HVAC Concepts and Fundamentals

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HVAC CONCEPTS & DEFINITIONS

HVAC (pronounced either "H-V-A-C" or, occasionally, "*H-vak*") is an acronym that stands for "heating, ventilation and air conditioning". HVAC sometimes referred to as climate control is a process of treating air to control its temperature, humidity, cleanliness, and distribution to meet the requirements of the conditioned space. If the primary function of the system is to satisfy the comfort requirements of the occupants of the conditioned space, the process is referred to as comfort air conditioning. If the primary function is other than comfort, it is identified as process air conditioning.

HVAC systems are also important to occupants' health, because a well regulated and maintained system will keep spaces free from mold and other harmful organisms. The term ventilation is applied to processes that supply air to or remove air from a space by natural or mechanical means. Such air may or may not be conditioned.

An air conditioning system has to handle a large variety of energy inputs and outputs in and out of the building where it is used. The efficiency of the system is essential to maintain proper energy balance. If that is not the case, the cost of operating an air conditioning system will escalate. The system will operate properly if well maintained and operated (assumption was that it was properly designed in the first place, however, should sizing be a problem, even a relatively costly redesign might prove financially beneficial in a long run).

The HVAC industry had been historically regulated by the manufacturers of HVAC equipment, but Regulating and Standards organizations such as ASHRAE, SMACNA, ARI, ACCA, Uniform Mechanical Code, International Building Code, and AMCA have been established to support the industry and encourage high standards and achievement.

SECTION - 1   **THERMAL COMFORT**

The prime requirement in respect of the indoor climate in a building is that room temperature should be at a comfortable level, regardless of the weather conditions outside. In addition, the indoor air must be acceptably clean, lighting and acoustic conditions must be good etc.

Thermal comfort can be defined as a subjective response, or state of mind, when a person expresses satisfaction with the surrounding environment (ASHRAE Standard 55). The environment must provide light, air, and thermal comfort. Proper acoustics and hygiene are also important for physical comfort. While it may be partially influenced by a variety of contextual and cultural factors, a person's sense of thermal comfort is primarily a result of the body's heat exchange with the environment. This is influenced by four parameters that constitute the thermal environment (air temperature, radiant temperature, humidity and air speed), and two personal parameters (clothing and activity level, or metabolic rate).

HVAC and Thermal Comfort

The basic purpose of an HVAC system is to provide interior thermal conditions that a majority of occupants will find acceptable. Occasionally this may simply require that air be moved at an adequate velocity to enhance convective cooling and evaporation from the skin. Much more commonly, however, providing for occupant comfort will require that an HVAC system add or remove heat to or from building spaces. In addition, it is normally necessary for moisture to be removed from spaces during the summer; sometimes moisture will need to be added during the winter. The heat and moisture control functions of HVAC systems provide the foundation for key system components. The additional functions of air circulation and air quality control establish further component requirements. In specific building situations, supplemental functions, such as controlling smoke from fires

or providing background noise for acoustic privacy, may be imposed on an HVAC system -- along with a potential need for additional components. In order to explain how thermal interactions affect human comfort, it is first necessary to define heat and temperature.

Heat and temperature

Heat: Heat may be defined as energy in transit from a high-temperature object to a lower-temperature object. This heat transfer may occur by the mechanisms of conduction, convection and radiation.

- **Sensible heat**: Kind of heat that increases the temperature of air. It is an expression of the molecular excitation of a given mass of solid, liquid, or gas.

- **Latent heat**: Heat that is present in increased moisture of air. It changes the matter from solid to liquid or from liquid to gas. Heat that is required to change solid to liquid is called latent heat of fusion, and that which is required to change liquid to gas is called latent heat of vaporization.

- **Enthalpy**: Sum of sensible and latent heat of a substance e.g. the air in our environment is actually a mixture air and water vapor. If the enthalpy of air is known, and the enthalpy of desired comfort condition is also known, the difference between them is the enthalpy that must be added (by heating or humidification) or removed (by cooling or dehumidification).

- **Units**: The common measure of quantity of heat energy is British thermal unit (BTU). It is the heat energy required to raise one pound of water one degree Fahrenheit. The rate of heat flow in this unit is BTUH. The unit is Joule in International System. It is the heat required to raise one liter of water one degree Celsius. The rate of heat flow in this unit is Joules/sec or Watts (W). One watt per hour is equivalent to 3.412 BTU per hour. (1 Joule

= 0.0009478 BTU = 1 Watt; 1 Watt-Hour = 0.0009478*60*60 = 3.412 BTUH.)

- **Temperature**: A measure of the degree of heat intensity. The temperature difference between two points indicates a potential for heat to move from the warmer point to the colder point. Unit in English system is Fahrenheit, and in International System is Celsius.

- **Dry-bulb temperature (DB)**: The dry-bulb temperature is the temperature of air measured by a thermometer freely exposed to the air but shielded from radiation and moisture. More specifically, it is a measure of the intensity of kinetic energy of the molecules in the air. It is one of "the most important climate variables for human comfort and building energy efficiency".

- **Wet-bulb temperature (WB)**: The temperature registered by thermometer whose bulb is covered by a wetted wick and exposed to a current of rapidly moving air. It is the temperature air would have if part of its energy were used to evaporate the amount of water it would absorb to become fully saturated.

- **Dew point temperature**: The temperature at which condensation begins when the air is cooled.

Variables affecting physical comfort

Human beings are essentially constant-temperature creatures with a normal internal body temperature of 98.6°F. Heat is produced in the body as result of metabolic activity. If the internal temperature rises or falls beyond its normal range, mental and physical operation is jeopardized or affected, and if the temperature deviation is extreme, then serious physiological disorders or even death can result.

The physiological interpretation of comfort is the achievement of thermal equilibrium at our normal body temperature with the minimum amount of bodily regulation.

