × | | |

The graph (Figure 7) shows how walls react to solar heat. The effect of solar impact depends on which way the wall is facing and the timing when sun rays start hitting the walls.

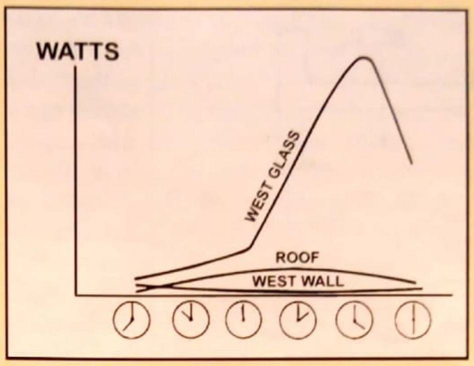


Figure 8

For the sake of better understanding , the graph shows commercial building construction with heavier concrete or brick structure.

The East wall heats up first and the peak load is around 2.00 PM. This takes into consideration the time lag between maximum solar heat impact on the East wall at around 10.00AM and the time this heat gets dissipated into the space.

The South side wall will have highest load around noon and therefore contributes the maximum load around 4.00 PM.

West side is just the reverse of East and peaks between 4.00 to 6.00 PM and the heat gain is therefore maximum at around 8.00 PM

North side gets some solar heat all the day, but as can be seen from the graph its intensity is the least.

The timing will shift depending upon the wall construction. Heavier the wall, it would take longer to heat up and longer to cool off. Thus it would admit more heat during later part of the day. It could be possible that the West side wall would be letting in more heat, may be, between 8 to 10 PM in the evening.

The graph (Figure 8) would show that the net effect of solar heat through walls, is very small compared to roof or glass and therefore if we pay more attention to reducing impact of heat gain through glass or roof, major reductions are possible.

**Transmission heat gain due to temperature difference-except walls and roof.** (Figure 9) The third part of the cooling load format includes heat load due to transmission on account of the temperature difference between external temperature and inside temperature condition. This part includes all

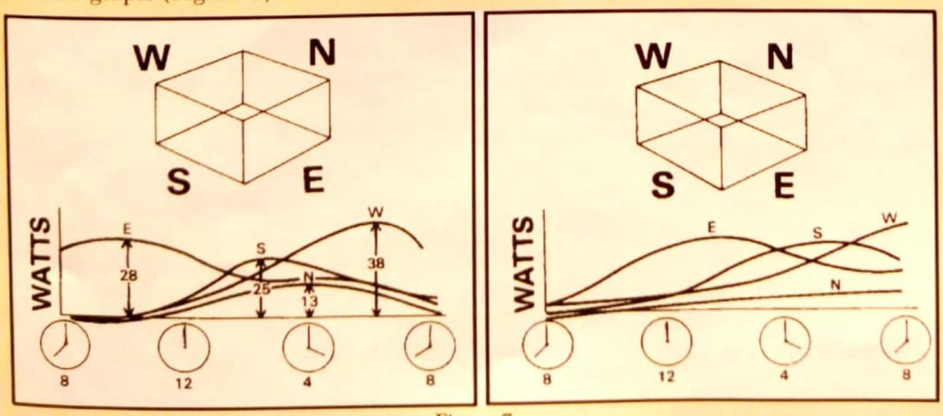


Figure 7

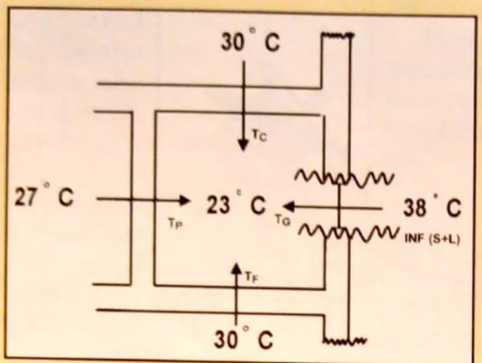


Figure 9 : External Load, transmission – except walls & roofs

glass areas added together plus partition walls, ceilings, floors as well as the sensible heat component of infiltration air load.

This calculation is based on a simple equation , Q= U×A×ΔT

Transmission gain through glass areas can be considered as a steady state load, since glass has very little heat storage capacity. The other portions are internal barriers, like floor, ceiling and partition walls. These can be considered as steady state since internal temperatures remain fairly constant. If the adjacent areas are air conditioned, then there is no temperature difference and the load would be zero. The cooling load format is as under, in Table 3.

Table 3

| Item | Net Area | ETD | Factor "U" | Watt |
|---|---|---|---|---|
| Trans. Gain - Except Walls & Roofs | | | | |
| All Glass | Sq. Mtr. × | | × | |
| Partition | Sq. Mtr. × | | × | |
| Ceiling | Sq. Mtr. × | | × | |
| Floor | Sq. Mtr. × | | × | |
| Infiltration | L/s × | ΔT | × 1.19 | |

## External Load

**Infiltration:** In addition to the transmission load, we also consider load due to leakage of untreated air getting inside the space through leakage across window seals, doors, porous exterior walls, floors, roofs etc. This infiltration load is dependent on pressure differential and is in turn caused by wind velocity, difference in air density, or due to pressurization caused by supply and exhaust systems.

A reference book gives tables for estimating this load.

**Ventilation:** In addition to infiltration that is by default, we require fresh air for the human occupants. When proper ventilation requirements of 7 to 15 L/s per person are considered, this air needs to be cooled. For the purpose of calculations, we have seen in the last article that the quantity of air, considered as bypassed air, contributes to the effective sensible room heat load, as this is untreated air.

For determination of EFFECTIVE ROOM SENSIBLE HEAT the losses encountered along the way from the cooling coil to the room have also to be considered, since the quantity of air must be adequate to overcome room sensible heat, plus load due to bypassed air and supply air duct losses.

## Internal Loads

The internal loads are mainly due to people and equipment, appliances, lighting.

**People.** The human body also generates heat and releases it to space. The amount of heat generated depends upon the activity level of the person. Human beings contribute both sensible as well as latent heat.

Table 4 gives some idea regarding the heat load due to occupancy. More detailed explanations and calculations may be obtained from reference books.

Table 4

| | Heat From People | | |
|---|---|---|---|
| | Sensible Watts | Latent Watts | Total Watts |
| Theatre | 55 | 45 | 100 |
| Office | 56 | 74 | 130 |
| Dancing | 70 | 180 | 250 |

**Lights/Equipment/Applinces.** Light bulbs or any other form of artificial illumination would generate heat and has to be considered in cooling load calculations.

Within the conditioned space, equipment like computers/ xerox machines, calculators or any electrical driven equipment would generate heat. Similarly, appliances like washing machines, cooking ranges, electric irons would produce heat. Some appliances would contribute to both sensible as well as latent heat (Table 5).

## Latent Heat Loads

The latent part of infiltration air, internal loads, bypassed out-door air load and supply duct leakage losses together form latent heat load, and formats for calculations are included in cooling load estimation forms.

## Room Load

All these loads discussed so far contribute to space load. The sensible heat component and latent heat component when added becomes the total room load.

## Effective Loads

When we add both sensible and latent heat load component of supply air losses to room load it is called "effective room load". We have dealt in detail with this aspect in our last article. This load establishes the quantity of air required to be cooled and dehumidified by the cooling coil, and should effectively be able to overcome room load as well as supply air duct losses, as the air is conveyed from cooling apparatus to the room.

## Other Loads

After the supply air has absorbed the room load, the

Table 5

| | | |
|---|---|---|
| LIGHTS INCANDESCENT = 3414 WATTS | | |
| FLOURESCENT = 3414 WATTS PLUS BALLAST | | |
| MOTORS: APPROX. 1100 - 1200 WATTS HEAT PER 1000 W SHAFT OUTPUT | | |
| COFFEE MAKER | WITHOUT HOOD | WITH HOOD |
| SENSIBLE W | 260 | 130 |
| LATENT W | 60 | 30 |

air is returned to the cooling apparatus for further processing. During this passage the returning air picks up heat as the duct passes through a space whose temperature is higher than the air. Similarly, the duct losses need also to be considered. Compared to supply air heat gain, return air losses are less. If booster fans are provided these also add to heat load.

The major portion of load however is due to ventilation air. While calculating room load we considered only the bypassed air, which is considered as not being affected while passing over the coil . The bypassed portion of the air is about 5 % depending upon the coil design, but the remaining 95% air needs to be cooled to the apparatus dew point, which is called "dehumidified air ".

### Grand Total Load

This is the sum of room loads, supply air losses, return air losses and outdoor air load. Grand Total Load is also known as "dehumidified load".

### Refrigeration System Load

The refrigeration unit needs to be sized to take care of "grand total heat load" plus chilled water piping losses, and circulation pump heat gains as the pumps are doing work on the water, in case of chilled water systems.

### Cooling Load Calculation Methods

There are many methods available for determining cooling loads. Many user friendly software packages are also available which makes estimating process simpler as it prompts and guides the user to fill up the required data, so that chances of making any error or omitting any relevant data is minimized. The standard formats developed by Carrier/ASHRAE are also useful and are being used by most practicing engineers.

As has been seen, to calculate effect and timing of each load component and the combined effect of the same is a very complex process and hence terminology like Equivalent Temperature Difference (ETD), which is not the actual temperature difference, but the effective temperature difference values, taking into account the combined effect of most of the variables as accurately as possible, have been arrived at.

ASHRAE introduced in 1967, an innovative method to account for the effects of these two variables, like sun load and building thermal heat capacitance, by introducing a system of Total Equivalent Temperature Differential values and system of time averaging (TETD/TA) to calculate cooling loads. This two-step method first calculates the heat gains from all sources to get an instantaneous heat gain. This is then converted to a space cooling load through use of weightage factors, which take into account influence of the building's thermal storage. This is a tedious method for manual calculations and some computer programs use this.

The second method known as Transfer Function Method (TFM) was introduced in 1972. This computer based procedure also occurs in two steps, of first calculating space heat gain and then space cooling load. This method helps in evaluating both the rate at which heat is removed from the space and the temperature of the space when a specific size of cooling unit is used.

The third method is a one step process that uses the Cooling Load Temperature Difference(CLTD) or a Cooling Load Factors(CLF), or combination of both, for each component of cooling load. This method was first presented in the 1977 ASHRAE volume.

Efforts are still continuing to seek refinements of computational methods, which would promise even more accuracy and flexibility. For the purpose of this article it is enough if the basic concepts are understood.

### Summary

We have considered major heat load areas. It is impossible to cover this subject in such an article completely, however the intention was to make the reader aware of various cooling load contributing sources and how to calculate each portion with the help of reference books/computer programs. Some suggested methods also have been indicated as to how to reduce the impact of these loads, if one gets associated from the start of the project.

### References

1. Carrier technical development program -System Design

2. Fundamentals of Heating & Cooling Loads- ASHRAE Continuing Education

**Next Issue:** Design of ducting or air distribution ❖

classroom

# Basics of Air Conditioning

## Part 6

### Air Distribution

In our previous articles, we have indicated that in comfort air conditioning applications, it is the air that picks up the heat from space to be cooled. This air then travels to the fan coil unit, where it is cooled and filtered. This cooled and dehumidified air is then supplied to the room again. It is thus seen that a closed loop is formed for the circulation of air.

