PDHonline Course M250 (4 PDH)

HVAC Air Handling Unit Design Considerations

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SECTION -1 AIR-HANDLING UNIT OVERVIEW

An air handling unit often abbreviated as AHU, is a factory fabricated assembly consisting of fan, heating and/or cooling coils, filters, dampers and other necessary equipment to perform one or more of the following functions of circulating, cleaning, heating, cooling, humidifying, dehumidifying, and mixing of air.

Air handling systems range in complexity from stand-alone units that serve individual rooms to large, centrally controlled systems serving multiple areas in a building. HVAC engineers refer to the areas served by an air handling system as zones. The smaller the zone, the greater the likelihood that good control will be achieved; however, equipment and maintenance costs are directly related to the number of zones. Some systems are designed to provide individual control of rooms in a multiple-zone system. Sometimes AHUs discharge (*supply*) and admit (*return*) air directly to and from the space served, without ductwork. They can be used in many other applications, for example, they can be used for providing smoke control, maintaining pressure relationships between spaces, and providing fresh air for occupants.

All standard air handlers are designed to move a certain amount of air typically 400 cubic feet per minute (CFM) for 1 ton of refrigeration. Air handling systems utilize various types of equipment, arranged in a specific order, so that space conditions can be maintained. Within each basic component there are different types and styles, each with their own operating characteristics and efficiency, method and materials of construction, and cost, all of which greatly affect the initial design and resulting operating economics of the system.

The units sizes range from small modular units of 2000 CFM to large custom built units of 30000 CFM and higher. Larger air handlers that condition 100% outside air, and no re-circulated air are known as makeup air units (MAUs). These are commonly used in places where the ventilation (outside air) load is very high. The roof top units (RTU) are other common air-handlers designed for outdoor use. These are usually mounted on roof curbs but can be also mounted on structural supports or on grade. These are particularly popular for single-story commercial buildings, warehouses and big stores. Supply air and return air ducts connect to the bottom (vertical discharge) or side (horizontal discharge) of the unit.

Air handling units are configured to be either blow-through or draw-through units. Blow-through unit is when one of the coil sections is located downstream of the supply fan. Draw-through unit is when one of the coil sections is located upstream of the supply fan. Draw-through units can be further configured to be either horizontal units or vertical units.

AHU Sizing Factors

All Air Handlers are designed to heat and/or cool specific amount of air. The size of the unit is normally estimated to meet the load demands in British Thermal Units (BTUs) and a specific amount of air flow rate in Cubic Feet per Minute (CFM) at a designed static pressure or pressure drop. The amount of motor horsepower required to move this amount of air at the designed static load is then applied.

Determine Cooling & Heating Load

A load calculation determines the capacity of AHU, more specifically the capacity of the heating/cooling coil. The thermal load is estimated in units of BTUs. It factors in a number of criteria such as building square footage, physical makeup of the various envelope surfaces, solar load factors, construction materials, number of windows, lighting and indoor appliances, occupancy, and year-round weather concerns. The total cooling load for the space is determined by the summation of all of the calculated heat gains. Whilst the subjects of heat loads and air quantity or temperature calculations are fully dealt with in PDH course M-196, “HVAC made easy – a guide to heating and cooling load estimation” and publications such as the ASHRAE guide, the cooling load calculations involves the following steps:

1. Determine the zone cooling load. This consists of external loads through the building envelope and internal loads from people, lights, appliances, infiltration loads, moisture loads, and other heat sources.

2. Calculate the system cooling load. This step adapts the selected air distribution system to the zone load and involves the introduction of the required outdoor air volume into the air conditioning system for ventilation.

3. Calculate the design temperature of supply air at the terminal (allowing for all normally calculated extraneous gains such as fan heat and duct wall heat pick-up) is arrived at by conventional psychometric calculations.

4. Calculate the individual maximum module or zone air volumes are calculated using the maximum room sensible heat gain figure and the temperature difference between supply air temperature at the terminal and design space temperature. These air volumes are then used to select air diffusers.

5. Maximum simultaneous cooling and air supply requirements are then computed in order to establish the capacity of air handling units and central cooling equipment. Care must be taken in design that ducting is adequately sized to convey the correct amount of air to every module or zone under all conditions.

A proper load calculation is important so that your system operates at maximum efficiency. Most facilities have systems that are oversized, so you end up paying more than you should to heat and cool your space. Also, systems that are improperly sized tend to cycle too quickly or too slowly and that can lead to maintenance problems, uneven temperatures, and an inability to control humidity. A properly sized system based on a correct load calculation can help ensuring long-lasting, reliable comfort in all seasons.

Determine Airflow

Air flow calculation determines the fan capacity, which in turn affects the physical size (foot print) of air handling unit. This is single most important factor while conceptualizing the space requirements for mechanical rooms and also the air-distribution ducts.

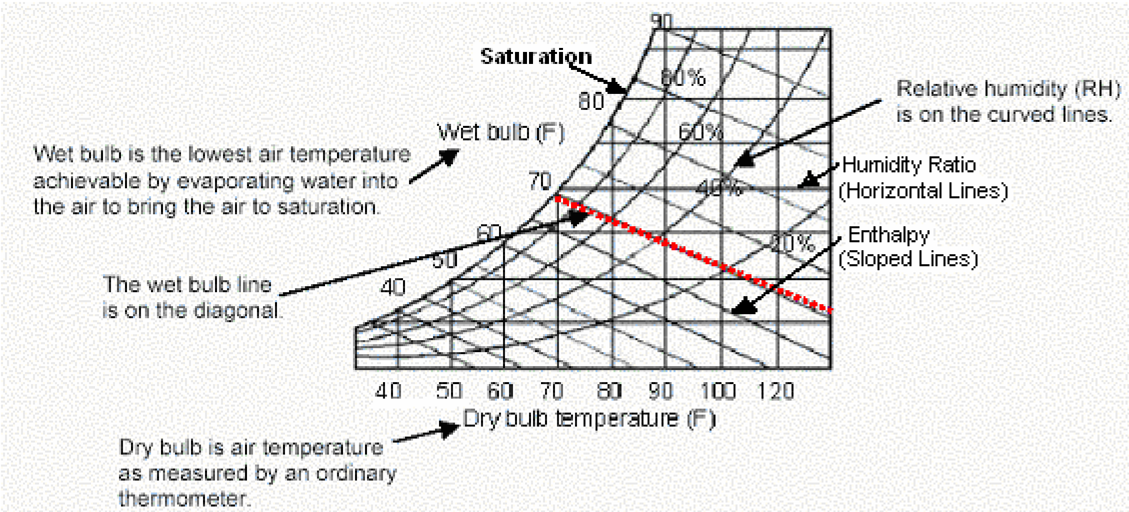
The volume of conditioned air depends on the heating, cooling and ventilation loads and is estimated in cubic feet per minute (CFM). Once the airflow rate is determined next step is to estimate the static pressure drop across the distribution system comprising of ductwork, fittings and AHU components. When both air flow and static pressure is known, the appropriate fan selection is made.

HVAC engineers use "Psychrometric" analysis for sizing the coils and for estimating the air flow rates required to be pushed into the ducting system. The psychrometric chart, a graphical representation of the thermodynamic properties of moist air, is an invaluable aid in illustrating and solving air conditioning problems. Let's refresh some elementary psychrometrics.

PSYCHROMETRIC CHART

Psychrometry deals with the determination of the thermodynamic properties of moist air and the utilization of these properties in the analysis of conditions and processes involving moist air.

The basic coordinate grid lines of the chart are enthalpy, which slopes up to the left, and the humidity ratio (or specific humidity) which runs horizontal. Dry-Bulb temperature is illustrated by vertical lines uniformly spaced. Wet-Bulb lines slope similarly to enthalpy lines, but the slope increases as the temperature increases. Note that wet-bulb line is not parallel to an enthalpy line; this is because of the heat added to the air mixture by the moisture, as it changes from dry to saturated air. Relative humidity lines are curved, with the saturation line defining the upper boundary of the chart.

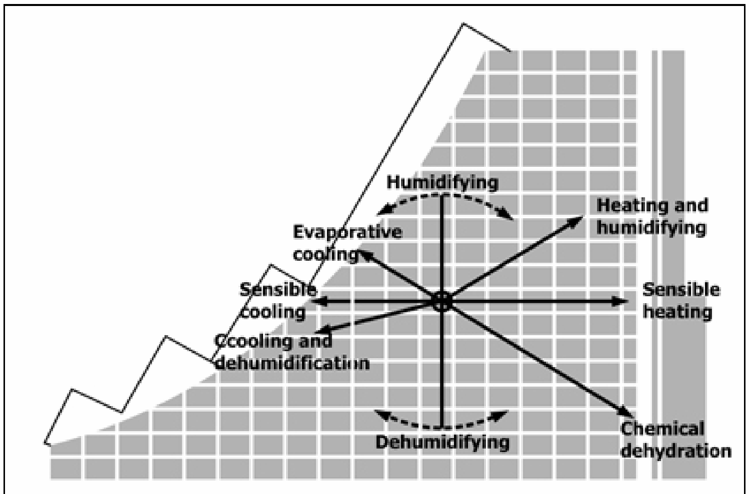


Psychrometric Parameters

AIR-CONDITIONING PROCESSES

Common processes include:

- Sensible cooling/sensible heating
- Cooling and dehumidification/heating and humidification
- Humidification/dehumidification
- Evaporative cooling/chemical dehydration



Air-conditioning Processes

Sensible Heating and Cooling

Sensible heating and cooling simply involve changing the dry bulb temperature of the moist air without changing the moisture content. The humidity ratio remains unchanged, and so we use a horizontal line on the psychrometric chart to represent this process. This provides no control over the relative humidity and may produce conditions that are uncomfortable. Heating will result in lower relative humidity and cooling will result in a higher one. Sensible heating/cooling process results in changes in dry-bulb, wet-bulb, RH, and enthalpy whereas specific humidity and dew-point temperature remain constant.

Heating and Humidification

In heating and humidifying both sensible heat and specific humidity increase-shown as a line sloping upward and to the right. Adding humidification is often necessary to make a heated space comfortable. One example of humidification is the adiabatic-saturation process, when air is passed over a moist surface to gain moisture by evaporation. In this case the dry bulb temperature could be increased or decreased, depending on the temperature of the evaporating water. Changes occur in dry-bulb, wet-bulb, dew-point temperatures, and enthalpy. A difference in RH depends on the slope of the line.

Cooling and Dehumidification

Cooling and Dehumidification involves lowering the air temperature below the dew point and letting some of the moisture condense out. On psychrometric chart, this is represented as line sloping downward and to the left. This process is assumed to occur as simple cooling first and then condensation. While the moisture is condensing the air is assumed to remain saturated. Cooling and dehumidification process results in changes in dry-bulb, wet-bulb, dew-point temperatures, specific humidity and enthalpy. Changes in RH are dependent on the slope of the line.

This process is used in air-conditioning systems operating in hot, humid climates. It is accomplished by using a cooling coil with a surface temperature below the dew-point temperature of water vapor in air.

Evaporative Cooling

This method adds moisture to the air stream while cooling it, and so would be useful in dry and arid regions where the warm air has a low humidity. The process takes place upward along the wet-bulb line. As sensible heat of the initial air vaporizes the water, the air's dry-bulb temperature will fall. The sensible heat used to vaporize the water enters the air as latent heat in added vapor; thus no heat is added or removed. Wet-bulb temperature remains constant. Dew-point temperature, RH, specific humidity and enthalpy will increase. (In most evaporative cooling installations, heat may be added or removed during the process due to outside sources, this amount is usually negligible.)

Chemical Dehydration

This is also referred to as desiccant (or chemical) dehumidification, which takes place when air is exposed to either solid or liquid desiccant materials. The mechanism of dehumidification in this case is either absorption (when physical or chemical changes occur) or adsorption (when there are no physical or chemical changes).

Dehumidification by solid desiccants is represented on the psychrometric chart by a process of increasing dry-bulb temperature and a decreasing humidity ratio. Dehumidification by liquid desiccants is also represented by a similar line, but when internal cooling is used in the apparatus, the process air line can go from warm and moist to cool and dry on the chart.

SENSIBLE HEAT FACTOR

The thermal properties of air can be separated into latent and sensible heat. Sensible Heat Factor is the ratio of sensible heat to total heat.

$$SHF = \frac{SH}{SH + LH} = \frac{SH}{TH}$$

SHF = sensible heat factor

SH = sensible heat

LH = latent heat

TH = total heat

The ratio of space sensible cooling to total cooling is useful for plotting the slope of the path that supply air travels after introduction into the space.

ROOM SENSIBLE HEAT FACTOR (RSHF)

The room sensible heat factor (RSHF) represents the psychrometric process of the supply air within the conditioned space. Room Sensible Heat Factor is the ratio of room sensible and room latent heat.

$$RSHF = \frac{RSH}{RSH + RLH} = \frac{RSH}{RTH}$$

RSHF = room sensible heat factor

RSH = room sensible heat

RLH = room latent heat

RTH = room total heat

The supply air to a conditioned space must have the capacity to offset simultaneously both the room sensible and room latent heat loads. The process is plotted on the standard psychrometric chart as below.

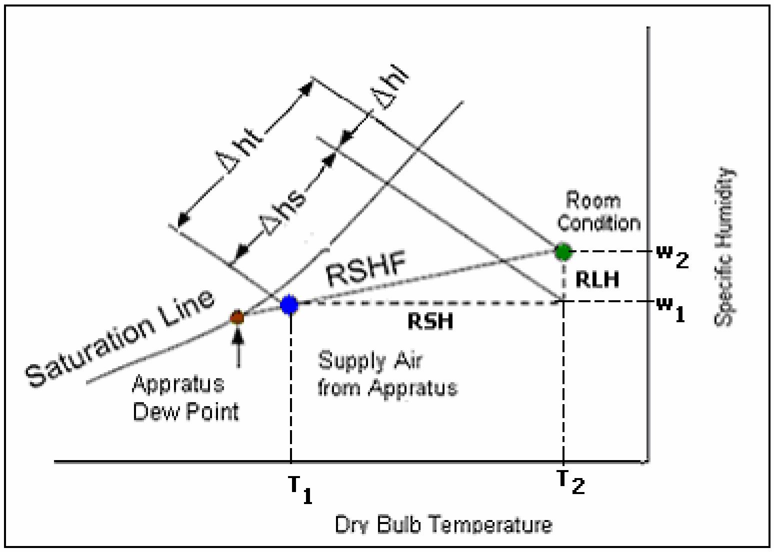


FIGURE - 1

The slope of the RSHF line illustrates the ratio of sensible to latent loads within the space. RSH is projected on the enthalpy lines to estimate the Δhs (sensible load) and RLH is projected on the enthalpy lines to estimate the Δhl (latent load). Room total load (RTH) is the enthalpy difference between the room design condition and the supply air condition.

Thus, if adequate air is supplied to offset these room loads, the room requirements will be satisfied, provided both the dry- and wet-bulb temperatures of the supply air fall on this line.

GRAND SENSIBLE HEAT FACTOR (GSHF)

Grand Sensible Heat Factor is the ratio of the total sensible heat to the grand total heat load that the conditioning apparatus must handle, <u>including the outdoor air heat loads</u>. This ratio is expressed as:

$$GSHF = \frac{TSH}{TLH + TSH} = \frac{TSH}{GTH}$$

GSHF = grand sensible heat factor

TSH = total sensible heat

TLH = total latent heat

GTH = grand total heat

The air which is passing through the AHU coil increases or decreases the temperature and/or the moisture content. The amount of rise or fall is determined by the total sensible and latent heat load that the conditioning apparatus must handle. The condition of the air entering the apparatus (mixture condition of outdoor and returning room air) and the condition of the air leaving the apparatus is plotted on the psychrometric chart as shown below.