The factors that affect physical comfort are the following:

- **Metabolic rate**: It is the rate at which food consumed is converted into electromechanical energy to maintain physical functions. Heat is produced in the body as a result of metabolic activity. Its production can be controlled to a certain extent, by controlling metabolism or oxidation. Metabolic rate can also be defined as the rate of body heat production under conditions that minimize extra requirements for energy. Given a set of metabolic rate, however, the body must reject heat at the proper rate in order to maintain normal body temperature. Metabolic rate is proportional to body weight, and is also dependent upon activity level, body surface area, health, sex, age, amount of clothing, and surrounding thermal and atmospheric conditions. Metabolic rate is measured in **Met** units. For an average person, one Met unit corresponds approximately to 360 BTU per hour. A Met is the average amount of heat produced by a sedentary person, and any metabolic rate can be expressed in multiples of this standard unit (e.g. Office work = 1 Met).

  - The unit of the electromechanical energy produced due to metabolism is the Calorie. A Calorie is defined as the amount of heat required to raise the temperature of 1 gram water by 1 degree Celsius at 1 atmosphere pressure. This measure is typically used for food values. 1 Calorie = 4.1868 Joules.

- **Conduction** is the spontaneous transfer of thermal energy through matter, from a region of higher temperature to a region of lower temperature, and

acts to equalize temperature differences. It is also described as heat energy transferred from one material to another by direct contact.

- **Convection** is usually the dominant form of heat transfer in liquids and gases. Convection is circulation of a fluid or gas/air caused by temperature difference. Commonly *an increase in temperature produces a reduction in density*. Hence, when water is heated on a stove, hot water from the bottom of the pan rises, displacing the colder more dense liquid which falls. Mixing and conduction result eventually in a nearly homogeneous density and even temperature. In HVAC applications, convection becomes increasingly effective at dissipating heat as air temperature decreases and air movement increases. This is because, "faster the rate of air movement, the larger the temperature difference between the body and surrounding air, and the larger the body surface area, the greater the rate of transfer".

- **Evaporation**: It is exclusively a cooling mechanism. Evaporative losses become a predominant factor when ambient temperatures are very high. When surrounding temperature is about 70°F, most people lose sensible heat at a rate that makes them feel comfortable. If the surrounding temperature rises to skin temperature, the sensible heat loss drops to zero. If the ambient temperature continues to rise, the body gains heat from the environment, and the only way it can lose heat is by increasing evaporation. The moisture carrying potential of the air determines the rate of evaporation and evaporative heat loss. It is dependent on the relative humidity (RH) of surrounding air and the velocity of air motion.

- **Radiation** is the only form of heat transfer that can occur in the absence of any form of medium; thus it is the only means of heat transfer through a vacuum. Thermal radiation is a direct result of the movements of atoms and molecules in a material. Radiation affects two bodies when they are in

direct line of sight of each other. The rate of radiant transfer depends on temperature differential, the thermal absorption capacity of surfaces, and the distance between the surfaces. The body gains or loses heat by radiation according to the difference between the body surface and mean radiant temperature (MRT) of the surrounding surfaces.

Predictions of Thermal Comfort

Two conditions must be fulfilled to maintain thermal comfort. One is that the actual combination of skin temperature and the body's core temperature provide a sensation of thermal neutrality. The second is the fulfillment of the body's energy balance: the heat produced by the metabolism should be equal to the amount of heat lost from the body. Excess body heat requires to be continuously dissipated in order to maintain physical comfort. Mathematically, the relationship between the body's heat production and all its other heat gains and losses is:

$$S = -Q_{skin} - Q_{respiration} - W_{mech} + M$$

Where:

- S = Rate of Energy Storage in Body

- Q_{skin} = Rate of Energy Loss Thru Skin

- $Q_{respiration}$ = Rate of Energy Loss Through Respiration

- W_{mech} = Rate of Mechanical Work Performed by Body

- M = Metabolic Rate.

The body always produces heat, so metabolic rate (M) is always positive. If environmental conditions are such that the combined heat loss is less than the body's rate of heat production, then excess heat must be stored in body tissue. But

the body heat storage (S) is always small because it has a very limited thermal capacity. Therefore, when the interior temperature gets warmer, the body reacts to correct the situation by increasing blood flow to the skin surface and increasing perspiration. As a result, body heat loss is increased, thereby maintaining the desired body temperature. Shivering occurs when heat loss is greater than heat production.

Determinants of thermal comfort

Dry-bulb temperature

- Affects the rate of convective and evaporative cooling

- A fairly wide range of temperatures can provide thermal comfort when combined with RH, MRT, and air flow.

- Comfort conditions may be affected when temperature variation between floor and ceiling is more than 5ºF.

- Floor temperature should be between 65ºF and 84ºF.

Humidity

- Amount of water vapor present in a given space.

- Density of water vapor per unit volume of air is **absolute humidity** (AH)-- expressed in units of lbs. of water/cu-ft of dry air.

- **Specific humidity**--weight of water vapor per unit weight of dry air, expressed in grains/lb.

- **Degree of saturation** = water present in air/max water vapor-holding capacity of air.

- **Percentage humidity** = degree of saturation x 100.

- **Vapor pressure** is the pressure exerted by the motion of molecules of water vapor. It is dependent on the amount of water vapor in the air and the temperature of the air.

- **Relative humidity** (RH) = (actual vapor pressure of air-vapor mixture/pressure of water vapor when the air is completely saturated at the same DB temperature) x 100.

  o Human tolerance to humidity variations is much greater than temperature variations. In winter, the range is from 20 to 50%; in summer, the range extends up to 60% @ 75ºF.

  o High humidity causes condensation problems and reduces body heat loss by evaporative cooling.

  o Low humidity tends to dry throat and nasal passages also can cause static electrical sparks.

Air movement

- Required for removal of heat and humidity; also for the removal of air contaminants and body odor.

- Affects body heat transfer by convection and evaporation.

- No minimum velocity is specified when ambient temperatures are within acceptable limits.