The room is normally at atmospheric pressure. As the air enters the return air passage/ duct, through the return air openings/grills from the room, its pressure starts dropping until it reaches the fan. The fan raises the pressure. Thereafter the pressure starts dropping again in the supply duct until the air is released to the space. Therefore the pressure on the suction side is negative and on the discharge side is positive.

Figure 1 illustrates the general arrangement of an air distribution system.

In addition to return air, infiltration and ventilation air also enters the space. In order to maintain the balance,

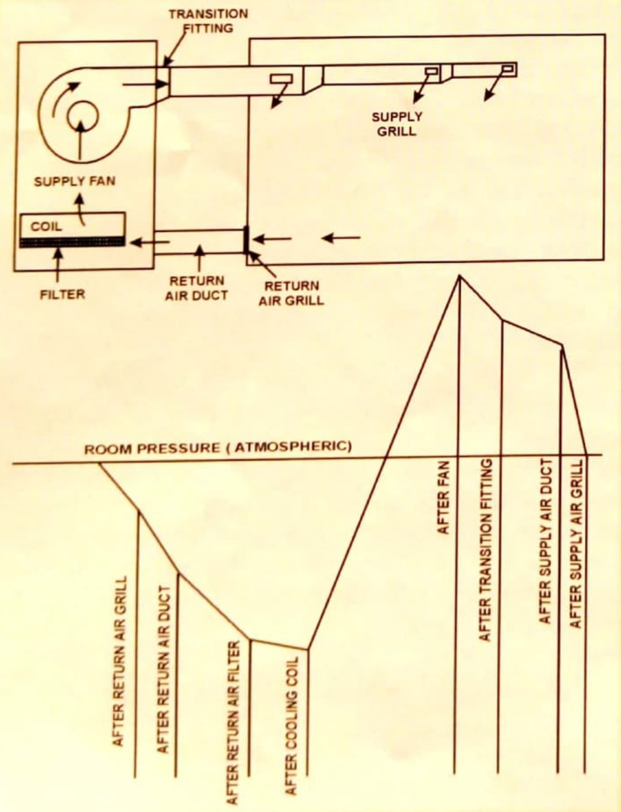


Figure 1: Air distribution system components

*This series of articles by Ramesh Paranjpey covers the basic principles of air conditioning. The articles will serve as a source of reference for newcomers joining our industry as well as for experienced engineers wanting to brush up on fundamentals*

it is essential that an equal amount of air is exhausted from the space. An analogy with a water tank will make this clearer. When the water tank is full, we cannot add any additional water. Similar is the case with air.

Proper distribution of air in the space, assumes utmost importance, since we are using air as a medium to pick up the heat from the space to be air conditioned.

It has been experienced by most practicing engineers that a well designed refrigeration system, with poor air distribution cannot satisfy design conditions, whereas an excellent air distribution can compensate to a very large extent, minor deficiencies in plant capacities.

The primary objective in designing the air distribution system can therefore be stated as – proper combination of air temperature/humidity and motion to assure comfort to occupants.

This objective can be achieved by avoiding :

- Excessive room temperature variations, maximum temperature variation within the room should not exceed 1°C
- Excessive air velocities
- Stagnant air pockets leading to suffocation
- Inadequate air quantities causing hot or cold spots.

In addition to these precautions, the designer has also to consider :

- Sufficient ventilation air to avoid odours
- Adequate and proper filtration of air to ensure clean air supply to the room
- Proper duct sizing and openings to avoid unwanted noise


### About the Author

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- Proper duct and joint sealing to avoid air leakage, so that designed quantities are delivered at the right places.

Considering these points and based on extensive surveys conducted over large strata of population, the general conclusions drawn are :

- A recommended velocity in an occupied zone should be below 50 fpm(0.254 m/s), with an ideal bench mark figure as 25 fpm (0.127 m/s). A velocity between 25 fpm and 50 fpm is acceptable to more than 85% of the population.
- Below 25 fpm velocity, people complaint of stagnant air.
- These are the recommended velocities around the neck to head region of the subjects, as the neck area of human being is twice as sensitive compared to feet levels.
- For industrial uses, higher velocities can be used even from 75 fpm to 350 fpm depending upon the application.

Figure -2 illustrates that the most desirable direction of cooled supply air is from the front side towards the face of a person. Cold air directly on feet or from the back on the neck is undesirable.

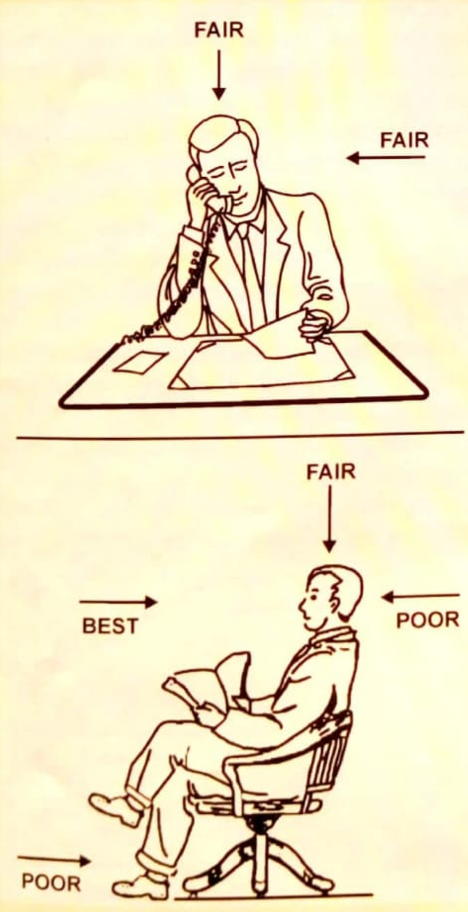


Figure 2

The table gives guidelines for recommended air outlet velocities for different applications :

| Application | Recommended Terminal Velocity(fpm) |
|---|---|
| General Offices | 1,000 – 1,250 |
| Dept. stores-Upper floor | 1,500 |
| Dept. Stores-Lower floor | 2,000 |
| Motion Picture Theatre | 1,000 |
| Hotel bedrooms | 500 – 750 |
| Apartments & Residences | 500 – 750 |
| Broadcasting Studio | 300 – 500 |

I have deliberately indicated velocities in fpm, since the values are sufficiently large and majority of us still are able to understand the significance better with these values in fpm, instead values in m/s. The conversion factor is-

Velocity in fpm x 0.00508 = Velocity in m/s

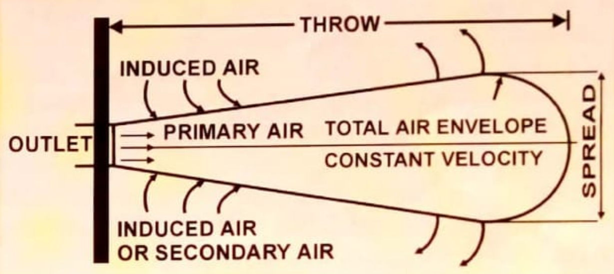


Figure 3

We shall now try to learn the behaviour of the air as it enters the space from duct outlets.

As the air comes out from the outlet, it mixes with room air and carries it in its stream. The air coming out from the duct outlet is termed as **primary air** and the room air that mixes with it is called **secondary air** or **induced air**.

Since this air coming out of duct outlet enters the space or free zone, it forms an expanding cone.

As the cone spreads, the air velocity drops down . A distance to which air travels till its velocity drops to about 50 fpm to 75 fpm is called **Throw or Blow** and is measured at 6'(1.8 m) above the floor level.

**Spread** – Spread is the angle of divergence of an air stream after it leaves the outlet. The spread is both in horizontal and vertical direction.

The throw is directly proportional to the air outlet velocity. The amount of secondary air getting inducted also depends upon air velocity. Higher the velocity, longer is the throw/blow and higher is the induction.

The temperature of air is lowest at the outlet and as the air travels, while the velocity is reducing, its temperature is gradually increasing since the secondary room air is mixing with the primary stream. During this process the air is absorbing the heat from the space to be cooled.

At the extreme end which means at the end of the

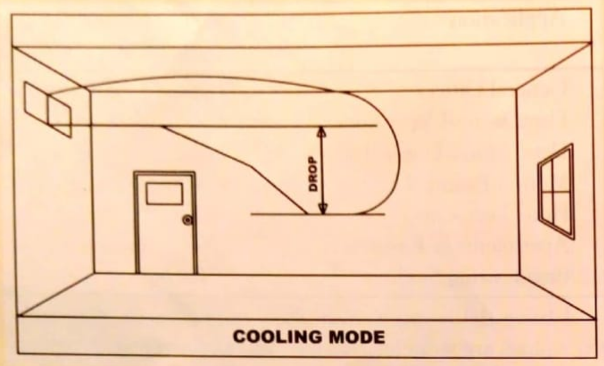


Figure 4

throw, the temperature of air is more or less same as room temperature.

In the in-between region, where the velocity of air is 150 fpm or higher, the density difference between primary cold air and relatively warm secondary air does not have much impact on the shape and direction of the envelop and the flow of air in the room is normally along the axis.

Beyond this point, the temperature difference, which means density difference, affects the direction and the shape of the envelope as well as throw/blow.

If supply air temperature and room temperature difference is high, the air envelope has drooping tendency, since cold air would tend to rush to floor level due to higher density. (Figure 4)

Normally up to 6 ft height above the floor is considered as occupied zone. If the air outlet is approximately 2 ft above this zone, it normally ensures that the air velocity in the occupied zone is within the acceptable range of 15 fpm to 50 fpm, and the temperature is also uniformly maintained. (Figure 5)

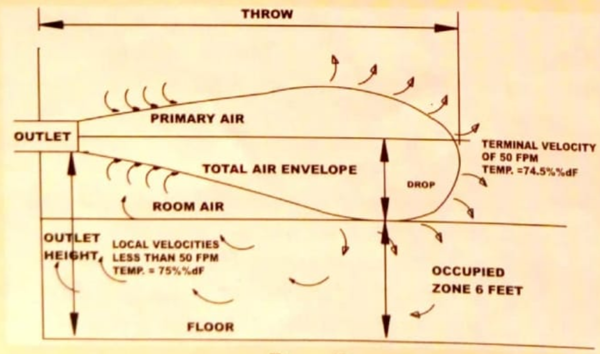


Figure 5

## Effect of Directional Vanes

It is easy now to understand that if the air outlet vanes are straight, the throw will be longest and the spread will be narrow.

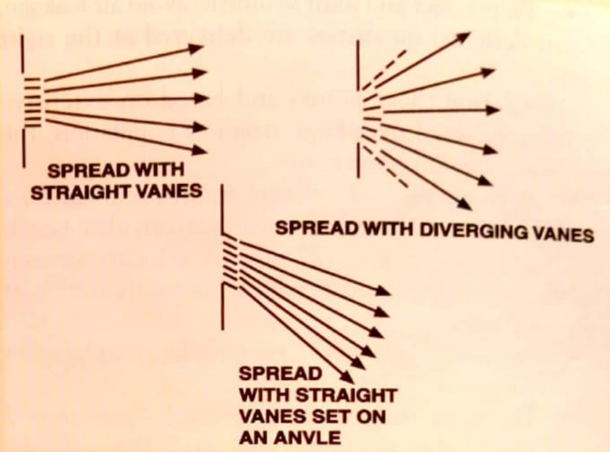


Figure 6

If the vanes are diverging, throw will be short but the spread area will be large, and if the vanes are facing in one particular direction, the air stream will follow the general direction in which the vanes are facing.