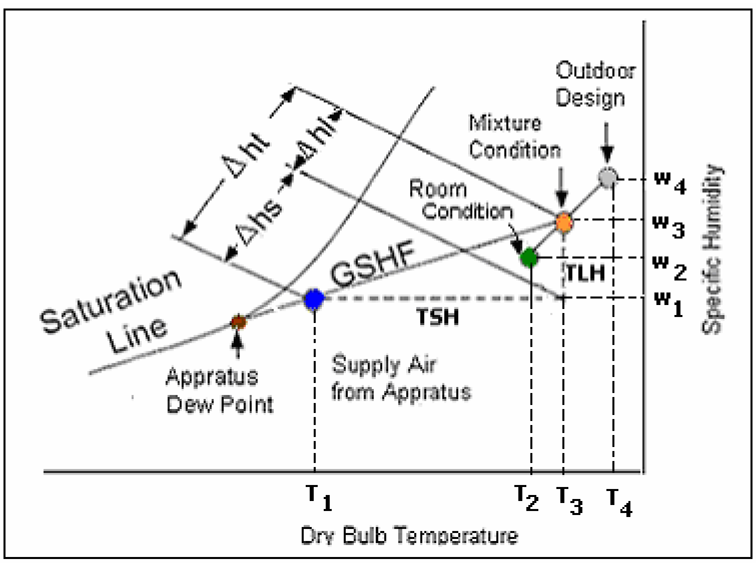


FIGURE - 2

GSHF Line is plotted between mixture conditions to supply air leaving condition from apparatus. The slope of the GSHF line represents the ratio of sensible and latent heat that the <u>apparatus must handle</u>. GSH is projected on the enthalpy lines to estimate the Δhs (grand sensible load) and RLH is projected on the enthalpy lines to estimate the Δhl (grand latent load). Grand total load (GTH) is the enthalpy difference between the mixed air condition and the supply air condition.

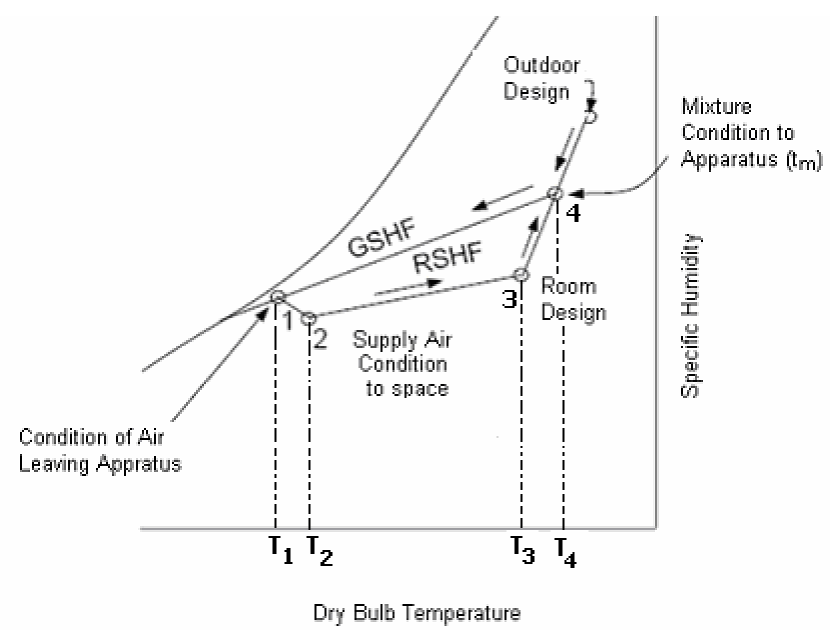
Notes

1. *The RSHF line in Figure -1 is the line sloping between room condition and the supply air temperature whereas the GSHF line in Figure – 2 is the line sloping between the mixture condition and the supply air temperature.*

2. *The Apparatus dew point (ADP) is the coil surface dew point temperature required to accomplish a cooling/dehumidifying process.*

REQUIRED AIR QUANTITY

The air quantity required offsetting simultaneously the room sensible and latent loads and the air quantity required through the apparatus to handle the total sensible and latent loads may be calculated, using the conditions on their respective RSHF and GSHF lines. For a particular application, when both the RSHF and GSHF ratio lines are plotted on the psychrometric chart, the intersection of the two lines represents the condition of the supply air to the space. It is also the condition of the air leaving the apparatus. The figure below illustrates what actually happens when these supplementary loads are considered in plotting the RSHF and GSHF lines.



RSHF and GSHF Line Plotted with Supplementary Load Line

Point (1) is the condition of air leaving the apparatus and point (2) is the condition of supply air to the space. Line (1 - 2) represents the temperature rise of the air stream resulting from fan horsepower and heat gain into duct. Point (3) is the room design condition and point (4) is the mixture conditions entering the cooling coil (result of mixing return air and outside air).

The air quantity (mass flow rate) required to satisfy the room load may be calculated from the following equation:

$RSH = m * Cp * \Delta T$

In volumetric terms air flow rate in cubic feet per minute (CFM) is estimated below:

For air at $\rho = 0.075$ lbm/ft^3 and $Cp = 0.24$

$RSH = 0.075 * 0.24 * CFM \times \Delta T / 60$

$RSH = 1.08 \times CFM \times (T_3 - T_2)$

$CFM = RSH / [1.08 \times (T_3 - T_2)]$

The air quantity required through the air conditioning apparatus, to satisfy the total air conditioning load (including the supplementary loads) may be calculated from the following equation:

$GSH = 1.08 \times CFM \times (T_4 - T_1)$

$CFM = GSH / [1.08 \times (T_4 - T_1)]$

For all practical purposes, the required air quantity supplied to the space is equal to the air quantity required through the apparatus, neglecting leakage losses.

In well designed, tight systems the difference in supply air temperature and the condition of the air leaving the apparatus ($T_2 - T_1$) is usually no more than a few degrees. The difference in temperature between the room and the air supply to the room determines the amount of air required to satisfy the room sensible and latent loads. As this temperature difference increases (supplying colder air, since the room conditions are fixed), the required air quantity to the space decreases. This temperature difference can increase up to a limit where the RSHF line crosses the saturation line on the psychrometric chart; if it is assumed that the available conditioning equipment is able to take the air to 100% saturation.

The value of T_4 is determined by hit and trial. When the supply, return, and outdoor air temperatures are known equation below can be used to calculate outside air percentage or the mixed air temperature.

$$\%\ \text{Outside Air} = \frac{T_{\text{Mixed Air}} - T_{\text{Return Air}}}{T_{\text{Outside Air}} - T_{\text{Return Air}}}$$

This equation assumes that perfect mixing occurs; the temperature or moisture content of the mixed air stream will be the same regardless of where they are measured in the air stream. This approach works best when there are significant differences between the outdoor air temperature and the return temperature.

Once the required air flow rate (CFM) is known, the room specific humidity differential (Δw in lb of water vapor per lb of dry air) can be estimated as:

$$\Delta\omega = \text{RLH} \times \frac{1}{\text{CFM}} \times \frac{1}{\rho} \times \frac{1}{h_w} \times \frac{1}{60}$$

Where

Δw = Specific humidity differential in lb_w/lb_a

RLH = Room latent heat, Btu/hr

CFM = calculated air flow rate

ρ = air density (0.075 lb/ft^3 at 60ºF)

hw = latent heat of vaporization (1059 Btu/lb, at 60ºF)

60 = min/hr

Example

Consider an interior core room of a building has a sensible load of 120000 Btu/h; a latent load of 30000 Btu/h and is conditioned by a single zone AHU. The inside conditions to be maintained are 78°F dry bulb and 50% RH and the design supply air temperature is 58°F. Determine the air flow requirements.

The design airflow rate (CFM) will be:

$CFM = RSH / [1.08 \times (T_{Room} – T_{Supply\ Air})]$

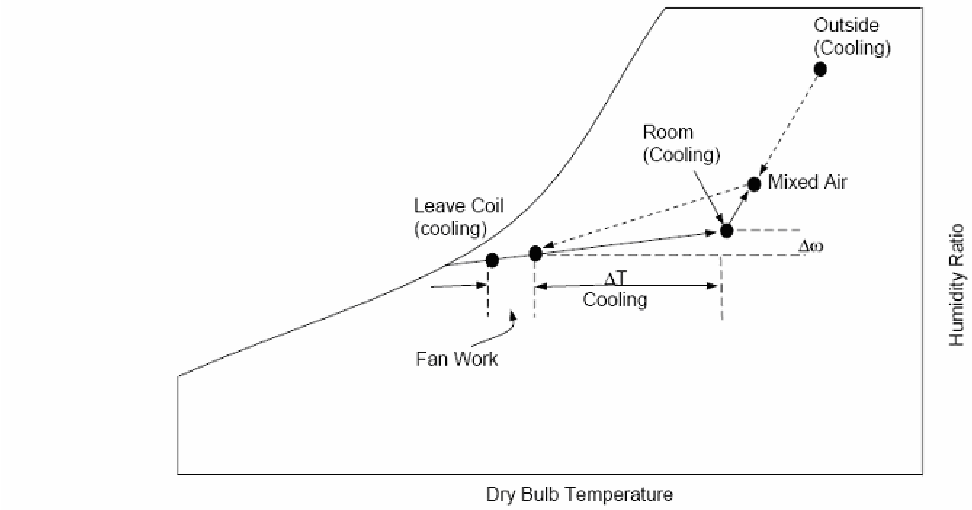
Air flow rate (CFM) = 120,000 / [1.08 x (78 - 58)]

Air flow rate (CFM) = 5555 CFM

The point defined by these two differential values can be plotted on a psychrometric chart. The validity of the point must be verified, based on the cooling capabilities of the coil, and the AHU arrangements.

Psychrometric representation for draw-through AHU arrangement

We have learnt that when both the RSHF and GSHF ratio lines are plotted on the psychrometric chart, the intersection of the two lines represents the condition of the supply air to the space. It is also the condition of the air leaving the apparatus. However this neglects fan and duct heat gain, duct leakage losses, etc. In actual practice, these heat gains are taken into account in estimating the cooling load. Therefore, the temperature of the air leaving the apparatus is not necessarily equal to the temperature of the air supplied to the space. For a draw-through arrangement, that is with the supply fan past the cooling coil- The supply air temperature will be greater than the "coil leaving" temperature, because of the heat added by the fan work.



For this example if a 5hp motor is required, the temperature rise will be:

$$\Delta T = P \times \frac{1}{cfm} \times \frac{1}{1.08}$$

$$\Delta T = 5 \times 2545 \times \frac{1}{5555} \times \frac{1}{1.08}$$

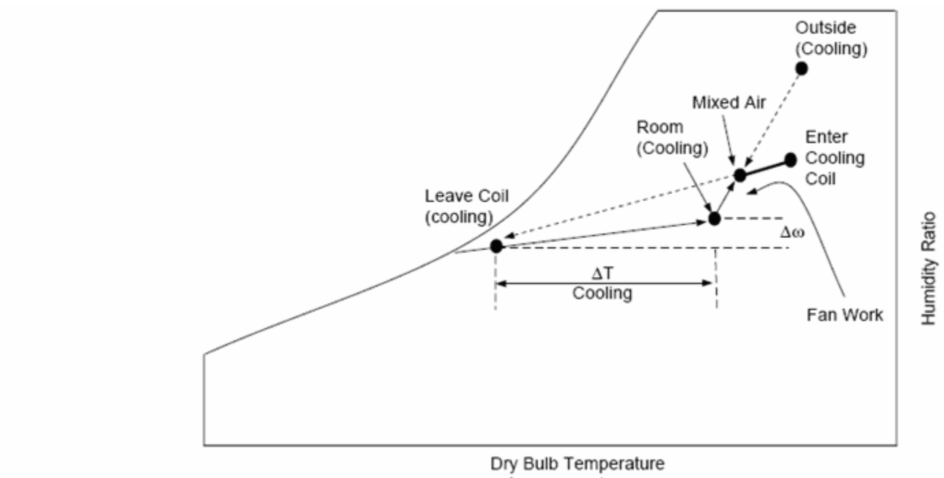
Where

P is the power rating in hp and 2545 is Btu/hp

$\Delta T = 2.1$ºF

Thus the actual coil leaving condition shall be 58 – 2.1 = 55.9ºF DB which should be plotted on the chart.

For a *blow-through* arrangement, the fan work causes an increase in the mixed-air temperature before the air goes through the cooling coil, and the process is as in figure below. For this case, it is necessary to increase the supply of air ΔT to 22ºF to get a valid "coil leaving" condition. This will reduce the air quantity to 5050 cfm and will require more care in the air distribution system to avoid cold air spillage and drafts. The Δw will be greater due to the reduction in cfm. Humidity control is not always required, but some upper limit will be inherent in any refrigeration-type cooling process-chilled water, brine, or direct expansion.



Psychrometric representation for blow-through AHU

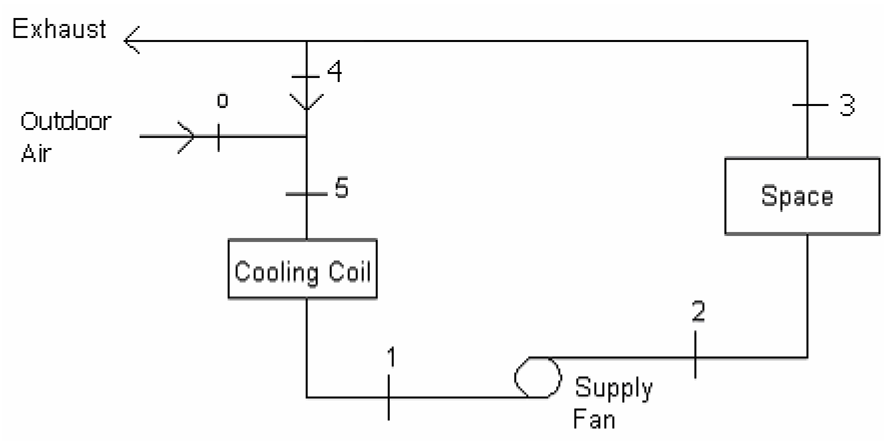
Summarizing, draw-through units typically require 10% more supply air than blow-through systems for the same temperature off the cooling coil. This will increase duct size and fan operating cost. The fan heat will ensure the supply air is not fully saturated, avoiding moisture issues.

Blow-through units add the fan heat (usually equivalent to 2-3°F) before the cooling coil. The leaving air temperature from the cooling coil then becomes the supply air temperature. This provides the maximum temperature rise between the cooling air and the space design temperature (or the least amount of supply air will be required). Since the air is often fully saturated and moisture may be an issue, blow-through should not be used with final filters downstream of the coils.

COOLING COIL CAPACITY

A cooling coil serving one or more conditioned spaces is sized to meet the highest sum of the instantaneous space loads for all the spaces served by the coil, plus any external loads such as fan heat gain, duct heat gain, duct air leakage, and outdoor air ventilation loads (sensible and latent). At design condition, a cooling coil provides design air flow at design off-coil air temperature and humidity, which are determined to meet each zone's temperature and humidity requirements. For dehumidification applications, the cooling coil should have adequate latent cooling capacity as well as sensible cooling capacity.