- Velocity to be increased when ambient temperatures are high.

- Optimum acceptable limit is a function of temperature, humidity, and MRT.

Mean radiant temperature (MRT)

MRT is defined as mean temperature of surrounding surfaces with which body exchanges heat by radiation.

- Affects the rate of radiant heat loss from the body.

- MRT for office workers should be in the range 65ºF to 80ºF.

Operative Temperature

The ideal standard for thermal comfort can be defined by the operative temperature. This is the average of the air **dry-bulb temperature** and of the **mean radiant temperature** at the given place in a room.

The **operative temperature** is one of several parameters devised to measure the air's cooling effect upon a human body. It is equal to the dry-bulb temperature at which a specified hypothetical environment would support the same heat loss from an unclothed, reclining human body as the actual environment. In the hypothetical environment, the wall and air temperatures are equal and the air movement is 7.6 centimeters per second. From experiment it has been found that the operative temperature

$$T_0 = 0.48t_r + 0.19[\sqrt{vt_a} - (\sqrt{v} - 2.76)t_s]$$

Where

- t_r is the mean radiant temperature

- t_a is the mean air temperature

- t_s is the mean skin temperature (all in degrees Celsius); and

- v is the airspeed in centimeters per second.

Clothing

Thermal comfort also depends on the clothing and activity level of a person. The amount of clothing is measured against a standard amount that is roughly equivalent to a typical business suit, shirt, and undergarments. Activity level is compared to being seated quietly, such as in a classroom.

- Clothing insulation is described in units of Clo. The clo is a measure of thermal resistance and includes the insulation provided by any layer of trapped air between skin and clothing and insulation value of clothing itself. 1 clo unit will maintain a sedentary man at 1 met indefinitely comfortable in an environment of 21oC (69.8oF), 50% RH, and .01 m/s (20 ft/min) air movement.

- Some Clo values: light short sleeve shirt = 0.14, light pants = 0.26, hat and overcoat = 2.

- Adding 1 Clo of insulation permits a reduction of heat loss equivalent to that obtained by decreasing air temperature by approximately 13° F.

- Clo units can be converted to R-value in SI units (K/ (W/m²) or RSI) by multiplying Clo by 0.155 (1 Clo = 0.155 RSI). (In imperial units 1 Clo corresponds to an R-value of 0.88 °F ft²hr/Btu.)

Thermal indices

Thermal sensations can be described as feelings of being hot, warm, neutral, cool, cold, and a range of classification in between. There have been various attempts to find a single index that could be used to determine thermal comfort conditions.

Wind Chill

Wind chill is the apparent temperature felt on exposed skin, which is a function of the air temperature and wind speed. The wind chill temperature (often popularly called the **wind chill factor**) is always lower than the air temperature, except at higher temperatures where wind chill is considered less important. In cases where the apparent temperature is higher than the air temperature, the heat index is used instead.

Heat Index

The **heat index** (**HI**) is an index that combines air temperature and relative humidity in an attempt to determine the human-perceived equivalent temperature — how hot it feels, termed the felt air temperature. The human body normally cools itself by perspiration, or sweating, which evaporates and carries heat away from the body. However, when the relative humidity is high, the evaporation rate is reduced, so heat is removed from the body at a lower rate causing it to retain more heat than it would in dry air. Measurements have been taken based on subjective descriptions of how hot subjects feel for a given temperature and humidity, allowing an index to be made which corresponds a temperature and humidity combination to a higher temperature in dry air.
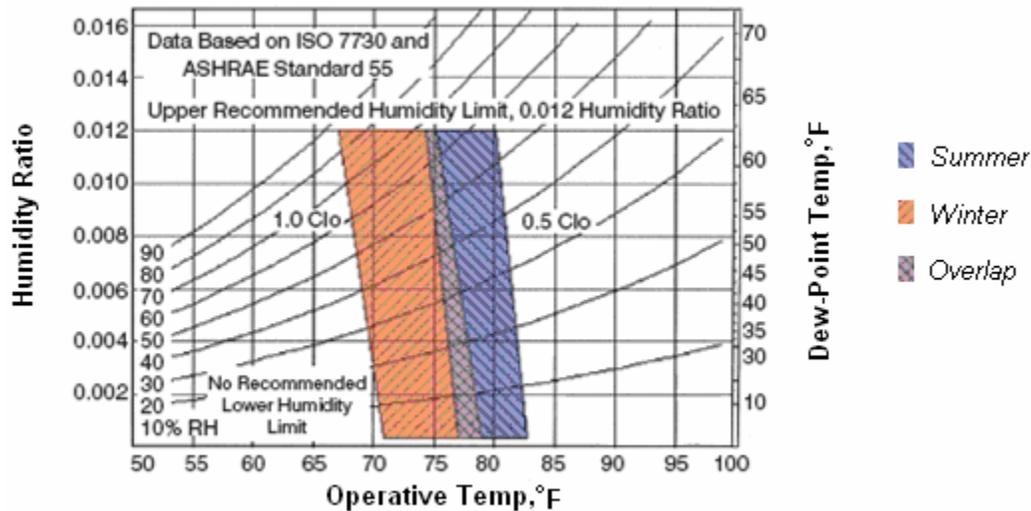
Effective temperature

Effective temperature is an experimentally determined index of the various combinations of dry-bulb, humidity, radiant conditions, and air movement that induce same thermal sensation. Effective temperature of a given space may be defined as the dry-bulb temperature of an environment at 50% RH and a specific uniform radiation condition; heat exchange of the environment is based on 0.6 clo, still air (40 fpm or less), 1 hour exposure time, and sedentary activity level (about 1 met). It is a reliable indicator of comfort with the thermal environment. Thus any space having an effective temperature of, say, 75ºF will induce a sensation of

warmth equivalent to 75ºF at 50% RH in almost still air and at a metabolic rate of 1 met.