## Effect on Air Stream due to Ceiling & Walls

Until now we discussed air stream behaviour in free blow applications. It is necessary to remember that air is a fluid and therefore invariably follows properties of fluids. It means that the air has a tendency to cling to the surfaces and travel along. The jet of air directed towards the ceiling will hug the surface and travel along the ceiling. (Figure 7)

Contrary to normal belief this results in longer throw compared to free delivery projection. It is therefore always a good practice to direct the air stream by angling the louvers towards the ceiling or side wall surfaces to get longer air travel.

This also helps in pushing the air envelop near the surface keeping it out of occupied zone and thus, improving overall air distribution.

Figure 8 illustrates that the acceptable throw would be L+H.

We should select fan characteristic based on required L+H. It would then ensure that the air envelope does

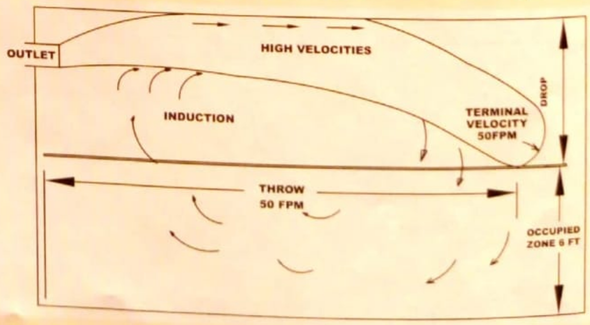


Figure 7: Effect of walls and ceilings

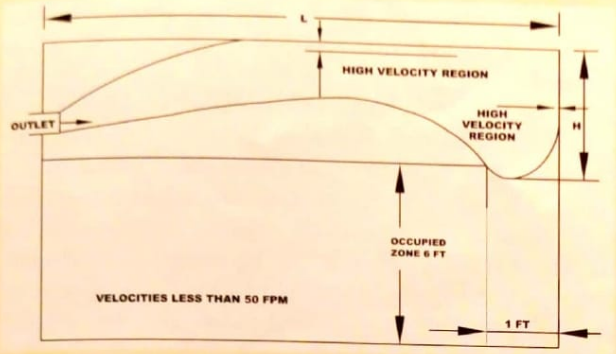


Figure 8 : Throw and drop for confined spaces

not drop in the occupied zone, as the air jet velocities will be high enough to keep the air up along the ceiling.

The distance 'H' selected should be such that it is above occupied zone, in other words for a 10 ft high and 8 ft long room, the designed throw would be 8 ft+ (10-6)=12 ft approximately.

While selecting the fans, for the radial air outlets same principle applies when they are installed in the false ceiling.

Having discussed air flow management principles, we shall now look at various type of supply air outlets normally used in air conditioning installations.

These could be :

- Grills-various shapes and sizes
- Ceiling diffusers
- Slot/linear/laminar diffusers
- Perforated ceiling panels

The considerations for selecting the particular type, will depend upon individual design, aesthetics and choice of the customer. However, the points to be considered while selecting the outlets from designers point are identical irrespective of the type used. These are :

1. The selection and location of the outlet should be such that the amount of air to be delivered by the outlet should be proportional to the load of the part of the space.

2. The locations should be governed by the condition of uniform air distribution and rapid temperature equalization.

3. The grills /diffusers should be provided with vertical/horizontal adjusting vanes–behind the visible portion, to ensure uniform air flow distribution through all openings of the grills/diffusers.

4. Due to velocity in the duct, the air, when diverted at right angles towards the outlets will lead to uneven distribution and it is therefore essential to straighten the path of air before it reaches the outlet. This will ensure uniform air distribution through all the openings of the grills/diffusers. This is normally achieved with the help of air path straightening vanes.

5. Whenever the duct velocity exceeds the outlet discharge velocity, a collar joint with vanes as shown in Figure 10 is preferred instead of fixing the grill/diffuser directly on the duct.

## Return Air Outlets

As mentioned earlier, it is essential to remove equal amount of air as supply air from the space, if air circulation is to be maintained. If this is not done, the air circulation would come to halt due to pressure build up in the space.

It is normally a good practice to locate return air outlets, in the areas most likely to have stagnant zones where air tends to accumulate.

The design of return air path is comparatively simpler than the supply air distribution

From the view of noise consideration, velocity in return air should normally not exceed 75% of supply air velocity at the air outlet.

In many installations, we observe that the return air passage is completely blocked affecting air conditioning adversely.

Keeping the return air filters clean, and providing unobstructed passage for return air is therefore very

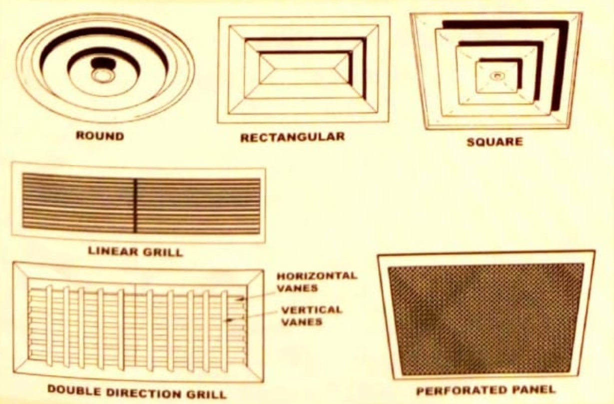


Figure 9

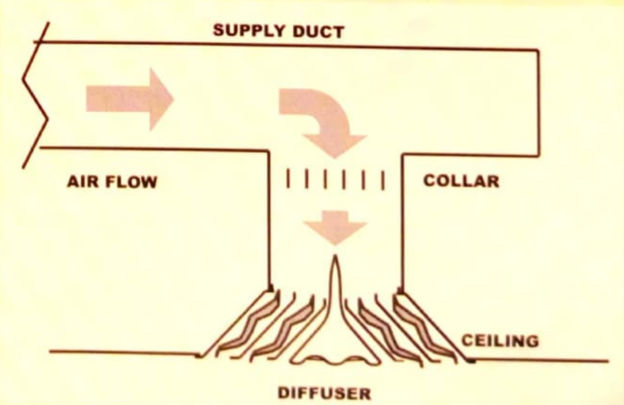


Figure 10

essential for good air conditioning system performance.

It can again be compared to a water tank normally provided in a chilled water system. If the return supply line is blocked or closed, hot water from the system will not return to the tank and would gradually empty out. Thus, the tank will not be able to supply enough quantity to the system. Similar is the case with return air passage, which needs to be kept without obstructions for getting good air conditioning system performance.

I have experienced in many installations, especially with bus air conditioning systems, where the return air opening is completely blocked by luggage piled up in front or a nice picture frame is fixed on the grill, as visible portion of grill or an opening provided for return air does not look good. The customers then complain about no cooling, which is obvious since there is no return air. It is my personal experience and recommendation that whenever a customer complain about unsatisfactory performance of an air conditioning system , it is always better to look at the return air passage first , and in many instances the problem would get resolved simply by ensuring that the return air flows back to the air handling unit smoothly without any restriction.

### Location for Return Air Grills

In many applications a hall, corridor or public space can be used as a return path with the use of large grill at the end of passageway.

Another popular way of getting back the return air in many buildings is to provide false ceiling and take the air from the cavity above the ceiling.

The precautions to be taken while locating return air grill is to ensure that there is no short circuiting between supply and return air

Also the return air grill or opening is not too near the machinery room. If it is too near, the noise from the machinery would get carried inside the room.

Floor return should be avoided whereever possible because it catches all dirt and imposes severe strain on both filters and cooling coils.

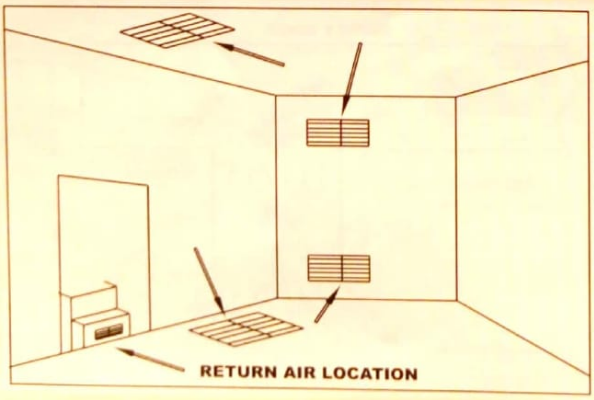


Figure 11

### Sound

So far we have considered recommended supply and return air velocities. We also now need to take into consideration, how to reduce unwanted sound in the space while selecting the air distribution system components. The sound power is the characteristic of the blowers and motors. It can be compared to electric power or the wattage of the light bulbs. Once we select a particular equipment it's sound power gets defined and cannot be altered since it is a source characteristic. It is measured as NC levels.

Noise – Any unwanted sound can be called as noise. Human ear is sensitive to sound pressure. The noise as sensed by human ear is measured in terms of sound pressure in decibels. An analogy between sound and light can illustrate this more clearly. The illumination effect can be adjusted or diffused by shading, or by providing different types of angles for light rays to impact or even by changing colour shades or reflecting surfaces of the walls. Similarly, with sound pressure, levels can be made acceptable by designing proper air distribution scheme, location of air supply and return openings, duct attenuators, lining of ducts etc.

The acceptable noise levels as defined in ASHRAE are :

| Type | NC decibel levels |
|---|---|
| Residences | 20 – 30 |
| Apartment buildings/Hotel rooms | 30 – 40 |
| Hotel public spaces | 35 – 45 |
| Conference rooms | 25 – 35 |
| General offices | 35 – 50 |
| Offices – Private | 35 – 45 |
| Offices – Executives | 30 – 40 |

As mentioned earlier, the sound power depends upon proper selection of air moving equipment and fans/ motors play an important roll besides duct sizing, air outlet selections etc.

In order to understand why a particular type of fan is selected for a particular application, let us look at some of the properties of various fan/blower types.

The fans can be either propeller type or centrifugal type.

Propeller fans are used where no push to the air is required i.e. for free delivery applications, without ducting. These type of fans move maximum air quantities for a given size compared to centrifugal blowers, and hence we select these for air-cooled condensers. Similarly, these are also used for exhaust air or for induced draft cooling tower or for car radiators.

The centrifugal fans can either be forward curved design or backward curved type.

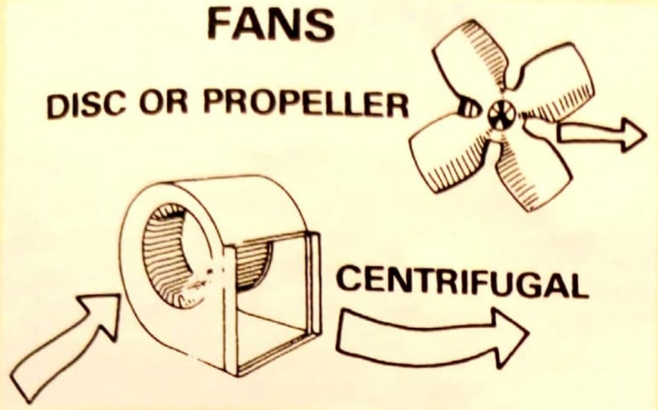


Figure 12

For identical diameter, forward curved fan delivers more air while running at the same speed compared to a backward curved fan.

In other words, for delivering the same air quantity, the forward curve fan would run at a slower speed compared to a backward curved fan.