Find below a simple cooling system:

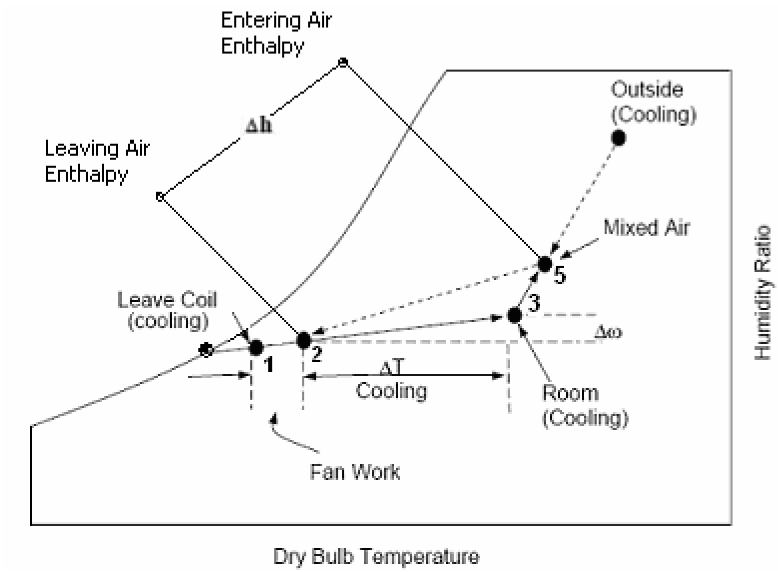


Simple Schematic of Cooling Unit

The nodes description is as follows:

1. The outside air conditions are located at point "o" in figure above.
2. The coil leaving conditions are located at point 1.
3. The supply air conditions are located at point 2 (note with the arrangement depicted, supply fan motor heat is added to the air stream)
4. The inside air conditions are located at point 3.
5. The mass flow rate of return air is located at point 4 (temperature conditions are essentially similar to point 5, but the mass flow rate varies depending on what portion of return air will be exhausted)
6. The mixture conditions are located at point 5 (note proportion of return air and outdoor air are mixed)

Plotting these nodes on the psychrometric chart:



The room air flow is given by equation:

CFM = RSH / [1.08 x ($T_3 – T_2$)]

Normally, zone air flow is calculated based on design sensible loads and design supply air flow, which may not meet humidity requirements. For dehumidification applications, the design air flow must meet both temperature and humidity requirements. A psychometric chart is used to determine the enthalpy at point 5 and point 2 and the mass flow rate at point 1. The total cooling coil load is calculated by the following equation:

$Q_{coil} = m_1 * (h_5 - h_2)$ or

Empirically

$Q_{coil} = 4.5 \times CFM \times (h_5 - h_2)$

Where:

- m_1 is the mass flow rate at point 1
- h_5 is the enthalpy at point 5
- h_2 is the enthalpy at point 2

Coil Load Calculation Example

Determine the minimum cooling coil load and required chilled water flow for a system serving an office space with a sensible cooling load of 30,800 Btu/h and a latent load of 13,500 Btu/h. Indoor design conditions are 75°F/50% RH, and Outdoor design conditions are 85°F DB/70oF WB. The minimum ventilation air is 25% of the total supply air which is delivered at 55°F. The fan selected is a 70% efficient "draw-through" arrangement, and delivers air at 4.0 inches total pressure. No manufacturer's coil data is available. Assume 10°F chilled water temperature rise through the coil.

RSH = 1.08 x CFM x ΔT

CFM = RSH / (1.08 x ΔT)

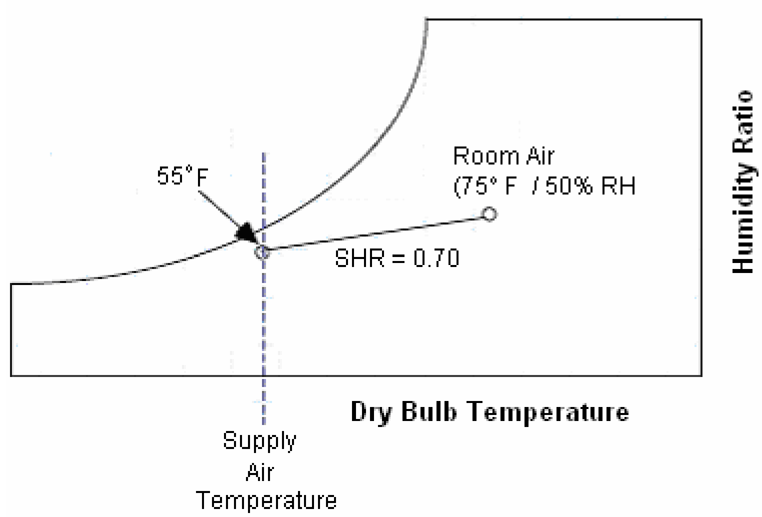
CFM = 30,800 / [1.08 (75 - 55)]

CFM = 1425

SHR = 30,800 / (30,800 + 13,500) = 0.70

<u>Psychrometrics</u>

Plot Space Conditions of 75°F/50% RH Find the intersection of the SHR line with space temperature and supply air temperature



Plot Outdoor Conditions of 85°F/ 70°F WB; Locate Mixed Air Condition (25% OA) and Account for Fan Δt (consider 2 degree rise).

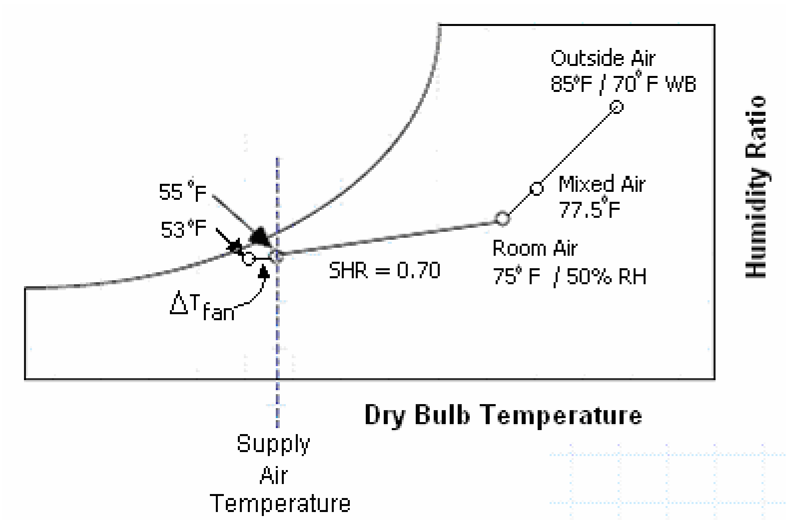
Assuming the perfect mixing occurs, the mixed air conditions can be found by equation:

$$\%\ \text{Outside Air} = \frac{T_{\text{Mixed Air}} - T_{\text{Return Air}}}{T_{\text{Outside Air}} - T_{\text{Return Air}}}$$

$T_{\text{Mixed Air}} = \%\ \text{Outside Air}\ [T_{\text{Outside Air}} - T_{\text{Return Air}}] + T_{\text{Return Air}}$

$T_{\text{Mixed Air}} = 0.25\ [85 - 75] + 75$

$T_{\text{Mixed Air}} = 77.5°\ F$

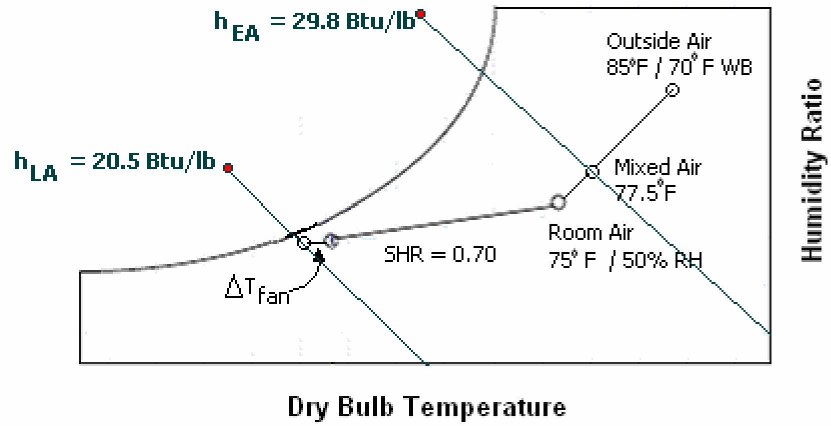


<u>Determine enthalpy for coil entering and leaving conditions</u>

Q coil = 4.5 x CFM x Δh

Q coil = 4.5 x 1425 x (29.8 - 20.5)

Q coil = 58,636 Btu/hr



Determine Chilled Water Flow

Q total = m* Cp * ΔT

For water at ρ = 62.34 lbm/ft^3 and Cp = 1.0

Q total = 500 x GPM x ΔT

GPM = Q coil / (500 x ΔT) = 58,590 / (500 x 10°F)

GPM = 11.7

Room Specific Humidity Differential (Δw)

$$\Delta\omega = \text{Latent heat} \times \frac{1}{\text{CFM}} \times \frac{1}{\rho} \times \frac{1}{h_w}$$

$$\Delta\omega = 13500 \times \frac{1}{1425} \times \frac{1}{0.075} \times \frac{1}{1059} \times \frac{1}{60}$$

$$\Delta\omega = 0.00198 \frac{lb_w}{lb_a}$$

Where

13500 = latent heat, Btu/hr

1425 = calculated air flow rate, cfm

0.075 = air density, lb/ft^3 (saturated air)

1059 = latent heat of vaporization at 60ºF, Btu/lb

60 = min/hr

STANDARD EQUATIONS IN AIR-CONDITIONING

The following formulae and factors are used in the air conditioning field:

Thermal Load

Btu = (lbs) * (sp. heat) * (ΔT)

Btu/hr = (lbs/hr) * (sp. heat) * (ΔT)

Btu/hr = (lbs/hr) * (hg - hf)

Mass –Volume relation

Lbs/hr (std. air) = (CFM) * (lbs/ft^3) * (60 min/hr)

= (CFM) * (0.075) * (60)

= (CFM) * 4.5

Sensible Heat

SH, Btu/hr (std. air) = (lbs/hr) * (sp. heat) * (ΔT)

= (CFM) * (4.5) *(0.24) * (ΔT)

= (CFM) * (1.08) * (Dt)

Air flow Rate

CFM = SH / [(1.08) * (room temperature - supplied air temperature)]

Latent Heat

LH, Btu/hr (std. air) = (lbs/hr) * (hg -hf) (grains of moisture diff. /7,000 grains/lb)

= (CFM) * (4.5) * (1,054) (grains diff. /7,000)

= (CFM) * (0.68) * (grains diff.)

Where (hg -hf) = 1,054 Btu/lb represents the heat of vaporization at 70°F. Variation in value for different conditions will be small.

Chilled water flow Rate

Lbs/hr (water) = (gpm) * (lbs/gal) * (min/hr)

= (gpm) * (8.33) * (60)

= (gpm) * (500)

Fan Power

HP air = [(CFM) * (DP)] / [(6,350) * (fan efficiency)]

Pump Power

HP water = [(gpm) * (DP)] / [(3,960) * (pump efficiency)]

Where

Dt = temperature difference

DP = pressure difference

SECTION -2 AIR HANDLING SYSTEM DESIGN CONFIGURATIONS

Air-conditioning practice includes five basic AHU arrangements, although there are many variations on these basic concepts.

a. Single Zone

b. Multi-zone

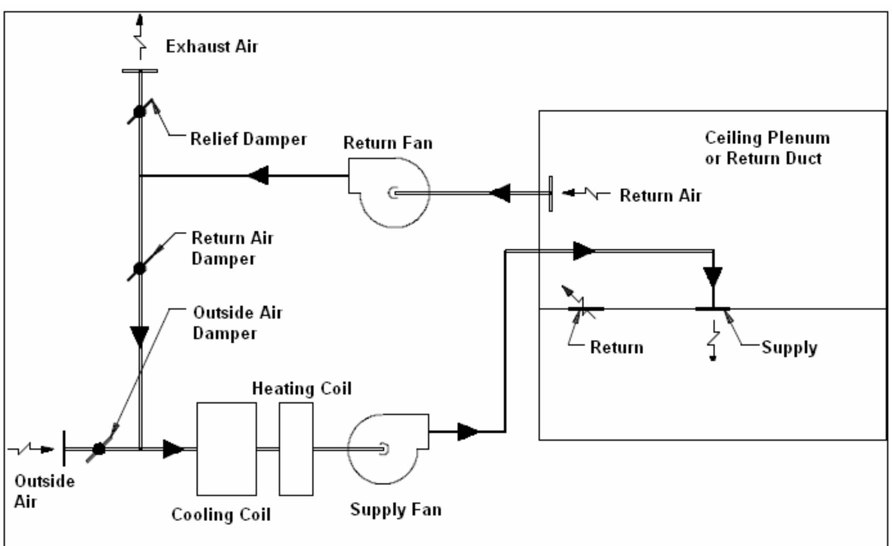
c. Double-Duct

d. Reheat Systems

e. Variable Volume (VAV)

Single-zone systems

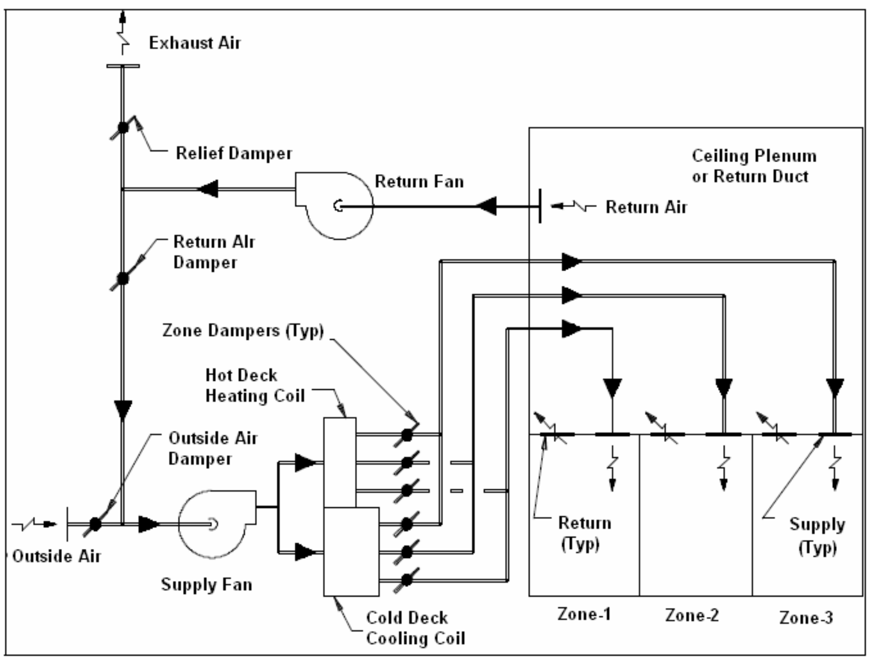
Single-zone systems serve just one temperature control zone and are the simplest type of all air systems. For this type of system to work properly, the load must be uniform all through the space, or else there may be a large temperature variation. Single -zone systems are, in most cases, controlled by varying the quantity of chilled water or refrigerant, adding reheat, adjusting face or bypass dampers, or a combination of these. If a close control of the humidity is required while in the cooling mode, a reheat system must be used. Figure below is the schematic for a typical single -zone central air handling system.



TYPICAL SINGLE ZONE AIR HANDLING SYSTEM

Multi-zone systems

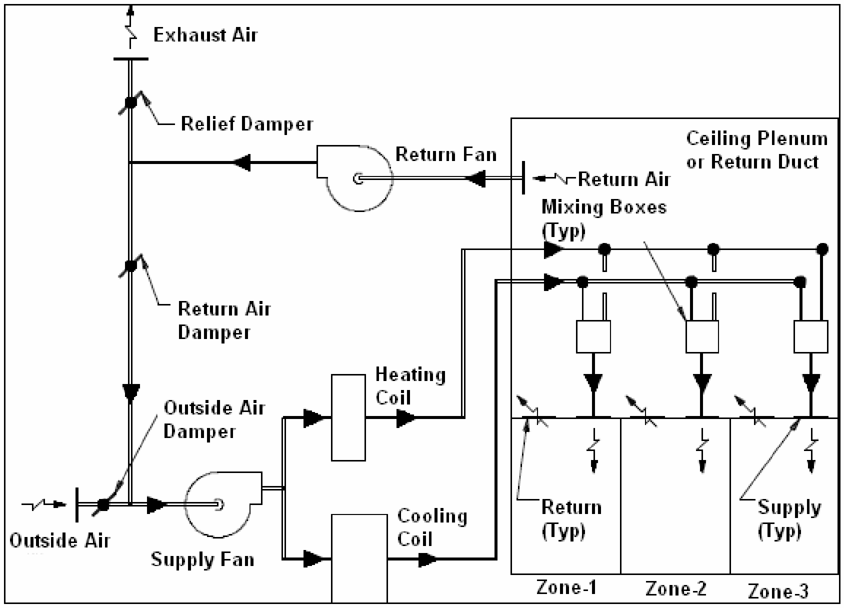
Multi-zone systems are used to serve a small number of zones with just one central air handling unit. The air handling unit for multi-zone systems is made up of heating and cooling coils in parallel to get a hot deck and a cold deck. For the lowest energy use, hot and cold deck temperatures are, as a rule, automatically changed to meet the maximum zone heating (hot deck) and cooling (cold deck) needs. Zone thermostats control mixing dampers to give each zone the right supply temperature. A typical multi-zone air handling system is shown on figure below.