Comfort zone

As said, thermal comfort is a subjective consideration i.e. under identical conditions one individual may feel comfortable, the second may not. Nevertheless, there is a range of conditions over which most people report that they are comfortable. It is an area plotted on the psychrometric chart that pertains to those conditions of dry-bulb temperature, wet-bulb temperature, wind speeds etc. in which most people wearing specified cloths and involved in specific activity will feel comfortable, i.e., neither too cold nor too warm. The comfort range of temperature varies between 70 to 76°F dry bulb temperatures and 45 - 65% relative humidity. This applies mainly to summer air-conditioning. During cold winters the comfort condition would be in the range of 65 to 68°F dry bulb temperature and relative humidity of a minimum of 30%.

PMV* Studies of personal comfort have shown that relative humidity ranges between 30% and 65% can be considered 'comfortable' depending on activity. However, from the standpoint of indoor air quality, upper ranges should be maintained below 50% (dust mite populations increase rapidly at relative humidity levels above 50% and fungal amplification occurs above 65%). Below 30% respiratory irritation occurs or static electric currents are a concern.

Acceptable range of operative temperature and humidity

Two comfort zones are defined by the shaded regions-one for winter and one for summer. The thermal conditions within these envelopes are estimated to be acceptable to 80 percent of the occupants when wearing clothing as indicated on the chart. To satisfy 90% of the people, the limits are reduced to one third of the ranges. In the region where the zones overlap, people in summer dress tend to be slightly cool, and those in winter clothing would feel a slightly warm sensation.

The upper and lower limits of humidity are based on respiratory health, mold growth, and other moisture-related problems apart from physical comfort.

*PMV represents the 'predicted mean vote' (on the thermal sensation scale) of a large population of people exposed to a certain environment. PMV is derived from the physics of heat transfer combined with an empirical fit to sensation. PMV establishes a thermal strain based on steady-state heat transfer between the body and the environment and assigns a comfort vote to that amount of strain. PPD is the predicted percent of dissatisfied people at each PMV. As PMV changes away from zero in either the positive or negative direction, PPD increases.

Learn more about "thermal comfort" --- refer

http://dt.fme.vutbr.cz/enviro/Pohoda/thermal.htm

SECTION – 2   **PSYCHROMETRICS**

Definition

- Study of atmospheric conditions

- Deals with the thermodynamic properties of air-water vapor mixture

- Describes and analyzes the combined interactions of air, heat, and moisture.

Psychrometric chart

A psychrometric chart is a graphical presentation of the thermodynamic properties of moist air and various air-conditioning processes and air-conditioning cycles. A psychrometric chart also helps in calculating and analyzing the work and energy transfer of various air-conditioning processes and cycles.

The most common chart used by practitioners and designers is the temperature - humidity ratio (w) chart in which the dry bulb temperature (DBT) appears horizontally as the abscissa and the humidity ratios (w) appear as the vertical axis.

Abridged sample of psychrometric chart is shown below:
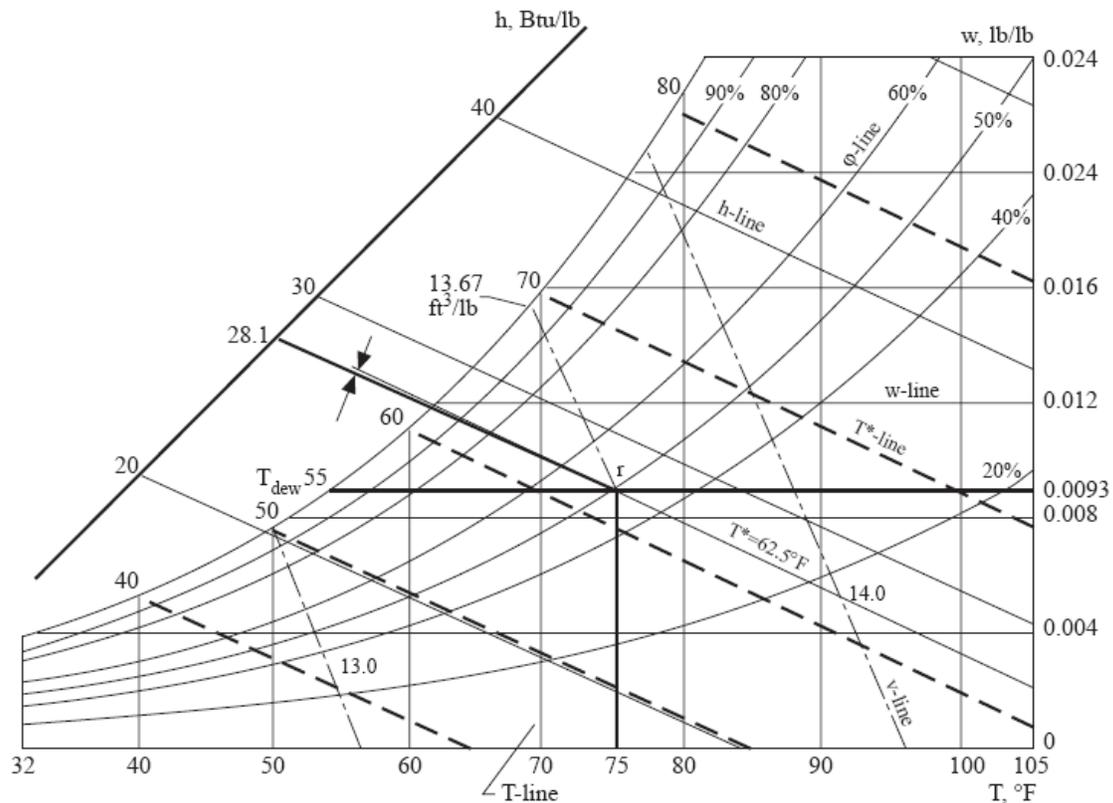
How to Read Psychrometric Chart

Moist air has seven independent thermodynamic properties (DBT, WBT, DPT, RH, humidity ratio, specific enthalpy, and specific volume). For a given air pressure or elevation, any additional two of the independent properties determine the state of moist air on the psychrometric chart and the remaining properties.