This is an important consideration from noise point. We all know that the noise levels are dependent on tip speed of the wheel, that determines the air outlet velocity. The tip speed is dependent on blower rpm and diameter of the wheel. Higher the tip speed, higher is the noise level.

The forward curved blower, rotates at a lower speed compared to a backward curved blower for delivering the same air quantity, or in other words for the same speed, it would be smaller in diameter, and its tip speed therefore will be lower, leading to more silent operation.

It is therefore customary to use fans having forward curved designs on all unitary products where the air conditioners or fan coil units are located near the occupants.

The backward curved blowers are normally used for large installations where air handling units are remotely located. The ducting lengths are long. These fans have

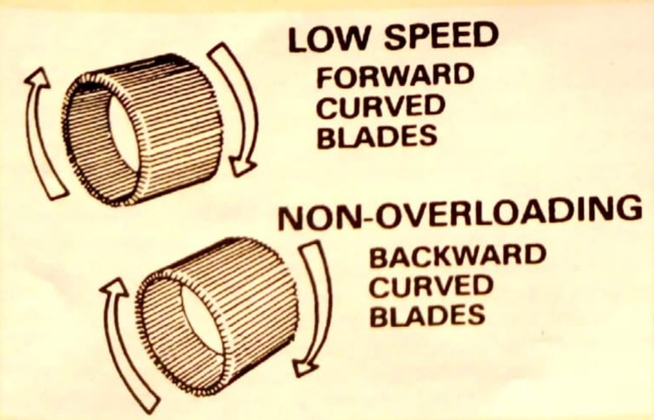


Figure 13

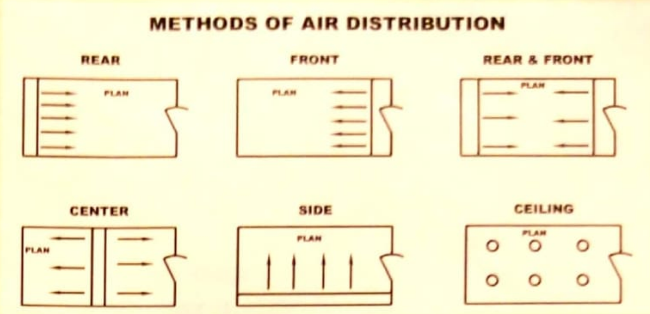


Figure 14

the advantage of limit load characteristic and therefore chances of motor getting overloaded due to incorrectly estimated static pressure is remote.

We shall now conclude this article by comparing various alternative locations for installing the air distribution units for a given layout .

**Grills at the rear.** If the throw is not properly calculated, there is possibility of hot spots at the other end, near the door.

**Grills at front.** Possibility of inducting outside air through door infiltration.

**Providing both, rear and front outlet.** More expensive, less chances of hot spots. Drafts can be avoided by ensuring streams do not over blow or meet in the center.

**Grills in the center header.** Better arrangement than the first three alternatives, medium cost.

**Grills along side.** Medium cost, excellent results. Design to ensure over blow or air bouncing from other wall.

**Ceiling diffusers.** More expensive but gives excellent results compared to all other alternatives.

In the foregoing text, we have touched upon various aspects which needs to be considered while designing air distribution system. We have not gone into more details of noise/sound characteristics. We have also not gone into more details of fan selections, fan laws etc. These could become independent articles. One can collect information on these from various text books or material specially dedicated to them.

For the purpose of this article, the intention of touching on noise level and fans was to bring out their importance.

## References

1. Carrier Handbook
2. Carrier Training Material
3. Refrigeration & Air conditioning – Text Book by C.P.Arora

**Next Issue :** Duct-designing methods normally used by design engineers and professionals.

# Basics of Air Conditioning

## Part 7

### Duct Design

In the previous article, we discussed the importance of proper air distribution for effective and satisfactory performance of the air conditioning system; we covered information on location and types of supply and return air outlets, recommended velocities as well as some information on fan selection and noise level criteria.

In this article we will cover methods of designing ducting, since ductwork is the component that transmits air from the air handling unit to individual supply air grilles/diffusers located in the space to be conditioned.

The air distribution system as a whole can be compared with the blood circulation system in a human body. As the heart supplies power to push the blood around the human body and bring it back, the fan does the same for air in the air distribution system. The arteries carry the purified blood to different parts of the body; similarly the supply air duct carries cold, dehumidified and filtered air to the required locations. The veins return the impure blood back to the heart; similarly the return air ducting or return air passages collect the warm and contaminated air and bring it back to the cooling coil/fan for reprocessing.

The importance of both the systems i.e. blood circulation in a human body and air distribution in a space normally goes unnoticed so long as these systems are performing to their expectations. Any malfunction in any part of either of the systems, affects the overall performance.

As in any building construction, the electrical wiring which is concealed, and is never noticed till some fault develops, similarly the air distribution system installed for an air conditioned area is not the focus of attention till the occupants start complaining about uneven/ insufficient air and then start criticizing the designer for inadequacy in his design.

### Objective

The prime objective is to design a duct system that is economical in both operating and first cost and that would transmit enough air at the proper temperature and velocity to satisfy the space air conditioning requirements.

The duct system designer, in addition to transmitting air into the space has to consider several other factors that are equally important. Some of these factors, are listed below:

*This series of articles by Ramesh Paranjpey covers the basic principles of air conditioning. The articles will serve as a source of reference for newcomers joining our industry as well as for experienced engineers wanting to brush up on fundamentals*

- **Available space.** In an office building or hotel, space is at a premium whereas in a factory installation space normally is of secondary importance. Therefore, the duct system for each of these areas would necessarily be completely different.
- **Sound level.** An objectionable sound level in a space can often become the first objection and most serious in an air conditioning system. A radio or TV studio has more stringent sound specifications than a cafeteria design. Therefore, a designer must be able to recognize the requirements and then design the duct system within the acceptable limits of the job.
- **Friction loss.** Friction in a duct system or the resistance to air flow must be kept within practical limits. Friction loss means revenue loss to owners.
- **First cost.** Obviously, anyone who designs a system has control over the first cost.
- **Heat and leakage losses and gains.** Heat gains or losses in a duct system can be appreciable. If a system is designed to deliver air to a room at 12°C but due to duct heat gain the temperature is 16°C or higher, the required room temperatures will not be accomplished, if this aspect has not been considered during designing.
- **Appearance.** The appearance may not be a major consideration in a factory building, but would certainly demand attention in places like conference halls, departmental stores, operation theatres etc. Proper concealment of ducts and selection of grills/diffusers matching the decor is a designer's responsibility.
- **Maintenance.** Doors for internal cleaning of ducts should be planned with an easy approach.

**About the Author**

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He is a visiting faculty for the Government College of Engineering for graduate and post-graduate courses, and a member of ASHRAE and ISHRAE. He can be contacted at *pramesh@vsnl.com* or on 020-5436142

Only after realizing that all these factors are going to influence our design and proper care taken to address these issues can one approach a project for duct design.

The first cost is determined by the size of the duct, bigger the duct size, more is the cost. The operating cost is determined by the fan power needed to push the air and depends on the system resistance or the pressure drop in the air distribution circuit. The designer must strike a balance between the two.

Duct distribution systems are normally classified as :

1. **Low velocity systems.** These systems are designed for air velocities upto 12.5 m/s (750 mpm) and static pressure upto a maximum of 12.5 mm of water gauge.

2. **High velocity systems.** These systems are designed for air velocities up to 23 m/s (1,400 mpm) and static pressure ranging from 25 to 75 mm.

High velocity and high pressure ducts are used for long distance conveying of air in certain applications and in order to meet space limitations one has to resort to high pressure systems. A typical example is in ship air conditioning and ventilation in which ductwork is more expensive in construction and uses spiral conduits or rectangular ductwork with flanged and gasketed or welded joints.

In this article, we shall deal only with low velocity duct design systems since the majority of our air conditioning applications fall in this category.

## Air Distribution

As the fan pushes air into the supply air duct the following changes take place :

- Air velocity increases
- Static pressure increases
- Temperature of air goes up slightly
- Air gets slightly compressed

These four changes are due to work done on the air by the fan. Since the pressures involved in air conditioning duct designs are very small, we do not take into consideration, effects of air compression. The temperature increase, due to fan motor work done is taken care of, while estimating the cooling load. Hence, we shall only concentrate on the remaining two aspects i.e. velocity changes and static pressure changes, while discussing the duct designing methods.

## Static pressure, Velocity pressure and Total pressure

Normally, total pressure and static pressure is measured as gauge pressure in inches of water or Pascal or mm of water, whereas velocity is measured in m/s.

We have learned in earlier parts that static pressure is exerted uniformly on all sides of the duct whereas velocity pressure is in the direction of airflow.

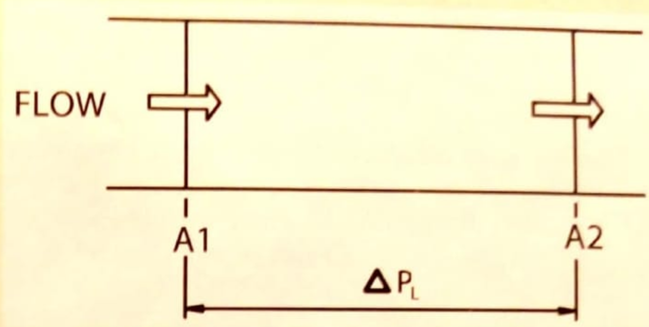



**Figure 1 : Flow with drop in total pressure**

If we assume frictionless air flow then the total pressure at A1 and A2 would be the same.

$$Pt = PS_1 + PV_1 = PS_2 + PV_2$$

In actual practice, pressure drop takes place as the air travels along the duct due to friction or other causes such as sudden changes in the direction or changes in the cross sectional area.

As a result of pressure drop, some of the pressure energy would be converted into heat energy and would appear in the form of insignificantly small temperature increase of the air. If we also take this pressure drop into account, then the equation would look like this:

$$PS_1 + PV_1 = PS_2 + PV_2 + \Delta P_L$$

$\Delta P_L$ represents the total pressure drop between the two sections. This pressure drop comprises of both duct friction loss as well as dynamic loss due to change in direction or velocity. For conveying the same air quantity, if the cross sectional area of the duct does not change, the mean velocity will also remain constant. The change in velocity will be only on account of cross sectional area changes.

As a result of pressure drop, the air actually expands a little, but the changes in volume are negligible. Hence, for all practical purposes, it is assumed that the density of air remains constant.

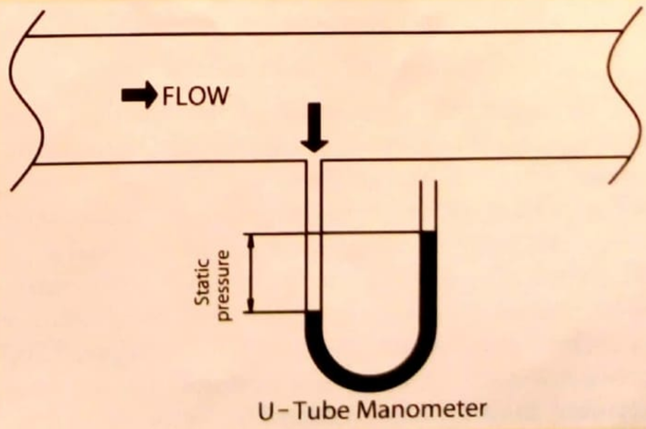



**Figure 2 : Static pressure**

In order to measure static pressure in the duct, the most common method used is to make a small pressure tap hole in any side of the duct and connect a 'U' tube filled with water.