TYPICAL MULTI ZONE AIR HANDLING SYSTEM

Dual-duct systems

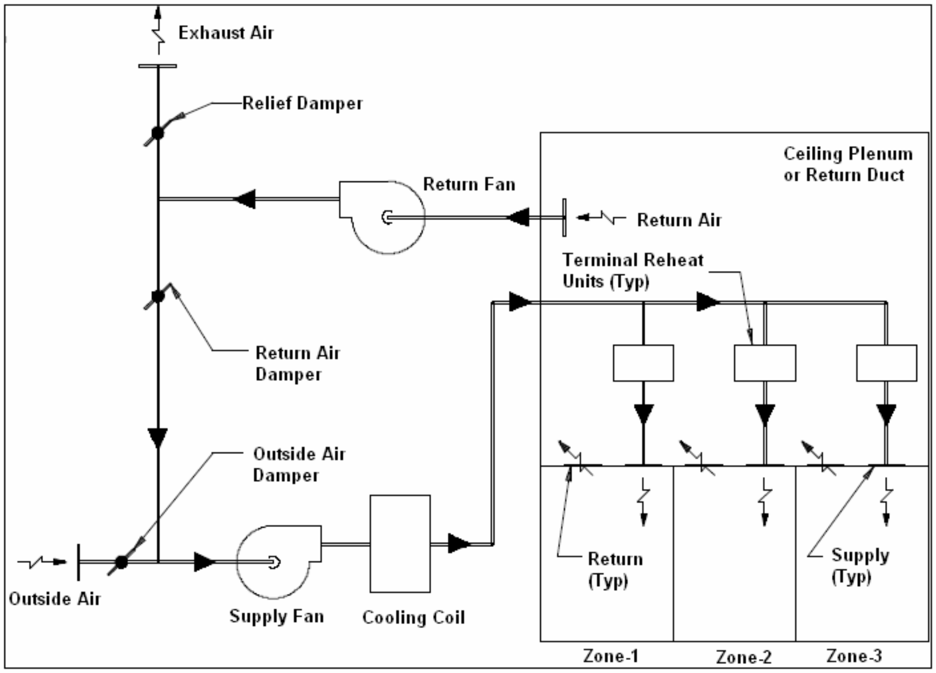
Dual-duct systems are much like multi-zone systems, but instead of mixing the hot and cold air at the air handling unit, the hot and cold air are both brought by ducts to each zone where they are then mixed to meet the needs of the zone. It is common for dual-duct systems to use high-pressure air distribution systems with the pressure reduced in the mixing box at each zone. A simple dual-duct system is shown on figure below.



BASIC DUAL DUCT AIR HANDLING SYSTEM

Reheat systems

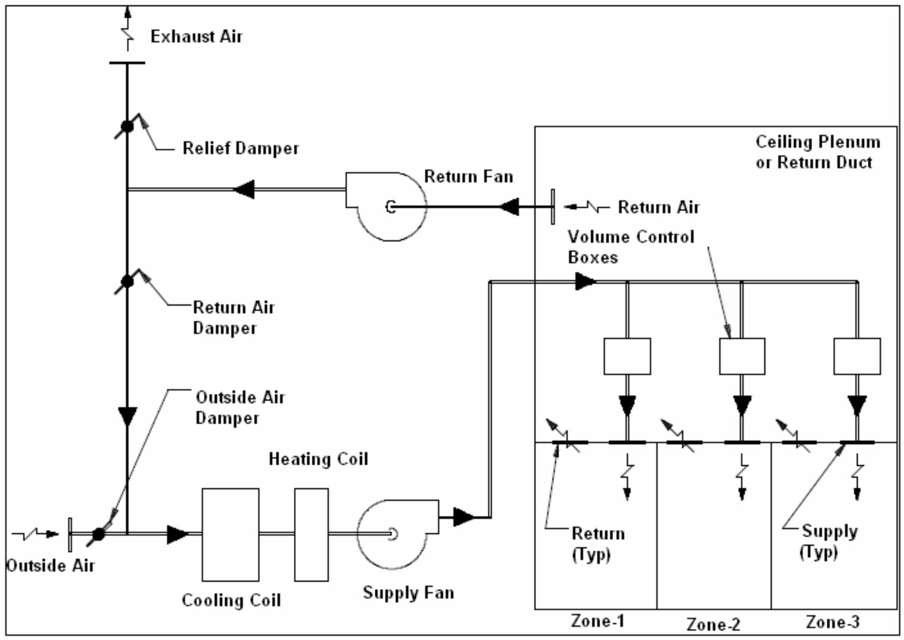
Reheat systems supply cool air from a central air handler as required to meet the maximum cooling load in each zone. Each zone has a heater in its duct that reheats the supply air as needed to maintain space temperatures. Reheat systems are quite energy-inefficient and have become rare in new buildings as the cost of energy has gone up. Most reheat systems are constant volume, thus the reduction of air volume to each zone as the cooling load gets less will keep down the use of the reheat units and reduce energy consumption significantly. Energy may also be saved by automatically resetting the temperature of the cold air, allowing it to rise as the temperature of the outside air falls. More energy savings may be found through the recovery of the refrigeration system's rejected heat and the use of this heat to reheat the air. An example of a reheat system that uses terminal reheat units is shown on figure below.



TYPICAL AIR HANDLING SYSTEM WITH REHEAT

Variable air volume (VAV) systems

The one feature that is common to all variable air volume systems is they change the volume of air in response to a change in load, rather than a change in the supply air temperature. Figure below shows a variable air volume, single-duct, multi-zone system. Variable air volume systems may change the volume of the whole airflow and/or the volume of each individual zone. Total system airflow may be varied by the use of inlet vanes, discharge dampers, speed control, and variable pitch blades. Zone airflow may be modulated in coordination with total system flow modulation or it may be varied by passing the excess flow right to the return air system with no variation in total system flow. Note that fan horsepower savings, which in most cases are found with variable air volume systems, are obtained from systems which modulate total system volume in response to zone volume modulation.



VARIABLE AIR VOLUME AIR HANDLING SYSTEM

AHU SIZE AND LOCATION

Air-handling units shall preferably be sized for duty not exceeding 25,000 CFM. Smaller units are encouraged to facilitate flexible zone control, particularly for spaces that involve off-hour or high-load operating conditions. To the extent possible, “plug-n-play” AHU configurations should be considered, facilitating easy future adaptations to space-load changes.

AHU location is an important consideration in light of first cost savings and operational savings. The AHUs located at the middle of the building zone will use less duct material and will have lesser pressure drop. This will provide cut in power costs and benefit in recurring energy savings. Also the units located indoors will ensure longevity. Units located outdoors will deteriorate at a faster rate.

Equipment longevity can also be ensured by configuring the air handling unit assembly in a manner that promotes ongoing maintenance and provides enough flexibility to allow the system to meet the changing needs of the building over time. Keep a clear gap at least equal to the width of the AHU on the filter removal side to plan for frequent maintenance.

SECTION -3 AIR HANDLING SYSTEM MAJOR COMPONENTS

A standard modular AHU consists of 4 main sections: 1) fan section, 2) coil section, 3) mixing section and 4) filter section.

1) Fan / motor section will consist of forward curved/backward curved or airfoil fan single width single inlet (SWSI) fan driven by open drip-proof (ODP), totally enclosed fan cooled (TEFC) or explosion proof super-E efficiency motors. The drive package will include V-belt designed for a minimum 1.3 service factor.

2) Coil section will consist of chilled water/direct expansion cooling coils and/or hot water/steam heating coils. Drain pan is considered below the cooling coils along with stainless steel / galvanized drain line of minimum 1-1/4 inch diameter.

3) Mixing box section will consist of dampers for outdoor air, return air and exhaust control. The face and bypass dampers are also included as an option.

4) Filters section will consist of totally sealed pre-filters: pleated media type (flat or angled arrangement), cartridge, bag and/or HEPA filters.

Usually manufactures provide various auxiliary attachments options such as humidity control equipment (spray type/steam pan), attenuator, blender, economizer section, return air fan/plenum section, inspection chamber, distribution ducts, and face & bypass damper etc. For horizontal AHU, the equipment length increases with each extra option.

Addressed below are the details of some standard components:

AIR HANDLING UNIT FANS

The fan is the heart of the air handling system. These are used to move air with a fan impeller that makes use of centrifugal or propeller action. Fans can be put into two general classifications: 1) centrifugal fans and 2) axial fans.

Centrifugal fans have the flow directed radially outward from the fan wheel while axial fan flow is moved parallel to the shaft. All fans have three basic parts: an impeller, motor, and housing. The impeller is the part of the fan that moves the air. In order for an impeller to move air, it must rotate. This is done by power from the motor. Housings are made to fit the individual fan types. Materials used in fan construction are generally steel. They are also built of aluminum and can be made of special materials, such as stainless steel or epoxy-bound fiberglass. Fans can be coated with compounds that are especially suited to the many kinds of corrosive atmospheres in which they must work. In some cases, spark-proof construction is required.

Small fans, especially those whose blades or wheel turn at higher speeds, are equipped with direct-connected motors. For larger size fans, and those that operate at lower speeds, V-belt drives are used. Belt drives have the advantage of giving a lower fan speed than the motor speed and have a built-in shock-absorbing ability.

In some air handling systems, there is a need to change the volume of air that is produced by the fan. Where such a change is made infrequently, the pulley or sheave on the drive motor or on the fan may be changed to vary the speed of the fan. Dampers may be placed in the duct system to vary the air volume. Variable -speed pulleys and drives, such as electric or fluid drives, may be used to change the fan speed. Two-speed motors and variable fan inlet vanes may also be used to control the airflow rate.

Centrifugal Fans

A centrifugal fan is built with a wheel that is mounted on a horizontal shaft and turns in housing. Air enters near the axis of the wheel and is discharged through the housing outlet. Air may enter the fan wheel at one or at both ends of the wheel's axis. The fans that are in use for air- conditioning, heating, and ventilating systems normally do not exceed 10 inches of water static pressure. The main feature that distinguishes one type of centrifugal fan from another is the curvature and the inclination (slope) of the fan wheel blades. The slope largely determines the operating characteristics of the fan. The three principal types of blades are the forward-curved blade, the radial blade, and the backward inclined blade.

1. The forward-curved blade fans are used primarily for the low-pressure heating, ventilating, and air-conditioning application. Domestic furnaces, low-pressure central station air handling units, and packaged air-conditioning units, such as window and rooftop air-conditioning units, use this type of fan.

2. In the radial blade fan wheel, the tip of the blade projects straight out from the fan shaft. The radial blade fan can work at a higher pressure than either the forward-curved or the backward-inclined blade fans. However, to move the same amount of air as the other two types, the radial fan wheel requires more horsepower. Due to its low efficiency, the radial blade fan generally is not found in heating, ventilating, and air-conditioning (HVAC) applications. It is used more for material handling applications, since the wheels are of simple construction and they can be fixed in the field.

3. In the backward-inclined blade fan wheel, the tip of the blade is inclined backwards away from the direction of the rotation of the fan wheel. This lets the backward-inclined fan move air at higher pressures than the forward-curved fan. It is more efficient (uses less horsepower) for most air volume and pressure ranges than the forward-curved blade fan. The backward-inclined blade wheel is also built with blades that are made in an airfoil shape. This wheel is the most efficient of all types and is the quietest at high static pressure. They are normally used for medium- and high-pressure systems, although they are, at times, used in low-pressure systems.

Axial Fans

An axial fan may consist of a fan wheel mounted on a motor shaft or it may have one, or in some cases two, fan wheel(s) mounted on a shaft and confined inside a housing or tube. Axial fans use either a direct drive or a belt drive. The three main types of axial fans are propeller fans, tube-axial fans, and vane-axial fans. Axial fans do not develop their static pressure by centrifugal force. The static pressure is gained from the change in velocity of the air when it passes through the fan wheel.

1. A propeller fan consists of a multi-blade impeller within an inlet ring or plate. Propeller fans are low-pressure, high-capacity units built either with the blades mounted on the shaft of an electric motor or a shaft for V-belt drive.

2. A tube-axial fan consists of an axial flow wheel within a cylinder or tube. Tube-axial fans may be used on low- and medium-pressure systems.

3. Vane-axial fans are tube-axial fans with guide vanes that straighten out the axial spiral airflow. Vane-axial fans can be used in low-, medium-, and high-pressure systems.

Good Engineering Tips

1. Centrifugal fans are able to move more air at higher pressures and with less noise than axial fans and are most commonly used in air handling units.

2. The centrifugal fan with a backward-curved impeller is the predominant fan used in "built-up" type air conditioning units, while the forward-curved impeller centrifugal fan is used in "package" type air handling units.

3. Fans should be selected on the basis of required horsepower as well as sound power level ratings at full load and at part load conditions.

4. Fan motors shall be sized so they do not run at overload anywhere on their operating curve. Fan operating characteristics must be checked for the entire range of flow conditions, particularly for forward curved fans.

5. Fan drives shall be selected for a 1.5 service factor and fan shafts should be selected to operate below the first critical speed.

6. Thrust arresters should be designed for horizontal discharge fans operating at high static pressure.

7. All fans shall bear the AMCA seal and performance shall be based on tests made in accordance with AMCA Standard 210.

COOLING & HEATING COILS

Coils are used in air conditioning systems either to heat or cool the air. The typical coil consists of various rows deep of finned tubing. The number of fins per inch varies from 3 to 14. The greater the number of fins per inch and row's depth that a coil contains, the greater its heat transfer rate will be. An increase in heat transfer surface results in an increase in heat transfer efficiency and also in increased airflow resistance which will, in turn, require increased fan horsepower.

The cooling process should always be plotted on a psychrometric chart to be sure that desired psychrometric changes are feasible. When selecting a coil, it should be remembered that if the required leaving wet bulb temperature is attained, the total load is satisfied and vice versa. Also, when the required leaving dry bulb temperature is met, the sensible load requirement is satisfied.

A cooling coil must meet both the total and sensible load requirement in order to achieve the conditions desired in the space to be cooled. Normally, the total load capacity is checked first. When the sensible to total load ratio (S/T ratio) is low, the coil selection is normally controlled by the total load even though the sensible cooling capacity may exceed the requirement. In some cases, if the leaving dry bulb temperature is too low, reheat may be required.

When the S/T ratio is high, the coil selection is normally controlled by the sensible cooling even though the total capacity may exceed that required. If the total capacity far exceeds the requirement, a recheck on the system should be made to be sure sufficient system capacity is available.

The cooling medium used is chilled water, brine, or refrigerant in a direct expansion-type coil. All are fabricated by expanding copper tube to aluminum collar spaced fins at a suitable pitch to meet key design performance requirements. The fins are mechanically bonded, providing excellent heat transfer characteristics, with staggered tubes in the direction of the airflow. Special materials and finishes, such as copper fins and electro tinning, can be provided to meet particular specification requirements.

DX Coil Chilled Water Coil

Chilled water or glycol coils

Refrigeration equipment generates cold water or glycol, which is then pumped to coils located in the air handling units to cool and dehumidify the air stream.