- **Dry bulb temperature (DBT):** This can be determined from the abscissa on the X-axis, the horizontal axis. Verticals lines designate DB. A standard psychrometric chart for air conditioning applications has a temperature range of 32 to 120°F.

- **Wet bulb temperature (WBT):** Diagonals rising upward from left to right having negative slope slightly greater than that of the h-lines. The slope of the line of constant WBT reflects the heat of vaporization of the water

required to saturate the air of a given relative humidity.A wet bulb line meets the DBT line of the same magnitude on the saturation curve (curved peripheral edge representing 100% RH).

- **Dew point temperature (DPT):** Follow the horizontal line from the point where the line from the horizontal axis arrives at 100% RH, also known as the *saturation curve*. **Dew point** is the temperature at which a moist air sample at the same pressure would reach saturation. At this saturation point, water vapor would begin to condense into liquid water fog or (if below freezing) solid hoarfrost, as heat is removed.

- **Relative humidity (RH - φ):** Curved lines that radiate from lower left to upper right are RH lines.

  o Horizontal line at the bottom represents 0% RH; the uppermost curved line is 100% RH line (also termed as saturation line).

  o Intersection point of a water content line and the saturation line is known as the dew point.

A saturation curve is a curve of the locus of state points of saturated moist air, that is, φ = 100%. On a saturation curve, dry bulb temperature (DBT), wet temperature bulb (WBT) and dew point temperature (DPT) have the same value.

- **Humidity ratio:** Humidity ratio w-lines are horizontal lines. Marked on the Y-axis, they range from 0 to 0.28 lb/lb. Also known as "moisture content", "mixing ratio", or "specific humidity", it is the proportion of mass of water vapor per unit mass of dry air at the given conditions (DBT, WBT, DPT, RH, etc.). For a given DBT there will be a particular humidity ratio for which the air sample is at 100% relative humidity. Humidity Ratio is

dimensionless, but is sometimes expressed as grams of water per kilogram of dry air or grains of water per pound of air.

- **Specific enthalpy (h):** Enthalpy lines incline downward to the right-hand side (negative slope) at an angle of 23.5° to the horizontal line and have a range of 12 to 54 Btu/lb. Enthalpy measures the heat content of air and is expressed in BTU per pound of air.

- **Specific volume (v):** Specific volume lines are represented by the diagonal lines close to 90º. Specific volume lines have a far greater negative slope than h-lines and WBT lines. The moist volume ranges from 12.5 - 15 ft^3/lb. Specific volume, also called Inverse Density, is the volume per unit mass of the air sample.

The versatility of the psychrometric chart lies in the fact that by knowing two properties, the other properties can be determined. Thermodynamic properties of moist air are affected by atmospheric pressure. The standard atmospheric pressure is 29.92in-Hg (14.697 psi) at sea level. ASHRAE also published charts for high altitudes of 5000 ft, 24.89 in- Hg, and 7500 ft, 22.65in- Hg. Both of them are in the normal temperature range.

Example

An air-conditioned room at sea level has an indoor design temperature of 75°F and a relative humidity of 50%. Determine the humidity ratio, enthalpy, density, dew point, and thermodynamic wet bulb temperature of the indoor air at design condition.

Solution

Find the room temperature 75°F on the horizontal temperature scale.

- Draw a line parallel to the 75°F temperature line and establish the point where it meets the relative humidity curve of 50% at point (r). This point denotes the state point of room air.

- Draw a horizontal line toward the humidity ratio scale from point (r). This line meets the ordinate and thus determines the room air humidity ratio w = 0.0093 lb/lb.

- Draw a line from point (r) parallel to the enthalpy line. This line determines the enthalpy of room air on the enthalpy scale, h = 28.1 Btu/lb.

- Draw a line through point (r) parallel to the moist volume line. The perpendicular scale of this line indicates v = 13.67 ft^3/lb.

- Draw a horizontal line to the left from point (r). This line meets the saturation curve and determines the dew point temperature, T_{dew} = 55°F.

- Draw a line through point (r) parallel to the wet bulb line. The perpendicular scale to this line indicates that the thermodynamic wet bulb temperature (WBT) = 62.5°F.

Psychrometrics Concept using Ideal Gas Law

Ideal gas law describes the relationship of thermodynamic properties of real gas, and is given by equation:

pv = n RT

Or

pV = n RT

Where

- p is the absolute pressure of the gas, psf (1 psf = 144 psi)

- v is the specific volume of the gas, ft^3/lb

- V is the Volume of the gas, ft^3

- n is the number of moles of gas [for air n = 28.9645 lbm / (lb-mol)]

- R is the universal gas constant [for air R is found by dividing the universal gas constant by the molecular weight i.e. 1545.32 / 28.9645 = 53.352 ft - lb / lbm-R]

- T is the absolute temperature, R

By manipulating the ideal gas equation, a relationship between the ideal gas law and the density for air can be developed. As the amount of substance could be given in mass instead of moles, sometimes an alternative form of the ideal gas law is useful. The number of moles (n) is equal to the mass (m) divided by the molar mass (M):

n = m / M

By replacing, we get

$$p\ V = \frac{m}{M}\ R\,T$$

From where

$$p\ = \rho\ \frac{R\,T}{M}$$

This form of the ideal gas law is particularly useful because it links pressure, density and temperature in a unique formula independent from the quantity of the

considered gas. Looking at the new equation one can see that the density is inversely proportional to the gas constant R.

Air is not 100% dry and contains moisture. The amount of water vapor contained in the moist air within the temperature range 0 to 100°F changes from 0.05 to 3% by mass. The variation of water vapor has a critical influence on the characteristics of moist air and can have a dramatic effect on our perception of comfort.