Static pressure is expressed in inches/mm of water or Pascal (Pa).

Conversion factors are : 1" of water = 249 Pa
1 mm of water = 9.8 Pa
1 psi = 6.895 Pa

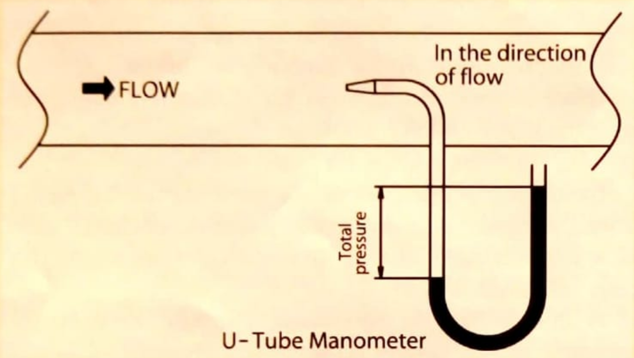



Figure 3 : Total Pressure

If another tube is inserted in the duct, in the opposite direction to the airflow, so that the air traveling along the duct is hitting into the tube opening, this would then indicate total pressure i.e. velocity pressure plus static pressure.

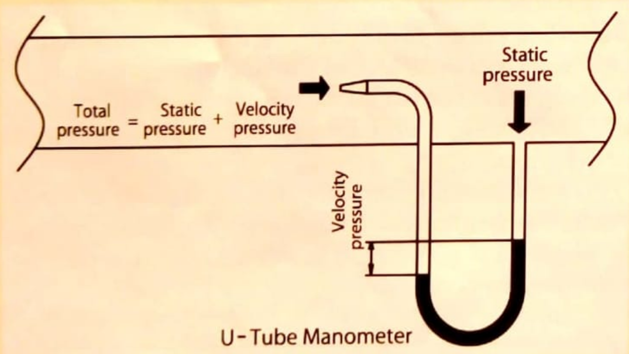



Figure 4 : Velocity Pressure

If we connect the open end back into the duct then the tube will indicate only velocity pressure as the static pressure effect will get nullified.

The velocity of the air in the duct is generally taken as a mean average velocity over the cross sectional area. This is calculated by knowing the total air quantity flowing through a particular section and dividing it by the cross sectional area of the duct. If for example, 5,000 liters per second air is flowing in a duct having cross sectional area of $0.5m^2$, then the velocity will be 5,000 litres/s= $5m^3/s$. $5m^3/s \div 0.5m^2 = 10m/s$

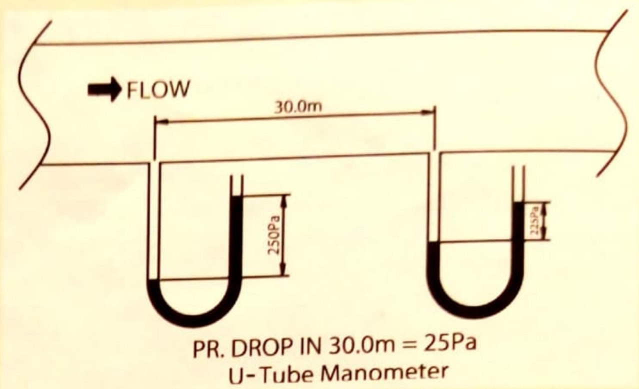



Figure 5 : Friction through ducts

As the air travels through a duct, the static pressure drops due to friction when air comes in contact with the duct surface. Figure 5 shows that there is a drop of 25 Pa over a length of 30 m of straight length of the duct run.

In actual practice a designer rarely comes across an application where only straight length of ducting is encountered. There are many enlargements, reducers, elbows, bends, tees etc and finally the air outlet grills/ diffusers. Each of these causes a pressure drop. The equipment also has an internal pressure drop through the cooling coil, filter and inlet/outlet openings as has been discussed in the previous article.

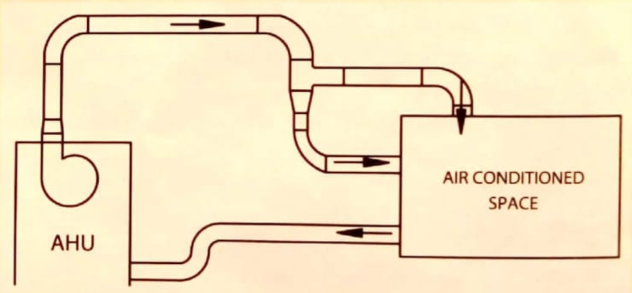
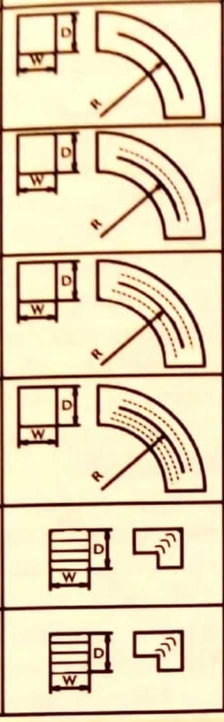



Figure 6 : Duct layout

A duct system designer should obtain from the equipment manufacturer the data on available static pressure at the unit outlet, which should be used by him to design ducting. This available static pressure would act as an available force to push the air to the farthest end of the longest branch of the duct.

Based on this data, a duct designer then has to estimate total equivalent straight length of the ducting. For this purpose, he has to take reference of manufacturer's catalogue for fittings that indicates equivalent straight length for various sizes and shapes of bends, fittings, grills, and diffusers.

The equivalent straight length means the particular item under consideration like bend or elbow, which will have same pressure drop or same friction, if a straight




| DESCRIPTION | DESCRIPTION | W/D | L/D RATIO |
|---|---|---|---|
| RADIUS ELBOW (NO VANES) R=1.25D | | 0.5<br>1.0<br>3.0<br>6.0 | 5<br>7<br>8<br>12 |
| RADIUS ELBOW (1 VANE) R=0.75D | | 0.5<br>1.0<br>3.0<br>6.0 | 8<br>10<br>14<br>18 |
| RADIUS ELBOW (2 VANES) R=0.75D | | 0.5<br>1.0<br>3.0<br>6.0 | 7<br>8<br>10<br>12 |
| RADIUS ELBOW (3 VANES) R=0.75D | | 0.5<br>1.0<br>3.0<br>6.0 | 7<br>7<br>8<br>10 |
| RECTANGULAR SQUARE ELBOW SINGLE THKNESS TURNING VANES | | 0.5<br>1.0<br>3.0<br>6.0 | 8<br>10<br>12<br>13 |
| RECTANGULAR SQUARE ELBOW DOUBLE THKNESS TURNING VANES | | 0.5<br>1.0<br>3.0<br>6.0 | 6<br>8<br>9<br>10 |

Figure 7 : Additional equivalent lengths of rectangular duct elbows

length of duct piece is used. Figure 7 has been included as a sample to illustrate this aspect. It may be noted that this is not the complete table and for various items, tables given in the textbook or manufacturers catalogues are available.

To illustrate the point, if we take a rectangular radius elbow with two vanes having R= 0.75 and W/D ratio 3, the L/D ratio indicated in the table is 10. If the designed round duct diameter of a straight length is 250 mm then the equivalent straight length of the elbow would be $L=10\times250/1000=2.5$ m

Similarly, equivalent lengths can be calculated for all such fittings and added to the length of straight portion to arrive at total equivalent straight length of the duct run.

In order to start designing/selecting duct size, we need to first calculate the main outlet size to be provided from the AHU, before going into designing branch sizes and air outlet sizes. This size of main duct is calculated by dividing the quantity of air delivered by the AHU divided by the allowable velocity. The velocity selection is based on the application involved, noise level limitations, available area for duct location and available static pressure at the unit outlet.

Table 1 illustrates normally recommended velocities in meters per minute for various applications.

There are three methods of duct sizing and these are:

1. Equal Friction
2. Velocity Reduction
3. Static Regain

## Equal friction method

Almost all designers use the equal friction method and it is the most popular method today because of its simplicity. This method is used to size supply, return and exhaust duct systems. The equal friction method is based on using the same friction rate or loss per meter length for the entire system. This method is far superior to velocity reduction method since it requires less balancing for symmetrical duct layouts.

The size of the first duct portion as usual is selected by selecting allowable maximum permissible velocity and the depth of the duct on the basis of height limitations. Then the friction loss per meter is calculated from the chart. The remaining sections, therefore, are designed using the same friction rate.

To determine the total friction loss in the duct system, it is necessary to calculate the loss in the duct run that has the highest resistance.

With duct systems, designed on an equal friction method basis, the calculated friction loss from one end to the other is not as great as calculated, since as the air proceeds down the duct the air velocity is reduced. This velocity represents energy. As energy cannot be created or destroyed, the same total energy must still be

Table 1 : Recommended duct velocities for different applications

| Application | Controlling factor: Noise | Controlling factor: Friction | | | |
|---|---|---|---|---|---|
| | Main Ducts-mpm | Main ducts-mpm | | Branch Ducts-mpm | |
| | | Supply | Return | Supply | Return |
| Residences, Hotel bed rooms, Apartments | 180 | 300 | 240 | 180 | 180 |
| Hospital bed rooms | 300 | 450 | 390 | 360 | 180 |
| Offices, libraries | 360 | 600 | 450 | 480 | 360 |
| Theatres | 240 | 390 | 330 | 300 | 240 |
| Restaurants | 450 | 600 | 450 | 480 | 360 |
| Cafeterias | 540 | 600 | 450 | 480 | 360 |

Ref: Text book by C.P. Arora

available. Hence, every time the velocity is reduced there is a conversion of velocity pressure to static pressure. This pressure will partially offset the friction in the next duct section.

A velocity pattern exists in a duct. High velocity air flows in the center of the air stream, while lower velocity air flows at the perimeter of the duct. At a take off the lower velocity air is peeled off. The higher velocity air continues down the duct.

Static pressure is potential energy. If air progresses down a duct with increasing velocity pressure, there will be a decrease in static pressure. Likewise, decreasing the velocity going through a duct increases the static pressure. The static regain, however, cannot be expected to be 100 per cent. Losses due to turbulence, construction of the duct, etc. will reduce this regain to some extent.

The static pressure difference between the first outlet and the last outlet, means the grills in the beginning have to be closed down considerably as compared to those in the last part. The balancing operation takes this difference into account. When compared to the velocity reduction method, the equal friction method results in a more economical selection of duct sizes.

If an equal friction design has a mixture of short and long runs of duct, the shortest duct will need a considerable amount of damper closing. This is a drawback of the equal friction method. This drawback can be overcome by resizing the shorter branches once again, based on available static pressure at the starting point of the branch.

This method is generally recommended and used since use of simple charts or "ductulator", speeds up the design work. Computer programs are also available and are used.

## Velocity reduction method

In this method, a starting velocity is selected at the fan discharge in a similar manner as the equal friction method. Arbitary reductions in velocity are made in the succeeding duct runs. Each duct section is sized using the formula V=Q/A.

The drawback of incorrectly designed ducting based on this method is, the duct sizes remain more or less of same size thus leading to higher costs, and secondly the friction rate variation is too big. Balancing of a system designed in this manner can be very difficult because the outlets must function with such a wide variation. Many times, the first outlet needs to be throttled considerably if one wants to get enough air at the far end.