The coil framework is a rigid galvanized formed section and the coil tubes are brazed to copper headers with flow and return screwed or flanged connections. The tubes may be staggered or placed in-line with respect to the airflow. The individual tubes in a coil are usually interconnected by return bends (U-bends) to form the serpentine arrangement of multi-pass tube circuits. The resulting velocity through the cooling coil is dictated by the air quantity, coil size, available space, and the coil load. Most manufacturers offer a wide choice of fin spacing varying from 6 to 14 fins per inch and varying number tube rows, usually two to eight in a single casing. Coils are also fitted with an air vent and drain plug as standard.

Capacity control is typically achieved by modulating the flow of water through the coil or by bypassing air around the coil. Generally, modulating the water/glycol flow is the most desirable approach since it has the potential to save chilled water pumping energy. Where dehumidification requirements lead to the need for reheat, mixed air or return air can bypass around the cooling coil for discharge temperature control while the chilled water coil valve is controlled to provide adequate dehumidification.

Direct expansion refrigeration

The general construction of DX coils is similar to that of the chilled water coils described above except that the cooling media is direct refrigerant such as Freon. DX is short for "**D**irect e**X**pansion". The framework is a rigid galvanized formed section with the coil interlaced to suit compressor load requirements, while the liquid lines are brazed to distributors. Compressors and condensers included as a part of the air handling package or located remotely, move the refrigerant through the piping and coil system and allow the heat absorbed in the air handling unit to be rejected to atmosphere.

If choosing a single-zone direct-expansion (DX) system, be sure to properly size the system for good dehumidification performance and consider including refrigerant sub-cooling to increase the latent capacity. Capacity control is achieved on the refrigeration side by an expansion device at the coil coupled with compressor unloading and hot gas bypass. Compressor unloading systems are generally step devices, which limit capacity modulation. At low load conditions, the compressors will cycle and unconditioned air will pass through the system during the off cycle, which can cause problems. Hot gas bypass can be used to maintain compressor operation continuously regardless of load, but this is an energy intensive solution since it maintains an active cooling coil at the expense of false loading the compressor. Face and bypass dampers are sometimes encountered for air-side capacity control, but they must be used with extreme caution since the low air flow rates that occur across the coil when the dampers are in bypass can cause severe problems in the refrigeration circuit.

HEATING

There are three different terms often used in connection with AHU systems. These are 1) preheat, 2) reheat and 3) heaters. Frequently, any of these heating devices is either a coil that uses steam, hot water, or electricity as an energy source. In some situations, the heating element is a fossil fuel fired furnace or an energy recovery coil.

From a psychrometric standpoint, not all heating elements are the same. The specific function they provide depends on:

a. The location of the heating element in the system relative to other components.

b. The manner in which the coil is connected to its supply of heating energy (to prevent freezing).

c. The manner in which the coil is controlled.

Preheat

The function of preheat is to offset heating requirements associated with ventilation and make-up air; protect the system and building from sub-freezing air. Preheat is required by an air handling system if it will see operating conditions that will result in supply temperatures that are lower than required to maintain the design conditions at the load served and will subject the system, its components, and/or the loads served to air at subfreezing temperatures and thereby cause damage by freezing.

True preheat applications are typically found on 100% outdoor air systems and on systems with high outdoor air fractions relative to their total supply flow. Unless they are located in an extreme environment, most air handling systems serving office environments will seldom require preheat if their minimum outside air percentage is 20-30% of the supply flow rate and good mixing is achieved. Preheat is generally located as first element after the intake for 100% outdoor air systems and first after the mixing box for re-circulating systems. Occasionally, in a moderate environment, preheat is required due to high ventilation rates.

Reheat

The function of reheat is to offset unnecessary sensible cooling that was done to provide dehumidification to meet the space design requirement. In contrast to preheat elements, reheat elements that are located downstream of the air handling system's cooling coils. The summer time cooling coil discharge temperature is typically set based on the amount of dehumidification required to achieve adequate dehumidification for the occupied zone. The required volume of air, when supplied at this temperature, can overcool the occupied zone under some load conditions. Typically, overcooling can occur in situations where the flow to the occupied zone is set based on air change requirements, ventilation requirements, or make-up air requirements rather than being set by space sensible gains and temperature requirements. Clean rooms and hospital surgeries are good examples of applications where this can occur due to the high air change rates associated with maintaining cleanliness. In these situations, the reheat coil is used to warm the discharge air off the cooling coil as necessary to prevent overcooling of the space while still maintaining the required air flow and space humidity condition. Reheat is an energy-intensive process since it is intentional simultaneous heating and cooling.

Some system designs provide the reheat function at the zone location rather than at the central system location. This allows the reheat process to be limited to only the areas requiring it due to the specific needs of the zone while optimizing the central system supply temperature based on the needs of the critical zone.

Heating

The function of heating is to offset space sensible losses through the building envelope associated with the rate of heat transfer exceeding the rate of heat gain. It is essentially for perimeter heating and infiltration loads. The location of heating element is not critical but first after the mixing plenum provides some measure of protection for the rest of the system. Heating control system requires sequence with other system functions to prevent simultaneous heating and cooling and to prevent using heating when the system is not on minimum outdoor air. The freezestat must be upstream since the coil would not typically be configured for subfreezing air.

Common approaches to heating include:

Hot water coils

Hot water coils are suitable for low, medium and high pressure applications. They are fabricated by expanding copper tube to aluminum collar fins spaced at a suitable pitch. These fins, which are staggered in the direction of the airflow, are mechanically bonded to deliver excellent heat transfer characteristics and are available in special materials and finishes such as copper and electro tinning to meet particular specifications.
The construction is similar to as described for chilled water coils. However, the most common circuiting arrangement is often called single -row serpentine. With this

arrangement, all tubes in each coil row are supplied with an equal amount of water through a manifold, commonly called the coil header.

Steam coils

Coils generally consist of a steam header and a condensate header joined by finned tubes. The headers may be both at a side of the unit, with U-bends between them, or sometimes an internal steam tube is used to carry the steam to the remote end of an outer finned tube. Vertical headers may be used with horizontal finned tubes, or sometimes horizontal headers at the bottom of the unit supply vertical finned tubes. Copper and aluminum are the most common materials used in the fabrication of low-pressure steam coils. Low-pressure steam coils are usually designed to operate up to 150 to 200 psig. For pressures higher than 200 psig, tube materials, such as red brass, Admiralty, or Cupro-Nickel, are used. Tubing made of steel or various copper alloys, such as Cupro-Nickel, are used in applications where corrosive materials or chemicals might attack the coils from either inside or outside.

Electrical Heaters

Designed for black heat operation, the electric heater elements are typically manufactured from 80/20 nickel/chrome resistance wire centered in a protected metal tube packed with magnesium oxide. While element tubes are of copper-coated mild steel tubular or finned construction as standard, stainless steel or incoloy are also common. The electric heat options are usually delivered complete, with safety cut-out and manual reset facilities, complying with IP42 and NEMA specification. Thyristor/SCR control option is also available with energize and de-energize cycling capabilities.

Good Engineering Tips

1. Direct expansion-type coils are used on small systems when a chilled-water system is not economical. Chilled water is used on all other systems when the air temperature required is above 50°F. When the air temperature required is less than 50°F, a brine solution is used as the cooling medium because of its exposure to subfreezing temperatures in the refrigeration machine.

2. Dehumidification cooling coils shall be selected at or below 500 fpm to minimize moisture carryover. Limit dry (sensible) cooling coils to 600 fpm air face velocity. Heating coils shall be selected at or below 750 fpm.

3. Cooling coil manufacturers usually provides computer programs for coil selection. To improve coil heat transfer performance and reduce air-side pressure drop, select coils preferably with a low face velocity of 250 to 300 fpm instead of 500 fpm, and low approach temperatures of 5°F to 8°F instead of 10°F to 15°F.

4. Individual finned tube coils should generally be between six and eight rows with at least 2.1 mm between fins (12 fins per inch) to ensure that the coils can be effectively and efficiently cleaned.

5. Equipment and other obstructions in the air stream shall be located sufficiently downstream of the coil so that it will not come in contact with the water droplet carryover.

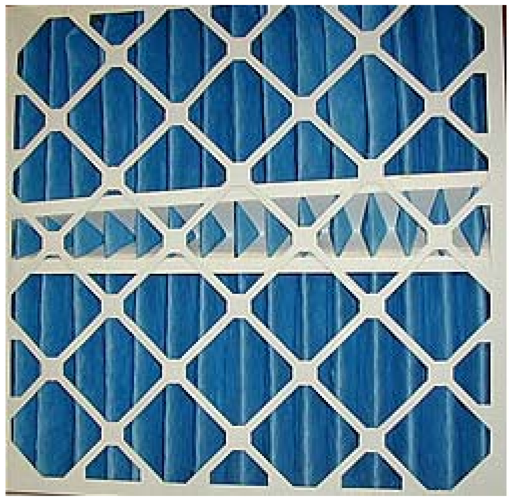
6. Water velocity in the tubes of approximately 3 to 6 FPS is desirable to attain high heat transfer rates with a reasonable water pressure drop. Water velocity above 8 FPS may cause erosion in copper tube coils.

7. Cooling coils shall normally be selected to have a finned length of three to four times the fin height for economy. Cooling coils should not normally exceed 54" fin height as the condensate draining from the top portion of the coil tends to load up on the lower portion of the coil and a significant reduction in airflow and performance may result. Where the fin height exceeds 54", two or more coils may be stacked one above the other with intermediate drain pan.

8. All coils shall perform in accordance with requirements of ARI 410.

AIR FILTERS

The air that is drawn into an AHU is contaminated to some degree. Such contamination may be as solid particles, liquids, fumes, smoke, or bacteria. Particle diameters range from molecular size up to 5,000 microns and the concentrations as high as 400 grains per 1,000 cubic feet may be encountered. However, air conditioning applications usually involve the removal of particles no smaller than 0.1 micron in diameter and as large as 200 microns.



Some of the filtration processes are described below:

1. Viscous impingement filters are of the panel or roll type with a viscous (tacky) coating on the media to hold the particles to the media. The coating is called an adhesive. Viscous impingement filters are made to trap large dust particles from the air stream. Most air-conditioning systems have this type of filter. There are four types of these filters: throwaway, cleanable, automatic renewable media, and automatic self-cleaning media.

2. Dry media filters, as the name suggests, do not have the tacky coating as applicable to viscous impingement filters. The dry media filters take out particles from the air by interception and straining. Interception means to filter out particles using the natural forces of attraction between molecules. Straining means to take out particles that are too large to pass through the openings between the fibers.

3. Activated carbon - A filter made of activated carbon will get rid of solid particles, as well as odor-causing gases and bacteria from the air stream. It is possible to clean and

reuse the carbon filters. However, this is best done by the manufacturers, who will take out the carbon and process it to be used again.

4. Electrical precipitator - The inability of standard dry or viscous type filters to take out fine dust particles from airstreams has led to the development of the electrical precipitator. The precipitation method consists of giving an electrical charge to each dust particle in the air stream by passing the air between electrodes and then collecting the dust on parallel plates as the air flows between the plates.

5. HEPA filters - HEPA stands for High Efficiency Particulate Air. The HEPA filters work on diffusion principle to remove particulate matter and are extremely important for clean room applications. These filter particles as small as 0.3 µm (microns) with a 99.97% minimum particle-collective efficiency. This is remarkable considering that the outside air we breathe may contain up to 5 million suspended particles of dust, smog, and pollen in one cubic foot. These filters typically use glass fiber media and are available in thicknesses of 6" and 12". These have pressure drop of 1 inch- H_2O when clean and generally need to be replaced when the pressure drop exceeds 2 inch- H_2O.

6. ULPA filters - ULPA stands for Ultra Low Particulate Air. Growing market demand from advanced science and technology led to development of ULPA filters which provide a minimum of 99.999% efficiency (0.001% maximum penetration) on 0.3 micron particles for achieving better cleanliness classes and cleaner working environments. These are used for ultra-clean rooms, where contamination levels have to be controlled at levels better than that which can be achieved with conventional HEPA filters.

7. CBR (chemical-biological-radiological) - These filters remove chemical, biological and radioactive particles from the intake air. Many blast-protected air intake systems allow intake air to bypass the CBR filter elements when the blast protection system is not in the blast mode of operation. While this prevents the performance of the CBR filters from being impaired by a buildup of dirt, the activated carbon filter element deteriorates from exposure to air over time and requires replacement on a regular basis, even if the filter bank has not been in service.

Air filters are rated in terms of arrestance, dust holding capacity and efficiency. *Arrestance* means a filter's ability to capture dust and describes how well an air filter removes larger particles such as dirt, lint, hair, and dust. The *dust holding capacity* of a filter is the amount by weight of standard dust that the filter will hold without exceeding the resistance 0.18 inch-H_2O for low-resistance filters, 0.50 inch-H_2O for medium-resistance filters and 1.0 inch-H_2O for high-resistance filters. Be aware that arrestance values may be high; even for low-efficiency filters, and do not adequately indicate the effectiveness of certain filters for chemical or biological protection. *Dust spot efficiency* measures a filter's ability to remove large particles, those that tend to soil building interiors. Dust holding capacity is a measure of the total amount of dust a filter is able to hold during a dust loading test.

Since large particles make up most of the weight in an air sample, a filter could remove a fairly high percentage of those particles while having no effect on the numerous small particles in the sample. Thus, filters with an arrestance of 90 percent have little application in clean rooms.

ASHRAE Standard 52.2-1999 measures the particle size efficiency (PSE). *Efficiency* measures the ability of the filter to remove the fine particles from an air stream by measuring the concentration of the material upstream and downstream of the device. If a supplier of filter only indicates efficiency as 95% or 99%, it does not really mean anything unless *it specifies the particle size range.*

Good Engineering Tips

1. All air-handling units shall have a disposable pre-filter and a final filter, rated in accordance with *ASHRAE Standard 52*. Pre-filters shall be 30 percent to 35 percent efficient. Final filters shall be with 85 percent efficiency capable of filtering down to 3.0 microns. Filter racks shall be designed to minimize the bypass of air around the filter media with a leakage not exceeding 0.5 percent.

2. Pre-filters shall be sized for face velocity of 350 -500 FPM and fine filters shall be sized at 250 FPM or lower.

3. Filter media shall be fabricated so that fibrous shedding does not exceed levels prescribed by *ASHRAE 52*. The filter housing and all air-handling components downstream shall not be internally lined with fibrous insulation. Double-wall construction or externally insulated sheet metal housing is acceptable.

4. The filter change-out pressure drop, not the initial clean filter rating, must be used in determining fan pressure requirements.

5. Differential pressure gauges and sensors shall be placed across each filter bank to allow quick and accurate assessment of filter dust loading as reflected by air-pressure loss through the filter and sensors shall be connected to building automation system.

6. The air filter that comes with the HVAC system is generally designed to keep the coils/blower/motor assembly clean. It is not designed for any thing else. Therefore.... when you add any other type of in-line air filter to the HVAC system, you must know the total system static pressure as to make sure the airflow is not affected by adding an air filter that is too restrictive. If, as we mentioned in the above paragraph, you add an air filter that increases the total system static to .6" or .7" you can reduce the airflow by 100 CFM to 200 CFM with a clean air filter. This additional static load increases as the air filter loads up, and decreases the airflow even further and decreases the overall efficiency of the HVAC system.

DEHUMIDIFIERS

Most AHU systems are intended to perform dehumidification by cooling the air below its dew point or by chemical means. The two most common types of dehumidification equipment are those which use refrigerant and some form of solid or liquid adsobents.

1. Refrigeration type dehumidifier is the most commonly used system where dehumidification is accomplished by lowering the air temperature below the dew point and letting some of the moisture condense out. Typically the moisture condenses on the outside of the refrigerant tubing.

2. Solid adsorbents are those which have the ability to make moisture cling to their surface. The products that are used the most are silica gel, activated alumina, and molecular sieve. These desiccant materials will take the water vapor from air or gas with physical or chemical change. The water vapor that they pick up can be released by passing hot dry air across the surface of the product. These products have submicroscopic cavities that hold the particles of adsorbed water vapor.