For engineering purposes, moist air can be considered as a mixture of perfect gases and obeys the ideal gas law. Thus,

- Dry air : $p_a V = n_a RT$

- Water vapor $p_w V = n_w RT$

Where

p_a and p_w are the partial pressures of dry air and water vapor, respectively. The terms n_a and n_w correspond to the number of moles of these components in the mixture.

These two equations may be combined.

$$\left(p_a + p_w \right) V = \left(n_a + n_w \right) RT$$

Dalton's Law

Dalton's Law states that the total pressure exerted by a mixture of perfect gases is the same as that exerted by the constituent gases independently. The sum of the partial pressures above, therefore, is the total pressure exerted by the moist air.

$p_{at} = p_a + p_w$

Where

- p_{at} = atmospheric pressure of the moist air, psia

- p_a = = partial pressure of dry air, psia

- p_w = partial pressure of water vapor, psia

Dalton's law is summarized from the experimental results and is more accurate at low gas pressure. A consequence to Dalton's law of Partial Pressure is that it can be extended to describe the relationship of internal energy, enthalpy, and entropy of the gaseous constituents in a mixture, for example, the total enthalpy of a mixture of gases will equal the sum of the enthalpies of each component part, i.e.:

$h = m_1 . h_1 + m_2. h_2 + m_3 . h_3 + ...$ etc

Humidity and Enthalpy

The **humidity ratio** of moist air, w, in lb/lb is defined as the ratio of the mass of the water vapor, m_w to the mass of dry air, m_a, or

$$w = \frac{m_w}{m_a} = \frac{0.62198 p_w}{\left(p_{at} - p_w\right)}$$

The **relative humidity** of moist air, φ, or RH, is defined as the ratio of the mole fraction of water vapor, x_w, to the mole fraction of saturated moist air at the same temperature and pressure, x_{ws}. Using the ideal gas equations, this relationship can be expressed as:

$$\varphi = \frac{x_w}{x_{ws}}\bigg|_{T,p} = \frac{p_w}{p_{ws}}\bigg|_{T,p}$$

And

$$x_w = \frac{n_w}{\left(n_a + n_w\right)} \quad ; \quad x_{ws} = \frac{n_{ws}}{\left(n_a + n_{ws}\right)}$$

$$x_a + x_w = 1$$

Where

- p_{ws} = pressure of saturated water vapor, psia

- T = temperature, °F

- n_a, n_w, n_{ws} = number of moles of dry air, water vapor, and saturated water vapor, mol

Degree of saturation (μ) is defined as the ratio of the humidity ratio of moist air, w, to the humidity ratio of saturated moist air, w_s, at the same temperature and pressure:

$$\mu = \frac{w}{w_s} \bigg|_{T,p}$$

The difference between φ and μ is small, usually less than 2%.

At constant pressure, the difference in specific enthalpy of an ideal gas, in Btu/lb, is:

$$\Delta h = Cp*\Delta T$$

Here Cp represents the specific heat at constant pressure, in Btu/lb. For simplicity, the following assumptions are made during the calculation of the enthalpy of moist air:

- At 0°F, the enthalpy of dry air is equal to zero.

- All water vapor is vaporized at 0°F.

- The enthalpy of saturated water vapor at 0°F is **1061 Btu/lb**.

- The unit of the enthalpy of the moist air is Btu per pound of dry air and the associated water vapor, or Btu/lb.

Then, within the temperature range 0 to 100°F, the enthalpy of the moist air can be calculated as:

$$h = c_{pd}T + w\left(h_{g0} + c_{ps}T\right)$$

$$= 0.240T + w(1061 + 0.444T)$$

Where

- c_{pd}, c_{ps} = specific heat of dry air and specific heat of water vapor at constant pressure, Btu/lb°F. Their mean values can be taken as 0.240 and 0.444 Btu/lb°F, respectively.

- h_{g0} = specific enthalpy of saturated water vapor at 0°F.

Moist Volume, Density, Specific Heat, and Dew Point

The specific moist volume v, in ft^3/lb, is defined as the volume of the mixture of dry air and the associated water vapor when the mass of the dry air is exactly 1 lb:

$$v = \frac{V}{m_a}$$

Where

- V = total volume of the moist air, ft^3.

Since moist air, dry air, and water vapor occupy the same volume,

$$v = \frac{R_a T_R}{P_{at}} \left(1 + 1.6078w\right)$$

Where

- R_a = gas constant for dry air.

Moist air density, often called air density r, in lb/ft^3, is defined as the ratio of the mass of dry air to the total volume of the mixture, or the reciprocal of the moist volume:

$$\rho = \frac{m_a}{V} = \frac{1}{v}$$

The sensible heat of moist air is the thermal energy associated with the change of air temperature between two state points. In Equation (9.2.8), (cpd + wcps) T indicates the sensible heat of moist air, which depends on its temperature T above the datum 0°F. Latent heat of moist air, often represented by wh_{fg0}, is the thermal energy associated with the change of state of water vapor. Both of them are in Btu/lb.

Within the temperature range 0 to 100°F, if the average humidity ratio w is taken as 0.0075 lb/lb, the specific heat of moist air cpa can be calculated as:

$$c_{pa} = c_{pd} + wc_{ps} = 0.240 + 0.0075 \times 0.444 = 0.243 \text{ Btu/lb } {}^\circ F$$

The dew point temperature T_{dew}, in °F, is the temperature of saturated moist air of the moist air sample having the same humidity ratio at the same atmospheric pressure. Two moist air samples of similar dew points T_{dew} at the same atmospheric pressure have the same humidity ratio w and the same partial pressure of water vapor pw

AIR CONDITIONING PROCESSES

Psychrometrics is widely used to illustrate and analyze the change in properties and the thermal characteristics of the air-conditioning process and cycles. An air-conditioning process describes the change in thermodynamic properties of moist air between the initial and final stages of conditioning as well as the corresponding energy and mass transfers between the moist air and a medium, such as water, refrigerant, absorbent or adsorbent, or moist air itself. The energy balance and conservation of mass are the two principles used for the analysis and the calculation of the thermodynamic properties of the moist air.