Though the advantage of this method is that it allows safe velocities, it is normally not used unless the user has considerable practical experience and knowledge to design and select velocities with reasonable accuracy.

## Static regain method

In designing duct work by the static regain method, the objective is to establish the same static pressure behind all outlets so that they will distribute the same quantity of air with the same blow, in which case, no balancing at the outlets is required.

If the velocity at the first outlet reduces from $V_1$ to $V_2$, the static pressure will increase. The increased static pressure is now available to offset duct friction loss from the first to second point. The static regain method requires finding out the correct value of velocity $V_2$ which will give a regain in static pressure exactly equal to the friction loss in the equivalent length of the duct. This can be done by a trial/error method by assuming various velocities till such a velocity is found which gives static regain equal to the friction loss.

Once the main duct size is selected, then based on the method of duct design selected by the designer further sizes of branches and outlets are calculated. This method meets the essential requirement of maintaining uniform static pressure at all branches and outlets. Thus, it is a balanced system and the duct work designed accordingly does not require the use of dampers anywhere. The metal quantity for ducting is however approximately 15 per cent more than designed by the equal friction method.

It is not necessary or desirable to use static regain method to design all parts of the system. The main duct may be designed by the static regain method whereas all identical branches could be sized by the equal friction method. A designer would be the best judge to decide which method is to be used, based on the actual layout of the ducting system.

Having discussed various aspects we shall now look at the basic chart normally referred for selecting the duct sizes while using equal friction method of duct designing.

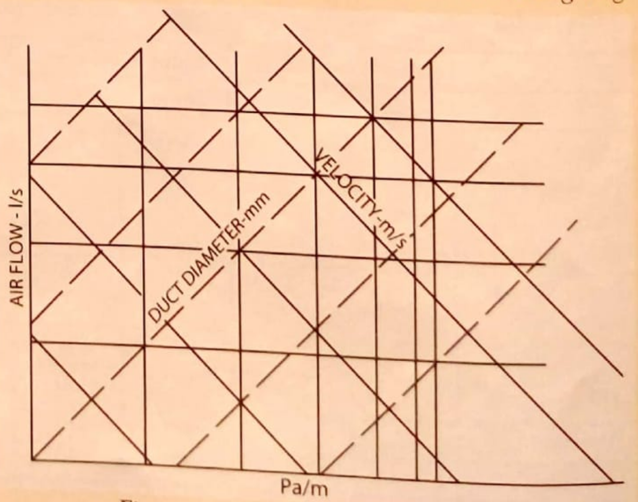



Figure 8 : Friction loss for round ducts

1. The 'X' axis of the chart represents friction loss in Pascal per meter length of duct
2. The 'Y' axis represents air quantity in liter per second
3. The velocity in meter per second, is represented by lines run diagonally from upper left to lower right
4. The round duct diameter in mm are the lines run from lower left to upper right.

From the procedure followed so far, we are now in a position to establish total equivalent straight length of the round duct run for the particular job. At first we should always design the duct section, which has the longest length.

We shall now try to explain why all the tables/charts or information so far referred is based on round duct diameter in spite of the fact that we rarely come across installations using round ducts in India. Normally the ducts are rectangular. To understand this we need to know what *aspect ratio* is and how it affects the duct friction.

Figure 9 shows four shapes of ducting having identical cross sectional area, say 1 sq. m. The shapes are however different. The first one is round duct, the second is square duct , the third is rectangular duct and the fourth is flat rectangular duct.

It can be seen that a round duct has the least perimeter and thus the smallest perimeter to cross-sectional area ratio., meaning the metal surface exposed while

| CROSS-SECTION | CROSS-SECT. AREA $m^2$ | PERIMETER m | ASPECT RATIO PERIMETER/ CROSS-SECT. AREA |
|---|---|---|---|
| 1sq.m | 1 | 3.54 | 3.54 |
| 1.0m × 1.0m | 1 | 4 | 4 |
| 2000mm × 500mm | 1 | 5 | 5 |
| 5000mm × 200mm | 1 | 10.4 | 10.4 |

Figure 9 : Effect of shape on friction rate

| | 4m × 4m | 3m × 3m | 2m × 2m | 1m × 1m |
|---|---|---|---|---|
| Cross sectional area $m^2$ | 16 | 9 | 4 | 1 |
| Perimeter - m | 16 | 12 | 8 | 4 |
| Aspect Ratio - perimeter / cross sectional area | 1 | 1.33 | 2 | 4 |

Figure 10 : Effect of size on friction rate

conveying the same quantity of air is least for a round duct. In other words, it means it has the least resistance to air flow.

Similarly if we consider different sizes of the duct but with the same shape, say for example four square ducts having sizes as 4×4, 3×3, 2×2, 1×1 and conveying air at the same velocity, we will notice that the smallest duct carrying least amount of air has the highest aspect ratio(4:1) compared to the largest duct of the same shape(1:1),again indicating that the duct having the highest aspect ratio, is the least efficient and hence we should try to aim for a aspect ratio nearer to 1 for an efficient, low initial and running cost air distribution duct system.

Ideally speaking, a round duct design is the best design as it gives the least pressure drop and all the tables/charts are therefore based on round duct size.

In actual practice we may have to use rectangular ducts and there are tables available, which provide information on equivalent size of rectangular duct, which gives the same pressure drop as if a round duct was to be selected.

Normal applications use rectangular ducts due to the following reasons :

Space consideration: since the ceiling height per floor is limited, most of the times, it is not possible to have enough space for installation of round duct and the designer has to therefore limit the height of the ducts as directed by the builder/architect or user. Hence, after calculating the round duct diameter as per the procedure mentioned earlier, the designer then selects the height of the duct and then referring to the table, establishes the width of the rectangle.

For example, let us assume a round duct of 658 mm diameter has been calculated to convey the air. We however cannot use round duct due to space limitations and are required to select an equivalent rectangular duct.

Let us assume the builder permits us a duct height of 300 mm only, and then from the table we can read the width to be provided as 1,400 mm. So that the rectangular duct of 300×1,440 mm will have the same friction drop as if we had installed 658 mm diameter round duct.

If the allowable height is 350 mm then the width would reduce to 1,100 mm whereas if the space available is very little and only 200 mm height duct is allowed then the width would be 2,500 mm, which would be a very flat duct.

As has been seen earlier, the flat duct will have the highest perimeter to cross section ratio and the maximum pressure drop. Such ducts are normally encountered in bus air conditioning applications. The headroom available for the duct is very little as space has to be provided for storing luggage over the passenger's head. This space is popularly known as hat rack.

The cost of the duct also goes up for a flatter duct as compared to a round or square duct, which is obvious as more material is required for the flatter duct to convey the same quantity of air.

A few general suggested guidelines to be followed are:

- Air should be conveyed as directly as possible to economize in power, material and space
- Sudden changes in direction should be avoided. Wherever bends are required, turning vanes should be used to minimize the pressure loss
- Air velocities within permissible limits should be used to minimize noise
- Diverging sections should be made gradual. The angle should not exceed 20 degrees
- Rectangular ducts should be made as nearly square as possible, keeping the aspect ratio minimum to reduce first cost and friction loss
- Ducts should be made of smooth materials such as galvanized steel or aluminum sheet metal
- Dampers should be provided in each branch outlet for balancing the system air flow

With this information we conclude our article on methods of duct designing.

**References**

Carrier Corporation training material

Refrigeration and Air Conditioning – a text book by C. P. Arora

**Next Issue :** Various types of systems used in air conditioning applications.

# Basics of Air Conditioning

## Part 8 (Concluding part)

*This series of articles by Ramesh Paranjpey covers the basic principles of air conditioning. The articles will serve as a source of reference for newcomers joining our industry as well as for experienced engineers wanting to brush up on fundamentals.*

This is the 8th and the last article on air conditioning fundamentals, appearing in the "classroom series". The first article appeared in July-Sept. 2002 issue of the *Journal*.

So far we have covered the following topics in earlier issues :

1. Introduction to human comfort and air conditioning
2. Properties of air
3. Understanding psychrometrics
4. Applied psychrometrics
5. Estimation of cooling load
6. Air distribution
7. Duct designing

In this concluding part, we shall cover the four normally used basic types of air conditioning systems. All these systems use the closed mechanical vapour compression refrigeration cycle to remove heat from an occupied zone and transfer it to the atmosphere.

We have discussed earlier that a well-designed air conditioning system should be able to control a) temperature b) humidity c) air quality and cleanliness d) odour by providing sufficient ventilation air and e) air velocity over the human body i.e. rate of circulation which will not cause either suffocation or undesirable drafts.

In addition to the above requirements, noise level in the occupied space, energy-efficiency of the system both at full load and partial load, ease of maintenance and flexibility of operation at full/partial loads, as well as initial cost and running costs are some of the important considerations to be taken into account while selecting a proper system for a particular application.

Obviously, most of the systems are not designed to perform all the above functions and depending on the importance of a particular parameter for the application, a system is selected which will meet at least the temperature and humidity requirements.

Air conditioning systems comprise of primary and secondary parts. The primary part is the central machine and the secondary part comprises of the air conditioning apparatus and the air distribution part.

While selecting/designing the secondary part of the system, it is first necessary to divide the space to be air conditioned into zones.

A zone can be defined as an area, which is controlled by an independent thermostat. The zone could therefore be an entire building or a floor of the building or specific areas like conference halls, visitor's waiting rooms or individual rooms.

With the cost of electronics coming down, and energy costs going up, use of multiple zones of controls are being increasingly used even for small and medium capacity projects to save energy.

As mentioned earlier, air conditioning systems can be categorized in four types, and the description of each system is itself self explanatory :

1. All Air Systems
2. All Water Systems
3. Air and Water Systems
4. Unitary Equipment /Direct Refrigerant Systems

### All Air Systems

*All Air systems* means a zone that is to be air conditioned is supplied with cool and dehumidified air as the only medium to pick up the heat. No additional cooling is required at the zone. The temperature of air within the

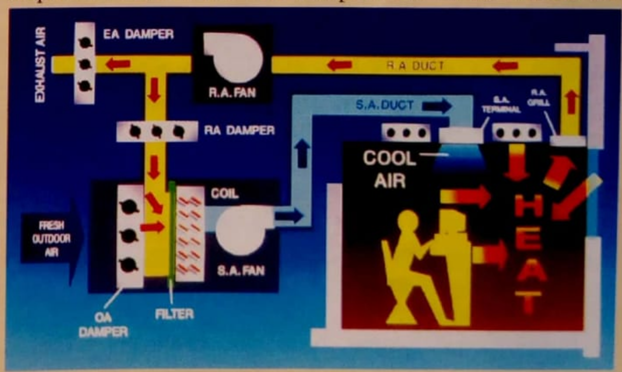



Figure 1 : Typical All-Air System

**About the Author**

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space is maintained at 24°C± 2°C and 50% RH ± 5%.

The cooled and dehumidified air picks up the heat and the warm air returns to the cooling coil where excess sensible and latent heat is removed and thus the air becomes available for reuse. This air is known as recirculated air. The latent heat removal from the air is in the form of sweating/condensation on the coil which is visible and condensed water is drained out from the air handling unit. The *All Air systems* use supply air and return air ducting for circulating air from an air handling unit to the conditioned space. The duct material could be metal, plastic, fiberglass or concrete spaces and is normally insulated to prevent heat gain as well as to avoid condensation. As can be seen from Figure 1, the air gets filtered and the contaminants are removed before the air enters the cooling coil. Depending on the requirements, there could be additional supply side filters or pre-filters of various types and efficiencies. These filters are designed to remove dirt/dust particles, pollen and other impurities from the air.