3. Liquid adsorbents - The systems that use a liquid absorbent to take the moisture from the air are designed to achieve close moisture control in the room that it is serving. These systems can be used as the only means of temperature/humidity control or can be used with a total air-conditioning system to give complete environmental control. The example of liquid adsorbent is lithium bromide.

Good Engineering Tips

1. The dehumidification using cooling systems is highly dependent upon the leaving temperature condition from the cooling element. The leaving air from the AHU must be cooled below the dew point temperature (typically below 52ºF) for effective dehumidification. Misapplied discharge temperature reset sequences can raise supply temperatures to the point where adequate dehumidification is not provided. This can occur if face and bypass dampers are used on the coiling coil without providing some means to control the cooling coil discharge temperature and ensure flow through the cooling coil to provide the desired dehumidification.

2. DX (direct expansion) systems with limited turn down capacity and no hot gas bypass capability can also have problems with dehumidification during the portion of the operating cycle when the compressor is off while the fan remains in operation. It can be particularly troublesome if the compressor is significantly oversized for the load it served, either by design or by the current load condition.

HUMIDIFIERS

Humidifiers are devices that add moisture to the airstream, thereby raising the relative humidity of the conditioned space. In most comfort air conditioning systems and in many industrial air conditioning systems, humidifying devices are commonly sparging steam or atomizing water directly into the airstream. There are several basic types of humidifiers:

1. Conventional steam humidifiers use an electrical or gas-fired heater to boil water and release steam into the air. Also known as pan type humidifiers in HVAC parlance.

2. Direct or live steam humidifiers inject steam directly into an air handling system or ductwork.

3. Ultrasonic devices vibrate a piezoelectric transducer at a very high frequency to create tiny water droplets instead of steam.

4. Specialized nozzles may be used to discharge a mist of atomized water and compressed air.

5. Moist porous pads may be used to enhance surface exchange between wet surface and air stream.

Pan type steam humidifiers are generally used for central air handling systems. These function best when deionized, demineralized or softened water is used. The humidifier must condition the steam to be completely dry and free of significant matter. It must respond immediately to control, provide precise output, and distribute steam as uniformly as possible into the air. Failure of the humidifier to provide conditioning, control and distribution characteristics will result in improper humidification.

Since the advent of energy conservation, the standards for comfort air conditioning systems have been reviewed and revised. One of the revisions eliminates the control of humidity as a comfort air conditioning system standard, since controlling humidity requires humidity requires additional energy year-round. In industrial air conditioning systems which employ humidity control, it is recommended that this need be reviewed and be reduced to the lowest degree the process will permit.

ELIMINATORS

When a cooling element is dehumidifying, it will tend to have a coating of water on the surfaces in contact with the source of cooling. Typically, these surfaces will also have air flowing past them. As a result, water droplets tend to be blown off of the element. The smaller droplets will evaporate and the larger droplets will fall from the air stream in a short distance. The condensate drain pan extends beyond the cooling element location to catch these drops as they fall out of the air stream. Where space constraints prevent an appropriate condensate drain pan extension, eliminators can be installed downstream of the coil to provide a surface to intercept and remove the droplets from the air stream.

If the accumulation of water on the surface becomes too heavy or the airflow velocity becomes too high, water can be carried past the drain pan or even blown back off of the eliminators, resulting in water damage and IAQ problems. Conditions that can lead to carry-over include:

1. Excessive coil face velocities - Generally, the rule of thumb is to keep coil face velocities below 500 fpm to prevent moisture carry-over.

2. Excessive dehumidification load - This condition can be created by coil entering conditions that are in excess of design or by operating the coil with a refrigerant temperature lower than design (usually reflected by chilled water that is colder than the design condition or a saturated evaporator pressure lower than design).

3. Poor condensate removal **-** If condensate is allowed to accumulate on the coil surface due to an inadequately arranged drainage system, then carry-over can occur. This typically happens a) with tall coils that are not provided with intermediate drain pans; b) if the drain lines from the intermediate pans become clogged and c) if the drain pans are not draining quickly enough.

CONDENSATE DRAIN PAN

Most cooling processes dehumidify the air, which can generate considerable mounts of condensate. The drain pan along with trap which collects and removes this condensate is most often inadequately described in specifications and contract documents. Key considerations to design include:

1. Drain pans shall be made of stainless steel, insulated and adequately sloped and trapped to assure drainage. Drains in draw-through configurations shall have traps with a depth and height differential between inlet and outlet equal to the design static pressure plus 1 inch minimum.

2. Drain pans need to extend far enough past the cooling coil to ensure that they collect all of the condensate, including droplets carried off of the coil element under all operating conditions. At least 6 inches extended drain pan is recommended.

3. Tall cooling coils or coils arranged one over other (stack) may require intermediate drain pans to prevent excessive carry-over from the lower portions of the cooling element under heavy dehumidification loads. The intermediate pan catches and removes condensate generated by the upper portions of the cooling element so that it does not have to flow over the lower portions of the cooling element to reach a drain pan. Minimizing the water flowing through the lower portions of the coil reduces the potential for carry-over and also reduces the air pressure drop.

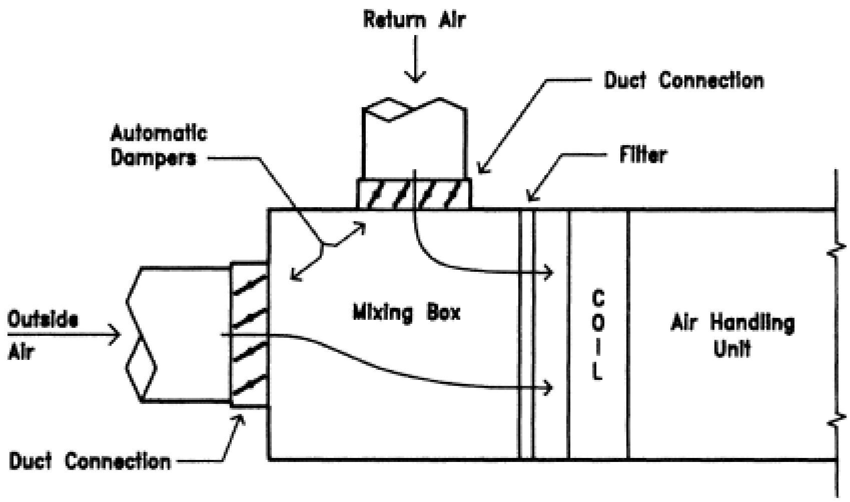
4. Drain pans should be insulated to prevent condensation problems associated with the temperature of the condensate itself.

5. For air handling equipment located above sensitive areas, consider providing a secondary drain pan and/or moisture alarms that will notify the operating staff of any overflow problems with the primary drain pan. A few extra dollars in first cost can have a quick pay back water damage from an overflowing drain pan is avoided.

6. If the trap on the condensate drain is intended to provide protection from infiltration of untreated air into the unit (an important consideration on some process applications) then trap primers may be required to keep the traps full when dehumidification is not occurring.

MIXING BOX

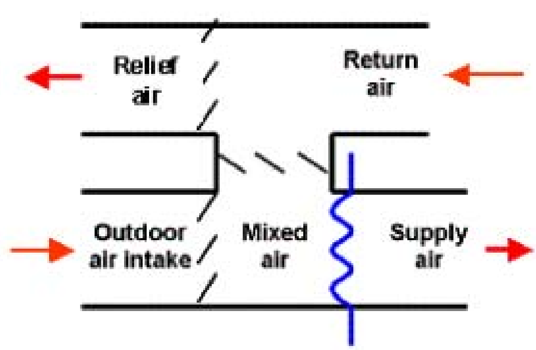
This is the portion of the AHU where return air is conveniently mixed with the outdoor air. The mixing section includes the outdoor air dampers and return air dampers.



Mixing Box for Return Air and Outside Air

A damper is provided for each air stream to balance the percentage of outside air versus return air. The dampers may be either parallel blade or opposed blade. Parallel blade damper sections are less expensive but opposed blade damper sections provide better control characteristics.

Plenum Boxes: Air-handling units shall be provided with plenum boxes where relief air is discharged from the air handling unit. Plenum boxes may also be used on the return side of the unit in lieu of a mixing box. Air-flow control dampers shall be mounted on the ductwork connecting to the plenum box.



Plenum Boxes

DAMPERS

The damper is used for three important functions in the air handling apparatus:

a. To control and mix outdoor and return air

b. To bypass heat transfer equipment

c. To control air quantities handled by the fan

Dampers are devices used to control or restrict the airflow. They fall primarily into three types: volume, backdraft, and fire dampers.

1. Volume dampers are devices used to vary the volume of air that passes through an air outlet, inlet, duct, fan, air handling unit, cooling tower, or condenser unit. They may vary the volume from 0 to 100 percent of capacity. Some volume dampers can be opened and closed by hand, while others are opened and closed by a pneumatic or an electric operator. The largest use of manual controlled volume dampers in cooling systems is for air balancing.

2. Backdraft dampers are devices used to limit the airflow within a duct to one direction and to stop airflow through a duct or opening when the fan is shut off. Backdraft dampers are opened automatically by the force of the airflow on the damper blades. They are closed automatically by a spring or weight counterbalance and by gravity. The counterbalances can be adjusted to allow the damper to pass the needed airflow.

3. Fire dampers are devices used to close off individual sections of a building during a fire. Fire dampers are normally installed where a duct passes through a wall, partition, floor, or ceiling which is specifically designed to provide fire resistance. If ducts pass through barriers having a fire rating of up to and not more than one hour of fire resistance and can be assumed to present no further fire hazard, there is no need for fire dampers. If the wall, partition, ceiling, or floor is required to have a fire resistance rating for more than one hour, a fire damper is then required to properly protect the opening where the ductwork penetrates the wall. Fire damper blades are held open by a fusible link (replaceable) during normal operation of the building. If a fire occurs, the fusible link melts and the damper blades close automatically. For a cooling system to operate properly, all fire dampers must be open all the way. Broken or damaged fusible links should always be changed, and fire dampers should never be wired open. Break-away type connections should be used to connect the ductwork to the fire dampers; solid connections should never be used.

4. Smoke dampers are used for either smoke containment or for smoke control. The damper is basically the same as a volume damper, except the damper is classified and listed in accordance with Underwriters Laboratories, Inc. (UL) 555S, UL Standard for Safety Smoke Dampers Fourth Edition (1999). The damper is a two-position damper, i.e., the damper is either open or closed depending upon the control requirements. The damper usually has low leakage characteristics.

An opposed blade double acting damper is used where control of air flow is required. This arrangement is superior since the air flow is throttled more or less in proportion to the blade position, whereas the single action type damper tends to divert the air and does little or no throttling until the blades are nearly closed.

LOUVERS

These are the elements which allow air to be drawn in from outside or discharged to outside, without allowing rain into the building. Louvers are shaped such that rain falling or being blown onto the louver is captured and channeled back to outside. Louvers may be incorporated into the side of a building, or they can take the form of free-standing penthouse arrangements. Some louver designs utilize blades constructed and arranged to decrease sound transmission.

Each louver should be complete with a bird mesh to avoid small birds or large insects from being drawn into the ventilation system. Variations include sand filters for desert areas and double bank louvers for particularly exposed areas.

An important louver parameter that is common to many of the louver performance specifications is louver free area. The free area of a louver is the area actually available for the passage of air. This should not be confused with louver face area, which is the overall cross sectional area of the louver perpendicular to the direction of airflow. The referenced Air Movement and Control Association (AMCA) standards indicate how free area can be calculated based on dimensions taken from the louver. Some manufacturer's state performance data in terms of free area velocity while others state performance in terms of face area velocity, so it is important to determine which velocity is being used when documenting louver performance criteria.

As indicated previously, one of the primary functions of a louver is to minimize the entry of rain water into an air handling system. In addition to being designed to resist penetration by water and channel water away from the interior of the system, louvers must be applied properly to achieve success in this area. As a general rule, face velocities should be 700 fpm or less to prevent water induction. It is also important to understand that a louver that is designed and tested to AMCA standards may still allow some water penetration. The test associated with the AMCA standard is designed to define the 'point of beginning water penetration', the louver face velocity at which water begins to make its way through the louver. This provides a consistent basis for comparison of different louver designs, but does not provide quantitative data on how much water can be expected to penetrate a louver under actual service conditions. For instance, the AMCA standard does not address driving rain, and real rainfall rates can exceed test standards. The design of the intake compartment should assume some rain penetration will occur, so the use moisture tolerant materials and including drainage is advisable.

The moisture droplets associated with heavy fog can penetrate louvers even when they have been properly sized and selected to minimize rain penetration. This moisture often becomes trapped in the first filter section in the system and can cause problems there unless the filters are selected and installed to minimize this complication. Systems that handle 100% outdoor air are particularly prone to this problem.

Problems encountered with louvers include louvers obstructed with leaves etc., supply and extract louvers which are located too close together allowing short-circuiting of stale air, or intake louvers located adjacent to areas in which vehicles run, allowing exhaust gases to be drawn into the building.

VALVES

Valves installed in the air handling system are to control water flow to the coils and to isolate equipment for ease of operation and maintenance. There are several different physical types of valves; globe valves, ball valves and butterfly valves are all commonly used in the HVAC industry.

1. ***Butterfly valves*** are most often used on larger pipes, typically 2½ to 12 inch and larger. They are often the only choice for controlling large-pipe central plant applications such as chillers, boilers, cooling towers, and thermal storage systems. A butterfly valve controls flow by rotating a disc within the valve body 90 deg from open to close. A key consideration when using butterfly valves for proportional control is their vulnerability to an extreme pressure drop in some applications. Unbalanced forces on the disc during high drops can cause oscillations, poor control, and/or damage to the linkage and actuator, even when the critical flow point is not reached. As a result, butterfly valves must be sized and selected using conservative pressure drop criteria.

2. **A globe valve** uses a contoured (throttling) plug that extends through the center of the seat to provide precise control for modulating service. The valve can be either push-down-to-close or push-down-to-open. Globe valves have strengths that make them popular for use with HVAC controls. First, they provide tight shutoff and precise control over a broad range of conditions. Second, they are available in many pipe sizes, typically from 1/2 to 6 in. Third, they can handle a wide range of Cv capacities (from <1 to 400), flow characteristics, temperature, and pressure requirements. Fourth, they are available in 2-way straight, or angle configurations and in 3-way mixing and diverting designs.

3. **Ball valves** provide tight close-off and are available in two- and three-way configurations. Two-way ball valves have equal percentage flow control characteristics, and the flow can be in either direction. Three-way models have linear flow control characteristics and can be used in diverting or mixing service, with some restrictions. When good but not precise control is necessary, motorized ball valves can be used instead of globe valves in many applications. These include heat exchangers; air handling unit heating/cooling water coils and steam heating coils; preheat coils; humidifiers; and unitary equipment such as reheat coils, unit heaters, convectors, radiant panels, fan-coil units, and unit ventilators. Lower cost is a major advantage of a ball valve; it might cost only half as much as a comparably sized linear globe valve or other valve type.

Globe valve or Ball valve

While motorized ball valves can be used in most common non-central plant HVAC control applications, they cannot always replace globe valves. Here are other factors to consider:

- Globe valves should be used when precision control is a higher priority than cost.
- In a modulating control loop, 1/2 in. ball valves with very small reduced ports are most suitable for two-position control due to the lack of a "throttling" ability of the small port.
- If noise is a primary concern, a globe valve may be a better choice.

- For low-flow modulating control at Cv <1, globe or zone valves are suggested.

CONTROL VALVES

Control valves are used to maintain space temperature conditions by altering the flow of steam, water, gas, and other fluids within an air-conditioning system. Valves can be two-position or modulating 3-port configuration.

Two-way valves throttle flow while three- way divert or mix flow. Two-way valves have two ports and are used to control the flow in variable flow systems. Three-way valves have three ports and can be piped for by-pass application either in mixing or diverting service.