Four basic processes for summer and winter air conditioning systems are 1) Mixing, 2) Sensible Cooling and Heating, 3) Cooling with Dehumidification and 4) Humidification
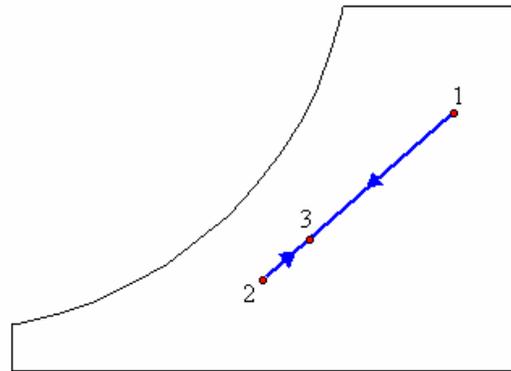
MIXING

Two air streams are mixed in air conditioning when fresh air (m_1) is brought in from outside and mixed with recirculated air (m_2). The resulting air mixture is shown below as (m_3).

The mixing ratio is fixed by dampers. Sometimes, in more sophisticated plant, modulating dampers are used which are driven by electric motors to control the mixture of air entering the system.

The diagrams below show mixing of two air streams.
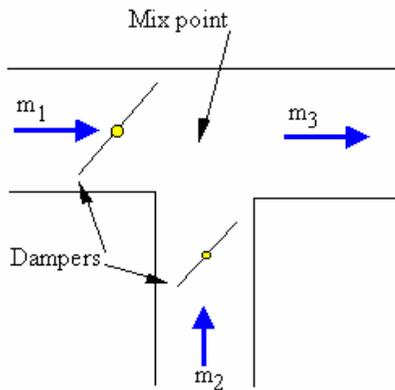
TWO AIR STREAMS MIXING

PSYCHROMETRIC CHART SHOWING TWO AIR STREAMS MIXING

Where two air streams are mixed, the psychrometric process is shown as a straight line between two air conditions on the psychrometric chart, thus points 1 and 2 are joined and the mix point 3 will lie on this line. Based on the principle of heat balance and conservation of mass:

$m_1 + m_2 = m_3$

$m_1 h_1 + m_2 h_2 = m_3 h_3$

$m_1 w_1 + m_2 w_2 = m_3 w_3$

$m_1 T_1 + m_2 T_2 = m_3 T_3$

In Equations above, m, represents the mass flow rate of air, lb/min; *h* the enthalpy, in Btu/lb; w the humidity ratio, in lb/lb; and *T* the temperature, in °F. Subscripts 1 and 2 indicate air streams 1 and 2 and 3 the mixture; also,

$$\frac{m_1}{m_3} = \frac{\left(h_2 - h_3\right)}{\left(h_2 - h_1\right)} = \frac{\left(w_2 - w_3\right)}{\left(w_2 - w_1\right)}$$

Similarly

$$\frac{m_2}{m_3} = \frac{(h_3 - h_1)}{(h_2 - h_1)} = \frac{(w_3 - w_1)}{(w_2 - w_1)}$$

Mixing point 3 must lie on the line that joins points 1 and 2.

SENSIBLE COOLING AND HEATING

When air is heated or cooled sensibly, that is, when no moisture is added or removed, this process is represented by a horizontal line on a psychrometric chart.

AIR STREAM IS HEATED OR COOLED SENSIBLY.

PSYCHROMETRIC CHART SHOWING SENSIBLE HEATING OR COOLING
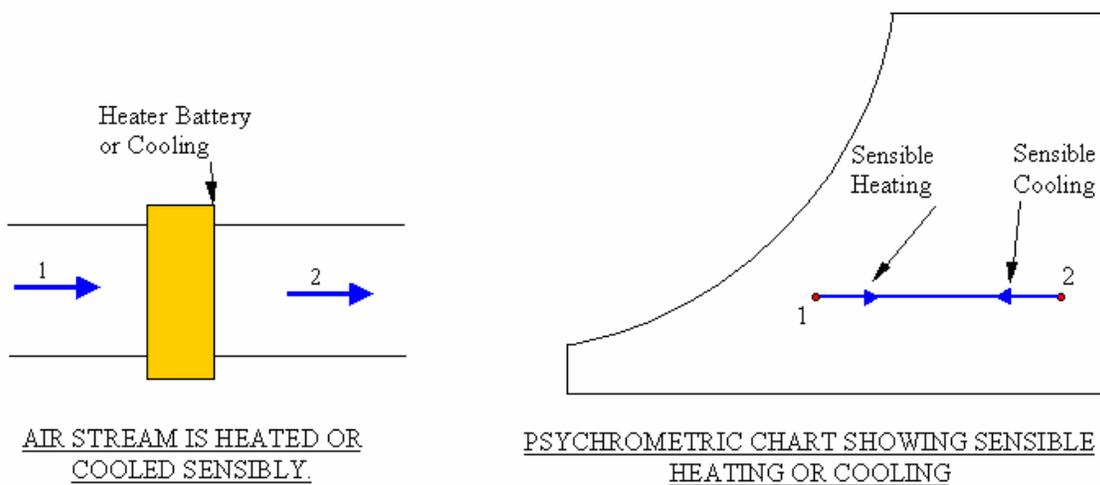
For sensible heating:

A sensible heating process adds heat to the moist air in order to increase its temperature; its humidity ratio remains constant. A sensible heating process occurs when moist air flows over a heating coil. Heat is transferred from the hot water inside the tubes to the moist air. The rate of heat transfer from the hot water to the colder moist air is often called the heating coil load ($Q_{sensible}$), in Btu/hr, and is calculated from Equation:

$Q_{sensible} = m * Cp (T_2 - T_1)$

Or

$Q_{sensible} = 60 * V * \rho * Cp (T_2 - T_1)$

Where

- $Q_{sensible}$ = sensible heat in Btu per hour

- m = mass flow rate of air, lbs/hr

- V = volume flow rate of supply air, cfm

- ρ = density of supply air, lb/ft^3

- T_2, T_1 = moist air temperature at final and initial states of an air-conditioning process, °F and the mass flow rate of supply air

Or more accurately from psychrometric chart:

$Q_{sensible} = m \times (h_2 - h_1)$

Where

- h_2, h_1 = moist air enthalpy at final and initial states of an air-conditioning process, Btu/lb

For sensible cooling:

A sensible cooling process removes heat from the moist air, resulting in a drop of its temperature; The sensible cooling process occurs when moist air flows through a cooling coil containing chilled water at a temperature equal to or greater than the dew point of the entering moist air. The sensible cooling load can also be calculated similar way as sensible heating equation.