The fan, which could be located before or after the coil, provides necessary push to the air to reach the conditioned space. The fan therefore has to overcome all the resistances on the supply/return air side ducting including the resistances in the filters, cooling coils and grills/diffusers. Many times, it also becomes necessary to provide a booster fan either in the supply and/or in the return air ducting.

In order to distribute the cold/dehumidified air at the proper locations inside the conditioned space, various designs of supply air grills/diffusers with fixed or adjustable dampers are available. They also help in balancing the air distribution.

As described above, in *All Air systems*, spaces in the building are cooled or heated solely by air supplied to them from the central air conditioning equipment.

*All Air systems* can be further categorized as :

1. Single zone system
2. Reheat type
3. Variable air volume (VAV) system
4. Induction type
5. Dual duct system
6. Multi-zone system

In a ***single zone system***, the primary equipment is located as much outside the conditioned space as possible in places like outside the house, basement, service area or roof areas in commercial buildings. Ducting and terminal vents are located in the conditioned space. The connection between the two could be direct refrigerant flow, or chilled water supply or conditioned air from a central AHU.

Central systems are normally used for spaces with uniform loads where they have a minimum external load component. Major load is internal for spaces like theatres, auditoriums, department stores and public places of most buildings. The air conditioning loads are fairly uniform throughout and variations can be handled by controlling supply air temperature, air flow rate and balancing of the system. These systems are also used for spaces requiring precision control of temperature and humidity like computer data centers and telephone exchanges.

***Single-Duct, Single-Zone, Constant Volume Systems***. The main feature of this system is that it

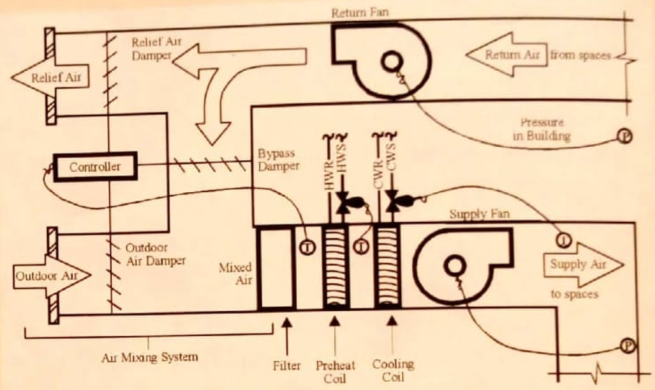


**Figure 2 : Single Duct, Single Zone, Constant Volume System**

controls the conditions in a given space, by using a single stream of air, that is pre-cooled to a desired temperature.

The scheme works as under :

Fresh outdoor air is drawn in through an intake and is mixed with return air. The mixed air is filtered, and then cooled /heated, before going to the conditioned space at a suitable temperature. A percentage of the return air must be exhausted through the relief air damper so as to maintain an appropriate ventilation rate. If the rate of exhaust air is the same as the ventilation air, the space will be maintained at atmospheric pressure. By adjusting the ventilation/exhaust flow rates, it is possible to maintain positive pressure in the conditioned space as required in some applications.

The cooling/heating coils may be before or after the fan. Return air is the air drawn from conditioned space. It is at a room temperature and full of contaminants picked up from the space. On the other hand, supply air is clean, conditioned air that is on its way to the conditioned space. The temperature of this air is lower than room temperature so that, it is able to pick up the heat load from humans, appliances, conduction through walls and any other loads.

The supply air is made up of a mixture of return and fresh air. The proportion of outdoor air and return air is normally controlled by three dampers. Cost of operating the plant reduces as the percentage of outdoor air reduces, however for human comfort and health, minimum outdoor air is essential. ASHRAE Standards suggest the required outdoor air quantities for various applications. A minimum 15 cfm per person is recommended. The proportion of outdoor air should not be allowed to be less than 25%, the minimum for adequate ventilation.

***Single-Duct, Constant Volume, Zoned Reheat Systems.*** This is a modification of the *single zone system*. The conditioned air is supplied at a fixed temperature from a central unit and designed to offset the maximum cooling load in all zones of the space. The reheat heater element is located at each zone, and is controlled by the thermostat of the respective zone. It heats the air when temperature in the zone falls below the set value. A fixed amount of air is delivered to each zone at a supply air temperature, usually 55°F. This air is adequate to meet the zone's peak load requirement. If the actual load is less than peak, then the reheat coil provides supplemental load equal to the difference between the peak and actual load.

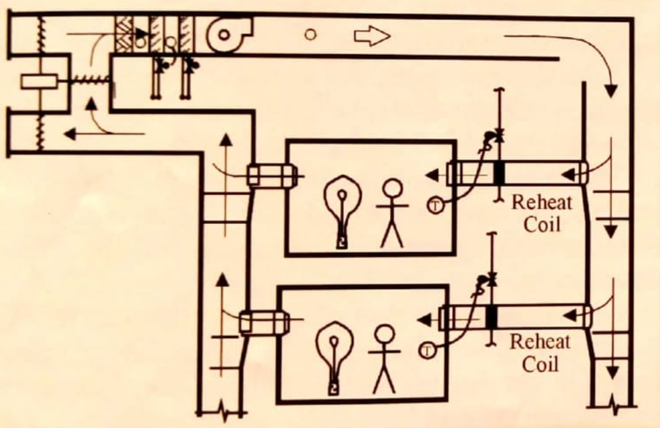


Figure 3 : Single Duct, Constant Volume Reheat System

This type of system is many times used in hospitals, laboratories, office buildings or other spaces, where wide fluctuations of load are expected.

The problem with all *reheat systems* is its energy inefficiency, whenever the cooling load is less than the peak load which normally is the operating condition in most of the applications. Thus for over 90% of the time, the cooling effect and the reheat are working against each other to neutralize their contributions. This means at no load condition, the refrigeration effect is full as well as reheat is maximum matching the cooling effect, thus both consuming very high energy. These days such type of systems are going out of favour although they are quite inexpensive and provide good temperature/ humidity control as well as good air circulation and quality.

There are also some variations possible in the system design such as the terminal reheat units may heat the primary air directly or may heat secondary air or induced air or room air directly.

***Single-Duct, Variable Air Volume Systems.*** The difference between the *reheat system* described in Fig.3 and *VAV system* (Figure 4) is that for adjusting to the varying cooling loads in different zones, a volume control damper is provided, instead of or in addition to, a reheat coil. The volume control damper is controlled by a thermostat. This means as the cooling load decreases, the cool air flow to the zone decreases until it reaches the minimum value necessary for adequate ventilation and air supply. When the minimum air flow is reached, any further reduction is made up by a thermostatically controlled reheat coil. This means some energy is wasted but it is far less than in a complete reheat system. The supply fan has a speed control, to get variable flow. Also there is an inlet vane control, a damper or a bypass duct under the control of pressure sensor in the supply duct.

The advantage of a *VAV system* is low initial and running cost due to a single duct run and simple control

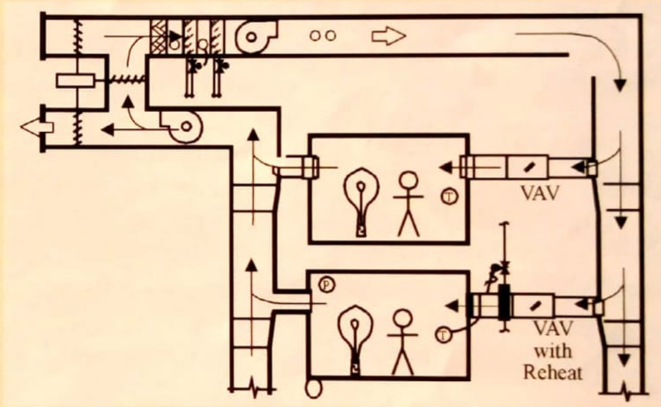


Figure 4 : Variable Air Volume System

at the end of the duct. Operating costs are low since the refrigeration and fan power closely follow the load pattern of the building, with very little overlapping of heating and cooling of the *reheat system*.

The problem encountered in such systems could be poor air circulation in the conditioned space at low loads leading to inadequate fresh air and poor humidity control.

*All Air systems* can have certain additional refinements such as :

1. Dual Path systems
2. Three Duct Multi Zone systems

***Dual Path System.*** In a single path system so far described, there is only one path carrying both cooling and heating coils as they are in series in a single duct, with a possibility of either reheat or VAV flow.

In *dual path systems*, the zone control is provided by two streams of air, one carrying cool air and the other hot air. This is done by placing cooling and heating coils in separate ducts and then mixing hot and cold air flow

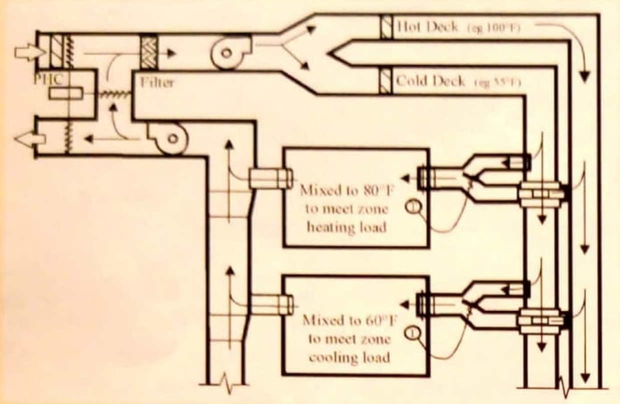


Figure 5 : Dual-Path System

streams before they enter the zone.

With the simultaneous availability of cold and hot air at each zone at all times, the system provides greater flexibility in satisfying variable loads and in providing prompt temperature responses as required.

***Three Duct System.*** In a *three duct system*, in addition to cool air and hot air, a neutral duct is provided which carries outside air. Thermal zones that require cooling receive a mixture of cold and neutral air whereas zones requiring heating use a mixture of neutral air and hot air. Thus, the *three duct system* avoids the energy waste that occurs when heating and cooling oppose one another. Therefore, operation of heating coil and cooling coil simultaneously is avoided.

The advantages of *All Air systems* can be summed up as :

1. Centrally located plant equipment allows operation and maintenance in unoccupied areas, thus also noise is contained outside the air conditioned space.

2. Wide choice of flexibility with respect to zoning and good humidity control.

3. Full design freedom for optimum air distribution, air motion and draft control.

4. Full flexibility in selecting the primary equipment.

So far we have discussed the secondary side of the system comprising of air distribution and terminal units. We shall now look at some features of a primary system.

All the primary systems use four basic components of a vapour compression systems i.e.:

1. Compressor 2.Condenser 3. Evaporator 4. Metering device. Besides these four components, there are many other accessories and controls which help in improving the system performance.

The type of each component and its location may change depending on the system selected.

The compressors could be either : 1.Rotary 2.Reciprocating (hermetic/semi-hermetic/open type) 3.Scroll 4.Screw or 5. Centrifugal type

The condensers could be either : 1. Air-cooled 2. Water-cooled or 3. Evaporative

The evaporators could be Direct Expansion or Flooded type, with or without pump circulation.

And the metering devices could be either : 1. Capillary tube/orifice tube 2.Thermostatic expansion valve or Liquid level controller.