There are two basic arrangements for three-way valves: mixing valves (two inlets, one outlet) and diverting valves (one inlet, two outlets). The type of three-way valve selected will determine its location in the system.

1. **Mixing Valves-** A three-port valve with two inlet flows and one common outlet flow is defined as a mixing valve, and it provides a variable temperature outlet at a constant flow rate. A three-port motorized valve can be used to MIX, in varying proportions, two flows of different temperatures while maintaining a constant rate of flow in the common outlet port. A Mixing Valve is used normally for radiator circuits.

2. **Diverting Valves-** A three-port valve may also be used to DIVERT a common flow in varying proportions. The valve will have one inlet and two outlets and provides a constant temperature and variable flow rate. A diverting valve is used normally for circuits with convective heat transfer such as; heat exchangers, primary coil in indirect cylinder, heater battery, cooling coil.

2-way Valve or 3-way Valve

2-way valve is best when applied with variable speed pumps. Rather variable-pumping systems should only use 2-way valve to reap the energy saving benefits.

The constant volume systems may employ 2-way or 3-way valve. While pumping costs will decrease to small amount with 2-way valve, other problems occur. The 2-way valves with constant volume system may some time lead to balancing problems in large network and may lead to water scarcity at some terminal locations. The pumps must incorporate the minimum recalculating system should the 2-way valve/s close to 100% close position. The systems incorporating 3-way valve ensure continuous circulation.

DISTRIBUTION SYSTEM

The distribution system is a network of ducts which transports the air between the conditioning equipment and the conditioned space(s). The system consists of outlet and inlet terminals (diffusers, registers, grilles) for distribution of air within the conditioned space, and dampers (automatic and manual) for control of air volume. The design of the distribution system greatly affects the amount of pressure drop (resistance) it adds to the total system. Low-pressure (low-velocity) systems are designed with duct velocities of 1,300 fpm or less for comfort air conditioning systems and up to 2,000 fpm for industrial air conditioning systems. High-pressure (high-velocity) systems employ duct velocities from 2,500 fpm on small systems (1,000 to 3,000 cfm) up to 6,000 fpm on large systems (40,000 to 60,000 cfm). Higher duct velocities result in higher duct system resistance (pressure drop) which results in increased fan horsepower.

Ductwork - Ductwork is the system of ducts and ductwork accessories that are used to connect air handling units and fans with the rooms, spaces, or exhaust hoods with which they are associated. The material used for a duct system must be based on the availability of the material, expertise of the duct installer, the type of duct already installed, the location of the installed duct, and the environment it is planned to be used in. For example, a fume hood that handles corrosive fumes should be connected with a non-metallic polyvinyl chloride (PVC) or stainless steel duct. Metallic ducts are usually built from sheets of aluminum or galvanized steel. The ducts may either be built with round or rectangular cross sections. Non-metallic ducts are usually built from fiberglass duct board, except for ducts handling corrosive fumes that are constructed from a PVC material. Fiberglass duct board sheets are generally in locations where the duct will not be damaged by objects or personnel. All joints should be sealed with a special pressure-sensitive tape made for this purpose; standard duct tape should not be used. Round PVC duct systems are built from standard PVC duct and standard fittings. Fittings and ducts are connected with glue. Rectangular PVC ducts are of a special construction and should be made by people who are skilled in this work. Flexible ducts can be bought and used directly without further fabrication. They are available either insulated or non-insulated.

HEAT RECOVERY DEVICES

Heat recovery devices are used to transfer heat between the intake fresh air and the exhaust air. This is useful in situations where a large amount of air has to be expelled, particularly when it is cold. In an office building situation, the fresh air quantities are normally quite small and this can make heat recovery fairly unattractive. But in large buildings with food courts, locker rooms and other spaces requiring large makeup air, it is very attractive.

Options for heat recovery include:

1. **Run around coil -** In this arrangement, waste heat recovery is facilitated by liquid coupled run around coils constructed to the same specification as the heating coils. A run-around coil system is a simple piping loop with an upstream precooling coil and a downstream reheating coil that sandwiches the main cooling coil. The circulating fluid is pumped to transfer heat from the warm mixed air to the off coil cold supply air. The run-around system reduces the cooling load on the main cooling coil; reheat is provided by the heat picked up by the circulating fluid in precooling coil instead of by an external source of expensive energy.

2. **Plate heat exchanger -** In this arrangement, a heat exchanger is situated to allow direct transfer of heat from one flow to the other. Heat is transferred from outdoor air coming into the air conditioner to the cold air leaving it. Again, the goal is to boost the portion of air conditioning capacity used for removal of latent heat by decreasing the need to remove sensible heat. To a limited extent, the exchanger itself will remove latent heat by condensing moisture on the entering air side. Outdoor air is introduced into one side of the heat exchanger, and is partially cooled. It then flows over the cold refrigeration coil for moisture removal and additional sensible cooling. The cold, saturated air then passes through the other side of the exchanger for warming before being introduced to

the air handler or the space. Cross flow type heat recovery provides up to 80% efficiency. While waste heat exchanger plates are typically manufactured in aluminum, special epoxy coatings or stainless steel plates are also common.

3. **Heat Pipe System -** In its simplest form, a heat pipe is a metal tube sealed at both ends, evacuated and charged with a vaporizable liquid (refrigerant). The liquid refrigerant at the bottom end readily turns to gas when that end of the pipe is warmed, and floats to the top end of the pipe. If that end is in a cooler environment, the gas condenses, releasing heat. The liquid then flows back to the bottom where the cycle begins again. The net result is heat transfer from bottom to top, without a compressor. Capillary action is sometimes used to help move the liquid, allowing for greater flexibility in configuration.

4. **Thermal Wheel -** In this arrangement, a rotating wheel is used to transfer heat from one air stream to the other. Energy recovery wheel can be incorporated into a ventilation system to transfer both the sensible and latent energy between outdoor and exhaust air streams. The core of an enthalpy wheel can be made from a variety of materials such as paper, metal or plastic, which is coated with a desiccant. This coating enables the wheel to transfer both sensible and latent energy between air streams. The enthalpy wheel provides first stage cooling or heating. For instance, in the summer the wheel removes moisture from incoming air and pre-cools it to lighten the load on the evaporator coil. Likewise, in colder temperatures (winter application) the wheel will inject moisture into supply air and pre-warm it to reduce the load on the reheat coil. In the summer and winter modes, the wheel saves energy. There are three general types of wheels being used today. They are sensible, enthalpy and regenerative.

   a. ***Sensible wheel*** - This wheel is not coated with a desiccant and therefore transfers only sensible energy. The wheel can be constructed of almost any material (paper, metal or plastic) and transfers energy between two air streams as the mass of the material gains or loses heat to the opposite air stream. The wheel rotates at a speed of 25 to 50 revolutions per minute.

b. ***Enthalpy wheel*** - It is similar to the sensible wheel except that a desiccant media is added to the wheel's surface. As the wheel rotates, it now can transfer sensible energy and humidity. This wheel also rotates at 25 to 50 revolutions per minute.

c. ***Regeneration wheel***- This wheel is used when low dewpoint conditions (<45°F) are required, such as industrial applications. It achieves low dewpoints by slowing the wheel to a speed of between 0.25 and 1 revolution per minute and by using an air stream heated to 250°F or more to drive off moisture and regenerate the wheel.

This heated air stream is typically focused on only 1/4 of the wheel's area thereby allowing 3/4 of the area to be available for the process side.

AIR HANDLER CONTROLS

The control system of an air conditioning system contains various control loops which automatically control selected functions of the air conditioning system operation. The control system can be very simple or very complex depending upon the size and complexity of the air conditioning system, the extent of operation, and the degree of sophistication desired.

Controls are essential to make the air behave in the manner desired. The basic controls for AHU are listed below:

1. **Thermostat -** A thermostat senses the temperature and regulates the control device. The control device may actuate a damper that directs the path of an airstream or may change the temperature of an airstream by directing its flow through a coil, or it can control the volume of air flowing in a duct system. Control device may actuate the 2-port or 3-port valve on the chilled water supply to the coil and through the action of a valve; it also may change the temperature of an airstream.

2. **Pressure Controls -** Pressure controls operate to control the pressure of air in a room, or the duct system through the action of dampers. A static pressure controller is needed for variable air volume system to signal variable frequency drive of supply and return

fan to adjust air flow rates. The damper being controlled is in the inlet duct to the supply fan, but it could equally well be an inlet vane damper or an outlet damper. The pressure controller could also operate a blower drive speed control. In this control, the pitch of motor and fan pulleys is varied by a device in response to a signal from the pneumatic static pressure controller.

3. **Airflow switches -** Airflow switches are mounted on the side of a duct with the blade inserted into the duct. The blade of the switch will move according to airflow in the duct, and it will make electrical contacts when air flows and break the contacts when airflow stops. The sensitivity of the switch to the airflow may be adjusted. Another control for the same purpose as the ones described above is the differential pressure switch that senses velocity pressure in the duct. This controller also has an adjustable pressure range.

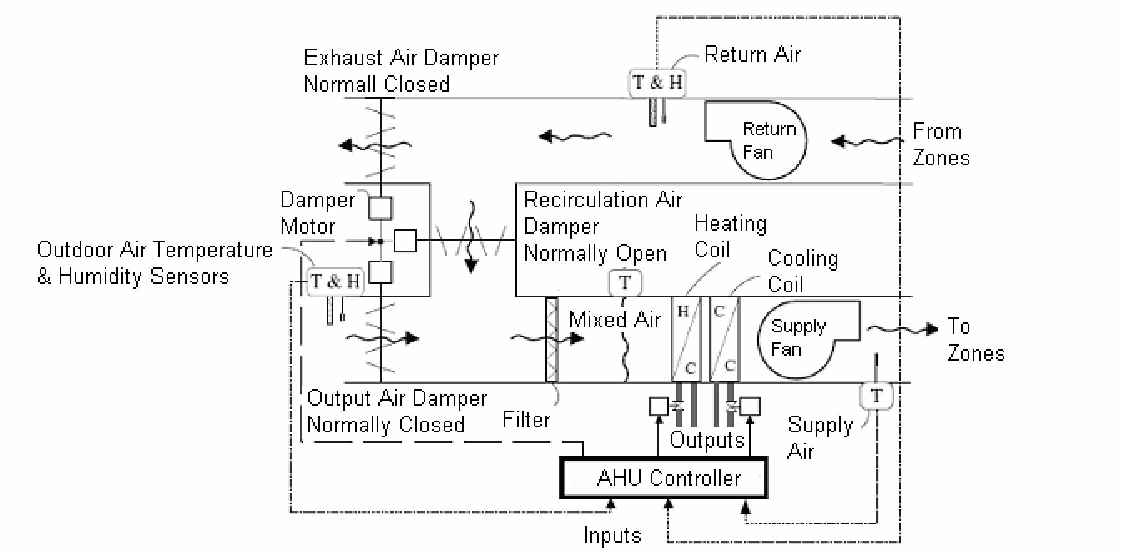
   Airflow control can be achieved by adjusting the fan speed, using inlet vanes or outlet dampers. The use of speed control method is most energy efficient as it results in maximum fan power savings and also cuts the noise down.

Control systems operate either pneumatically or electronically, or a combination of both can be used. For the most economical operation of the air conditioning system, controls must be maintained. Their calibrations should be routinely checked along with the proper operation of valves and dampers.

How these controls work in a sequence, is described in detail in section-4 of this course.

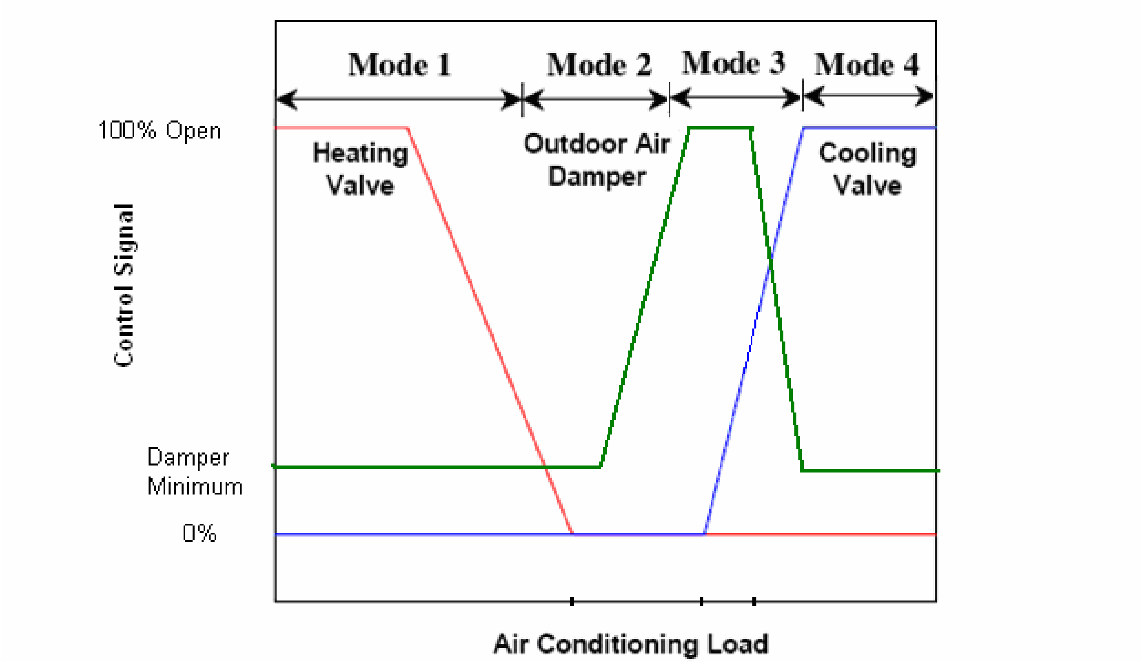
SECTION – 4 AIR HANDLING UNIT CONTROL

Air handling units are typically controlled to maintain a setpoint temperature at a location in the supply duct downstream of the supply fan. Outdoor air enters the AHU and is mixed with air returned from the building. The mixed air passes over the heating and cooling coils, where if necessary, it is conditioned prior to being supplied to the building.



Schematic Control Diagram of a Single Duct Air-Handling Unit

The typical operating sequence for AHU consists of four primary modes of operation during occupied periods for maintaining the supply air temperature at the set point. The relationship of the four modes to the control of the heating coil valve, the cooling coil valve and the mixing box dampers is shown in figure below.



In the heating mode (Mode 1), the heating coil valve is controlled to maintain the supply air temperature at the set point and the cooling coil valve is closed. The outdoor air damper is positioned to allow the minimum outdoor air necessary to satisfy ventilation requirements. As cooling loads increase, the AHU transitions from heating to cooling with outdoor air (Mode 2). In this mode, the heating and cooling coil valves are closed and the mixing box dampers are modulated to maintain the supply air temperature set point. As the loads continue to increase, the mixing dampers eventually saturate with the outdoor air damper fully open and the AHU changes modes again to mechanical cooling. When the AHU is operating in the mechanical cooling mode, the cooling coil valve modulates to maintain the supply air temperature set point, the heating coil valve is closed, and the outdoor air damper is either fully open or at its minimum position. Economizer control logic often uses a comparison of the outdoor and return air temperatures or enthalpies to determine the proper position of the outdoor air damper such that mechanical cooling requirements are minimized. Hence, the third primary mode (Mode 3) of operation is mechanical cooling with 100 % outdoor air and the fourth primary mode (Mode 4) of operation is mechanical cooling with minimum outdoor air.