The design and selection of the components from the list mentioned above depends on individual manufacturer/assembler of packages and the system designer.

An *All Air system* does not necessarily mean that on the primary side only an air-cooled condensing unit is used. The condensing unit could be air-cooled or water-cooled, as shown in Figure 7, where the primary system is using chilled water as a medium for circulation in the cooling coils. These type of coolers or water chillers are common for projects above 30 to 40 ton capacity requirements. The chilled water is normally supplied at 6 to 7°C and after picking up heat returns to the chiller

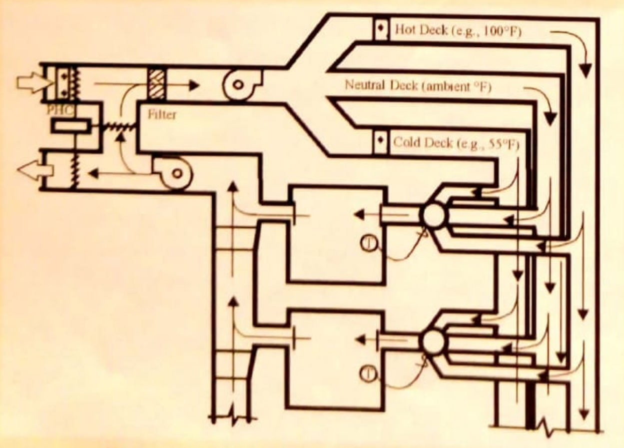


Figure 6 : Three-Deck Multi-zone System

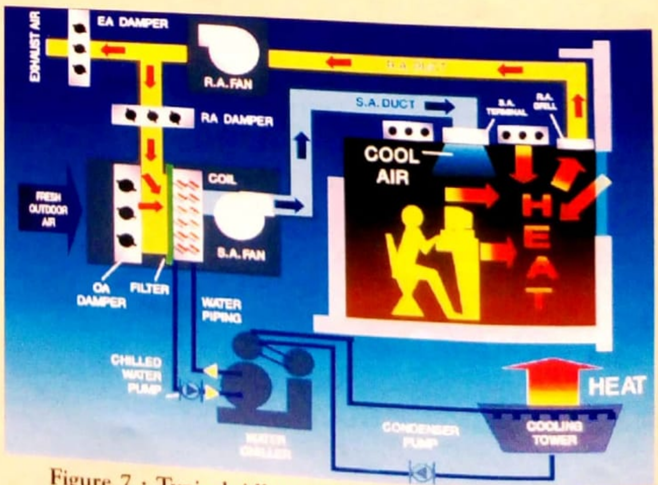


Figure 7 : Typical All-Air System (Water-Cooled Type)

at around 12°C. The shell and tube condenser rejects heat to the atmosphere through a cooling tower. The water inlet temperature could be 32°C with water outlet temperature from condenser around 36°C. This 36°C water rejects heat to the atmosphere in a cooling tower and gets cooled to 32°C for reuse in the condenser. The condenser could be also air-cooled, in which case the hot refrigerant rejects heat directly to atmosphere.

Whether to use an air-cooled condensing unit or a water-cooled unit depends on several factors such as the availability of water year round, quality of water as well as location of the space to be air conditioned.

In general, water-cooled systems tend to be more efficient and consume less energy compared to air-cooled systems.

## All Water Systems

*All water systems* rely on the cooling/heating effect to be delivered to air conditioned spaces solely from the water loop that supplies chilled water generated from a centrally located chiller and the pump to carry chilled water to fan coil units. The outdoor air is supplied to the room by a fan or by infiltration. Thus, there is no primary treated air. The water drain is essential to remove condensation from the coil since air while getting cooled sheds excess moisture and gets dehumidified.

The greatest advantage of these systems is that they carry water instead of air to the space. The density of water being higher, less space is required. In hotels and commercial buildings, where space is at a premium, this serves as a great advantage. For example, a 5/8" diameter pipe carrying 2.4 gpm at 6 fpm velocity has the same cooling effect as 8" diameter ducting carrying 400 cfm at 1200 fpm velocity air.

In addition to space saving, the other advantage is that these systems can be modified or retrofitted easily. They also have individual room control responding quickly to room thermostat settings and there is no possibility of air getting mixed with air from other conditioned spaces.

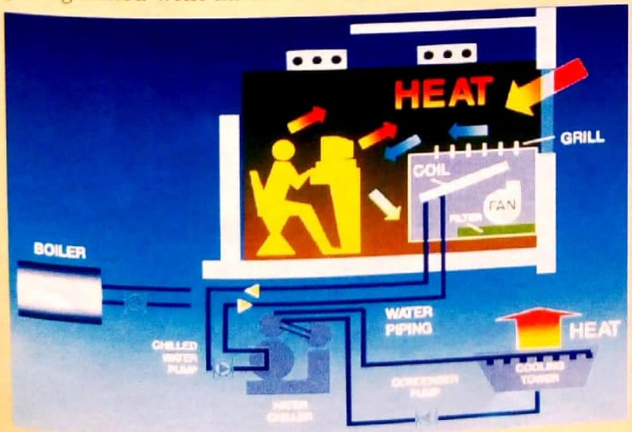


Figure 8 : Typical All-Water System

## Air and Water Systems

In *air and water systems* both air and water are distributed to the conditioned space to control temperature and humidity. The air is cooled at a central machine room and is circulated to different zones with ducting similar to all air systems. The water is also cooled in a central plant and then pumped to coils located in the conditioned space.

The air delivered from the central air handling unit is known as primary air. The room air is circulated over the chilled water coil located in the conditioned space and termed as secondary air.

The combination of *air and water system* is more expensive and complicated as it includes additional piping for chilled water and it's related controls. The advantage being, each room has individual temperature

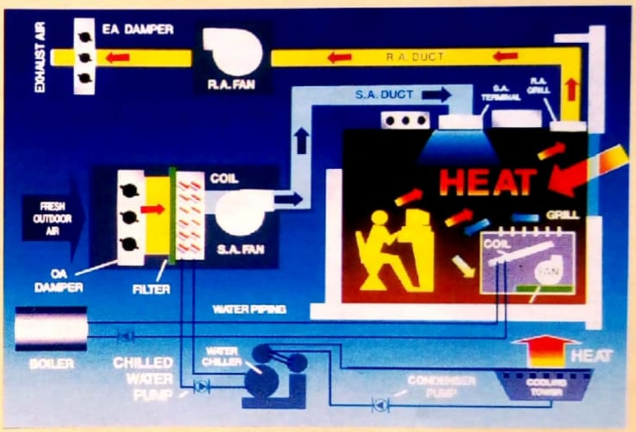


Figure 9 : Typical Air-Water System

control and provides more uniform and closer temperature control throughout the year. Since the load can be shared by two systems, the quantity of primary air to be provided can be reduced, thus saving in duct space and use of smaller air handling units.

## Unitary Systems

Unitary air conditioners/heat pumps are factory assembled units using matched components like compressor, condenser, evaporator and metering device. A pre-determined and optimized refrigerant is charged in the unit and then each unit is factory tested for performance prior to delivery.

Only the external duct work, electrical power connection and condensate piping are required for completing the installation. Many times the equipment is in two parts requiring connecting refrigerant piping between the indoor and outdoor condensing sections.

There are many types of combinations available with the window air conditioner, as shown in Figure 10 as the simplest form. These are mostly used in capacities upto 2 ton.

Split air-conditioners are also popular up to 3 tons which use an indoor fan-coil unit and an outdoor condensing unit.

Beyond this capacity, one can have an option of using a floor mounted evaporator unit in the conditioned space and an air-cooled condensing unit outdoor. More than one indoor unit can be connected to a common outdoor unit.

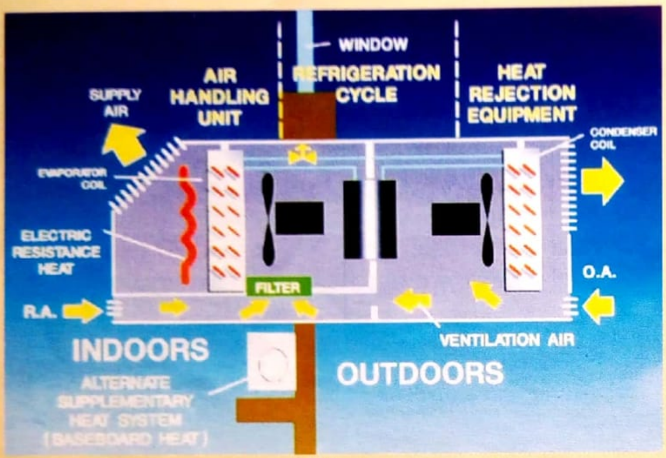

Figure 10 : Typical Window Air Conditioner

For capacities of 5 ton and above, the most popular units are single package water-cooled floor standing unit and these are used mostly up to capacities of 15 ton. Multiple numbers can be installed in a zone if the capacity requirement is higher.

Air-cooled ductable splits are also popular for capacities of 3 ton and above up to 20 tons capacity. These could be mounted on a terrace or in the space above a false ceiling or in the side room.

The advantages of unitary products are :

1. Consistent performance as per published ratings and assurance by the manufacturer using matched components.

2. Simple and inexpensive room control with individual air distribution for each room.

3. Ready availability and replacement in case of malfunction.

4. Quick and easy installation.

The disadvantages are :

1. Fixed air quantity and no possibility of fresh air provision in some designs.

2. Less efficient because of compact size and factory matching components rather than custom matching for optimum performance.

3. Use of air-cooled condensers most of the time thus not taking advantage of lower temperatures available with evaporative condensers.

4. Mostly use on-off control, thus not very energy efficient and this decreases thermal performance.

We have described the major categories of various systems. Besides these, there are some other systems which are also becoming popular now a days, such as thermal storage systems generating cooling effect during off peak hours and making use of this during normal working hours. The advantage is taken where low electricity tariffs exist at night time during off-peak hours. Heat recovery wheels for improved energy efficiencies are also being used increasingly.

There are also direct refrigerant circulation systems (VRF), and these can be termed as more advanced versions of split systems. The system has a common condensing unit located outdoors and connected to one or more different types of indoor units which could be cassette type, floor mounted, wall mounted or ceiling suspended type. It is not necessary that all indoor units should be of same type and same capacity. Different combinations are possible with a common outdoor condensing unit.

The quantity of refrigerant circulating directly in the low side unit is regulated with variable speed control known as inverter control for compressors. The system has long refrigerant lines running from the condensing unit to individual fan coil units and requires very high skills of installation and quality components to ensure refrigerant does not leak. The refrigerant piping could run up to 100 meters. These systems save considerable energy and are extremely flexible. Individual zone units can be switched off when not required and proportionate energy saving is possible.

As can be seen from the foregoing there are many different permutations and combinations of various components possible. It is up to the system designer to select an appropriate combination which meets the customer's needs and gives satisfactory performance throughout the year while keeping the owning and operating costs to the bare minimum.

With this information on various systems, we conclude our series on air conditioning. The author hopes that readers who have followed these articles have benefited to some extent in getting the basic information. There are plenty of good reference books, like ASHRAE volumes, training materials, journals publishing latest trends in air conditioning technology besides a wealth of information on the various topics available on the internet, and it is up to the individual to collect and enrich his own knowledge on this subject as a continuous on-going effort.

## Reference

1. Carrier Corporation training material.

2. Fundamentals of HVAC Systems-ASHRAE Continuing Education Series.