The air-handling equipment must be controlled as a system and not as a series of conventional open-loop controllers tuned to individual set points. There are various other controls that sense the condition of air in the space and act upon other devices to make the air behave in the manner desired. Find below is a practical example of the AHU control scheme for an Oil & Gas facility that depicts the control philosophy for engineered chilled water HVAC system as applicable to the control rooms. The control sequence is designed to provide one or more of the following requirements:

a. To provide a comfortable environment for personnel,

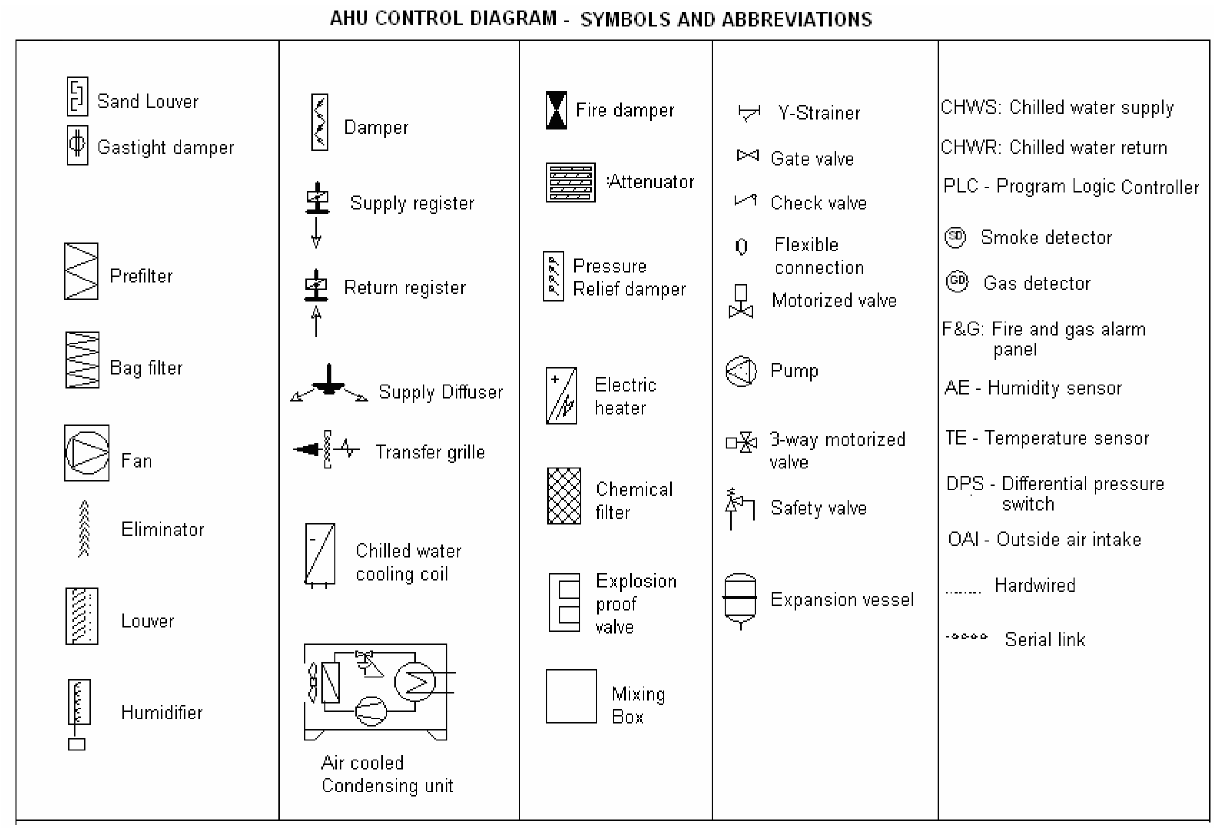
b. To provide controlled environment for essential electrical, instrument and control equipment,

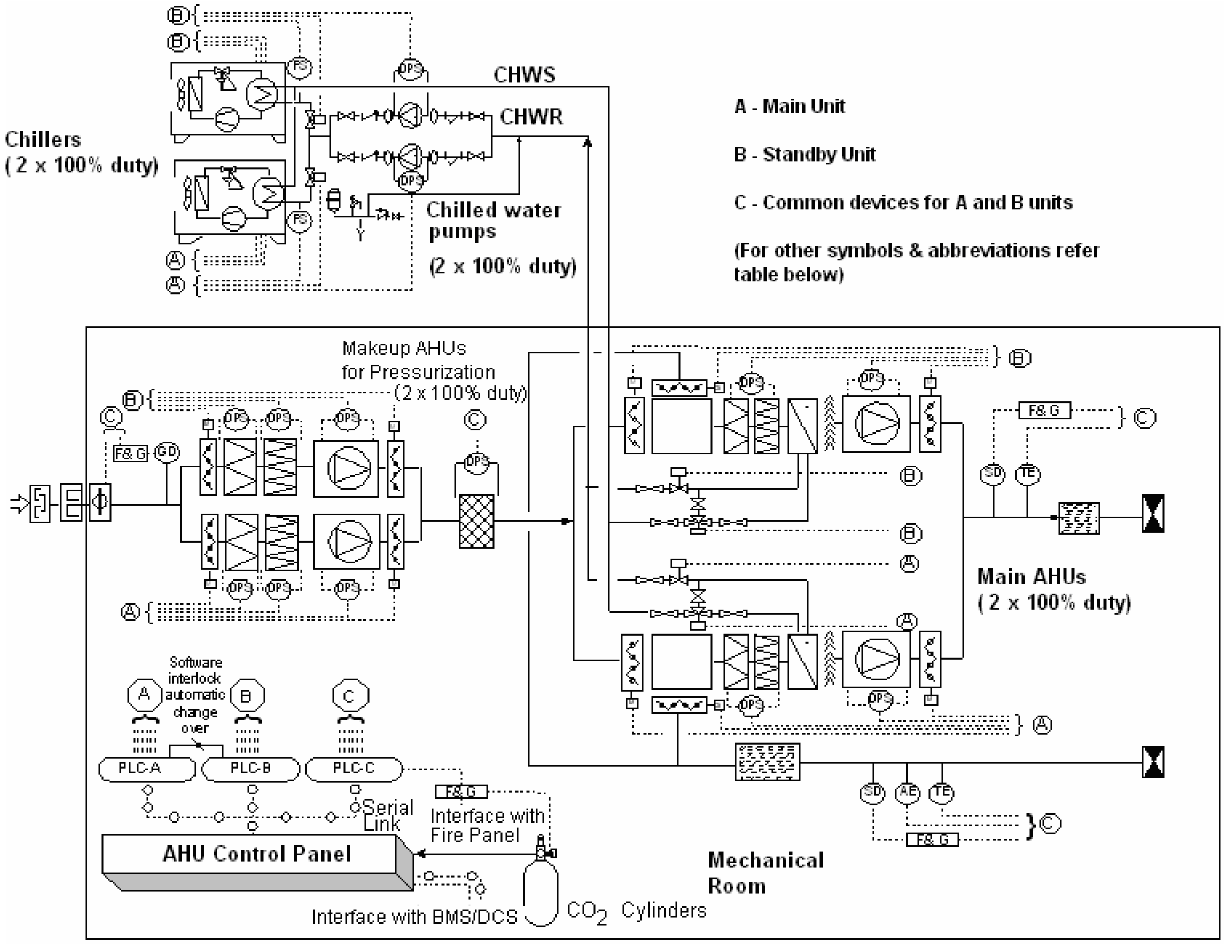
c. To pressurize the sensitive electrical and instrument areas to prevent ingress of dust,

d. To extract polluted air or dangerous gas when required (toilets, battery rooms etc.),

e. To provide safety interlocks against the dangers of fire, smoke and hazardous gas

A typical control logic schematic is shown below. Refer to other symbols and legends below:


AHU CONTROL DIAGRAM - SYMBOLS AND ABBREVIATIONS
Sand Louver
Gastight damper
Prefilter
Bag filter
Fan
Eliminator
Louver
Humidifier
Damper
Supply register
Return register
Supply Diffuser
Transfer grille
Chilled water cooling coil
Air cooled Condensing unit
Fire damper
Attenuator
Pressure Relief damper
Electric heater
Chemical filter
Explosion proof valve
Mixing Box
Y-Strainer
Gate valve
Check valve
Flexible connection
Motorized valve
Pump
3-way motorized valve
Safety valve
Expansion vessel
CHWS: Chilled water supply
CHWR: Chilled water return
PLC - Program Logic Controller
Smoke detector
Gas detector
F&G: Fire and gas alarm panel
AE - Humidity sensor
TE - Temperature sensor
DPS - Differential pressure switch
OAI - Outside air intake
Hardwired
Serial link



TYPICAL CONTROL DIAGRAM FOR OIL & GAS FACILITY

Note that in the scheme above all the major HVAC equipment such as chillers, air-handling units, exhaust fan and chilled water pumps are duplicated to provide 100% duty/standby operation. The master equipment is designated as A, the standby as B and the common equipment as C. HVAC system for this critical facility is designed for continuous running and is provided with a PLC based control system, which shall include modulating controls, sequence controls, operation controls, status and alarms.

In the control scheme above, the electrical/control panel consists of 3 sections: one for each group of duplicated duty/stand-by plant (A/B) and one for non-duplicated equipment (C). Software interlock switch is provided to select the duty/stand-by operation. In the event of failure of any duty equipment, the control system shall automatically initiate a sequenced changeover to stand-by equipment, which will be controlled via its corresponding controller.

The control panel is provided with status/ alarm of all components of HVAC system. The HVAC control panel shall monitor all necessary parameters required for precise environment control/safety and shall include but not be limited to the following:

a. Equipment On/Off/Trip

b. Duty/Standby Operation (A/B)

c. Fail/Out of service

d. Dampers Close/Open position

e. Motorized Valve Close/Open position

f. Filters Clean/dirty/clogged indication

g. Fire/smoke/gas detection alarm

h. Zone Temperature/Humidity indication

i. Building pressure positive/negative

Control Requirements

A test/auto switch spring loaded to the auto position shall be provided for all power equipment.

Gas tight shut-off dampers and fire dampers shall be electrically actuated and shall be fail-safe. Each damper shall be fitted with open and closed position switches, which shall provide interlocks and alarms. Fire dampers shall be electrically activated and configured so that loss of power to the damper causes the dampers to close.

The HVAC control panel shall be interfaced with the fire and gas (F&G) system for the area protection, isolation and shutdown purposes.

The plant distributed control system (DCS) or building management system (BMS) shall be used for remote monitoring of the HVAC system. The control panel shall be provided with necessary volt free dry contacts for connection to the DCS/BMS and F&G alarm panel.

The inside of HVAC control panel shall be protected by automatic release of carbon-dioxide system as per the requirements of design specifications and NFPA 12. Manual activation of CO_2 system shall be provided, in case the automatic actuation failure.

MCC shall be provided with two separate incoming power supplies: one connected to the bus section 'A' and one connected to bus section 'B', a bus coupler 'A' – 'B' shall be provided.

Start Up Operation Sequence

The HVAC system shall be started manually using local hand switch and shall be placed in auto mode. All the HVAC components shall be designated for duty/standby (A/ B) and common items (C) in their respective PLC's. The following start up sequence shall be adopted:

<u>Step#1</u>

Air-handling fan is the first equipment to be initiated. The sequence shall be as follows:

a. Common gas tight shut-off dampers/ fire dampers shall be commanded to 'Open' position by their respective actuators.

b. Motorized control dampers of the duty air-handling units shall be commanded to 'Open' position by their actuators.

c. Damper position shall be verified 'Open' by the control panel using limit switches.

d. Upon confirmation, supply fan of the duty air-handling units shall start

e. Differential pressure switch shall provide a 'Fan Run' status on the control panel.

f. Upon detection of low pressure across the fan, the fan shall trip and its motorized dampers shall be commanded to the closed position.

g. Following a duty air-handling fan trip indication, the stand-by air-handling unit's dampers are opened. Upon confirmation, the stand-by air-handling unit fan shall start and shall continue to run until stopped manually. Each fan shall have the following status indicated on the control panel:

- Fan running
- Fan stopped
- Fan failure

The extract fan of battery room air shall be continuously running all the time. Upon detection of low pressure the duty fan shall be tripped and the stand-by fan shall be started.

Step#2

Following 'Fan Run' indication of air-handling fan, a 'demand for cooling' signal is sent to the refrigeration chiller equipment. The chilled water pump shall be started first followed by the chiller. The sequence shall be as follows:

a. Motorized shut off valve for the duty chiller located at duty pump discharge shall 'Open'.

b. Motorized shut off valve for the air-handling unit cooling coil of the duty air-handling unit shall 'Open'.

c. Motorized valves position is verified open by the control panel using limit switches.

d. Upon confirmation, duty chilled water pump shall start.

e. The differential pressure switch on the pump provides 'Pump Run' status

f. On detecting low pressure across the pump, the pump shall trip and its shut off valve shall close.

g. Following a signal of duty pump trip, the motorized discharge valve of stand-by chilled water pump is ‘Opened’. Upon confirmation the stand-by pump shall ‘Start’. Each pump shall have the following status indicated on the control panel:

- Pump running
- Pump stopped
- Pump failure

Step#3

Once the chilled water pump is operational, the respective chiller is started. Following sequence is adopted:

a. Flow switch at chiller inlet shall sense flow through the chiller

b. Upon confirmation of required flow, the respective chiller shall be enabled.

c. Once the chiller is ‘Enabled’, it shall operate under the dictates of its own temperature control and safety interlocks

d. The chiller operational status shall be communicated and indicated on the control panel

e. The stand-by chiller shall be enabled when control system detects a failure of the duty chiller

Normal Operation

Following the initial start up, the system shall be automatically controlled through the temperature sensors, humidity sensors, gas and smoke detectors common to the HVAC system. The following sequence is adopted:

1. Temperature Control

   A temperature sensor installed in the return air duct, measures the space temperature. In event the space temperature exceeds or drop in relation to the pre-set limit, the chilled water flow through the air-handling unit cooling coil is modulated. A 3-way modulating valve located on the cooling coil inlet shall by-pass the chilled water flow to the return line at part load conditions.

2. Humidity Control

   A humidity sensor installed in the return air duct, measures the space humidity. In event the space humidity exceeds the pre-set limit; it will initiate a de-humidification program that overrides the temperature control program. Under excess humidity conditions, the 3-way modulating valve shall operate at full open position with zero by-pass. When the resulting dehumidification brings down the space humidity with in the limit, control of the 3-way modulating valve shall be transferred again to the space temperature control program.

All filtration system (including filters in the AHU and the chemical filtration unit) shall have a differential pressure switch that shall indicate the need for cleaning/replacement on high-pressure loss alarm.

The gas and smoke detection signals shall initiate HVAC system shut down as addressed below.

System Shutdown and Failure Routines

Once the HVAC system is started, the system is left operational all the time. Each item of HVAC equipment shall be provided with a local disconnect hand switch for manual shutdown.

The HVAC system shall be interlocked with the local F&G detection system. In event fire/smoke/hazardous gas is detected by the F&G system, the control system shall initiate a shutdown operation as follows and indication shall be provided on the control panel.

a. All gas tight shut-off dampers shall be closed.

b. All air handling units and fans shall be stopped.

c. All fire/smoke dampers shall be closed.

d. All motorized dampers of air handling units shall come to complete close.

e. Refrigerant chiller shall be tripped

On clearance of a fire/gas/smoke detection condition, the system shall follow a start up sequence as detailed above.

Planned Shutdown

The filters require cleaning periodically. Each air filter section shall be provided with a differential pressure switch to actuate a high-pressure loss alarm. The HVAC control panel shall indicate the filter clean, dirty and clogged status. The filters shall be removed for cleaning after air-handling unit fan shutdown. The HVAC system shall shut down in reverse order to its start up procedure.

The above example is chosen to describe the complex control features as applicable to the critical facilities. This can easily be adapted for smaller buildings requiring much lesser controls.

Concluding……

The modular packaged equipment offered by many of the major manufacturers is very standard. Designers can simply spec the performance requirements and pick and choose from a wide array of standard building blocks to assemble a unit that meets their specific project needs. HVAC designer must precisely estimate the capacity and airflow requirements, select the fan, coil and other auxiliaries and integrate these properly to the building control system.