COOLING AND DEHUMIDIFICATION

In a cooling and dehumidifying process, both the humidity ratio and temperature of moist air decrease. Some water vapor is condensed in the form of liquid water, called a condensate.

The most commonly used method of removing water vapor from air (dehumidification) is to cool the air below its **dew point**. The dew point of air is when it is fully saturated i.e. at 100% saturation. When air is fully saturated it cannot hold any more moisture in the form of water vapor. If the air is cooled to the dew point air and is still further cooled then moisture will drop out of the air in the form of condensate.

This can be shown on a psychrometric chart as air sensibly cooled until it becomes fully saturated (the dew point is reached) and then the air is cooled latently to a lower temperature.

This is apparent on the psychrometric chart as a horizontal line for sensible cooling to the 100% saturation curve and then the process follows the 100% saturation curve down to another point at a lower temperature. This lower temperature is sometimes called the **Apparatus dew Point (ADP)** of the cooling coil. In reality the **ADP** of the cooling coil is close to the cooling liquid temperature inside the coil.

Chilled water or refrigerant may be the cooling liquid.

The psychrometric process from state point 1 to 2 to 3 may be shown as a straight line for simplicity.
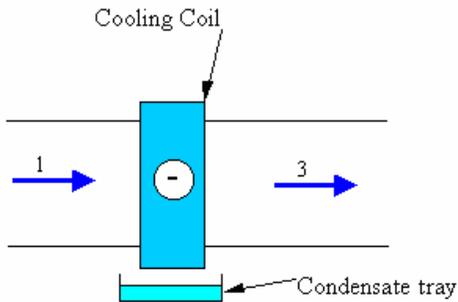
AIR STREAM IS COOLED AND
DEHUMIDIFIED
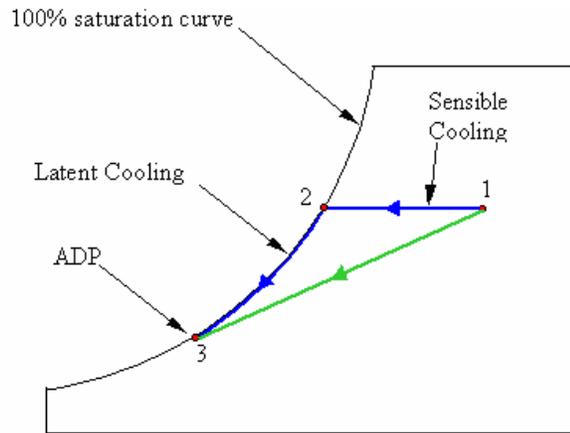
PSYCHROMETRIC CHART SHOWING COOLING
AND DEHUMIDIFICATION

The total amount of cooling input to the air approximates to;

$$Q_{1-3} = m \times (h_1 - h_3)$$

The sensible heat removed is:

$$Q_{1-2} = m \times (h_1 - h_2)$$

The latent heat removed is:

$$Q_{2-3} = m \times (h_2 - h_3)$$

Where:

- Q = Cooling energy (Btu)

- m = mass flow rate of air (lbs/hr)

- h = specific enthalpy of air (Btu/lbs) found from psychrometric chart.

In the absence of a suitable psychrometric chart the following formula may be used;

The sensible heat removed is: $Q_{1-2} = m \times C_p \times (T_1 - T_2)$

The latent heat removed is: $\quad Q_{2-3} = m \times hfg \times (w_2 - w_3)$

Where:

- Q = Cooling energy (Btu/hr)

- m = mass flow rate of air (lbs/hr)

- Cp = Specific heat capacity of air, may be taken as 1.01 Btu/lb- degF

- T = Dry bulb temperature of air (^{o}F)

- hfg = latent heat of vaporization of water, may be taken as 1060 Btu/lb

- w = moisture content of air from psychrometric chart (lb/lb-dry air)

Sensible Heat Factor or Ratio (SHR)

The sensible heat ratio (SHR) of an air-conditioning process is defined as the ratio of the change in absolute value of sensible heat to the change in absolute value of total heat, both in Btu/hr:

$$SHR = \frac{Q\ sensible}{Q\ total} = \frac{Q\ sensible}{Q\ sensible + Q\ latent}$$

SHR is a useful indication of dehumidification requirements. Lower SHR value indicates that the dehumidification requirement will be high and the supply airflow rate will be less (refer end of this section for explanation).

The key indicators are:

- SHR from 0.95 - 1.00 for Precision air conditioning (computers and data canters)

- SHR from 0.65 - 0.75 for Comfort cooling (office applications, people)

  o SHR from 0.50 - 0.60 for Dehumidification (pools and outside air)

HUMIDIFICATION

In a humidifying process, water vapor is added to moist air and increases the humidity ratio of the moist air entering the humidifier if the moist air is not saturated. Large-scale humidification of moist air is usually performed by steam injection, evaporation from a water spray, atomizing water, a wetted medium, or submerged heating elements.

Humidification increases latent heat and is represented by moving upward and normally accompanies heating. [Dehumidification decreases latent heat and is shown by moving downward – it is associated with cooling].

The psychrometric process is shown below.
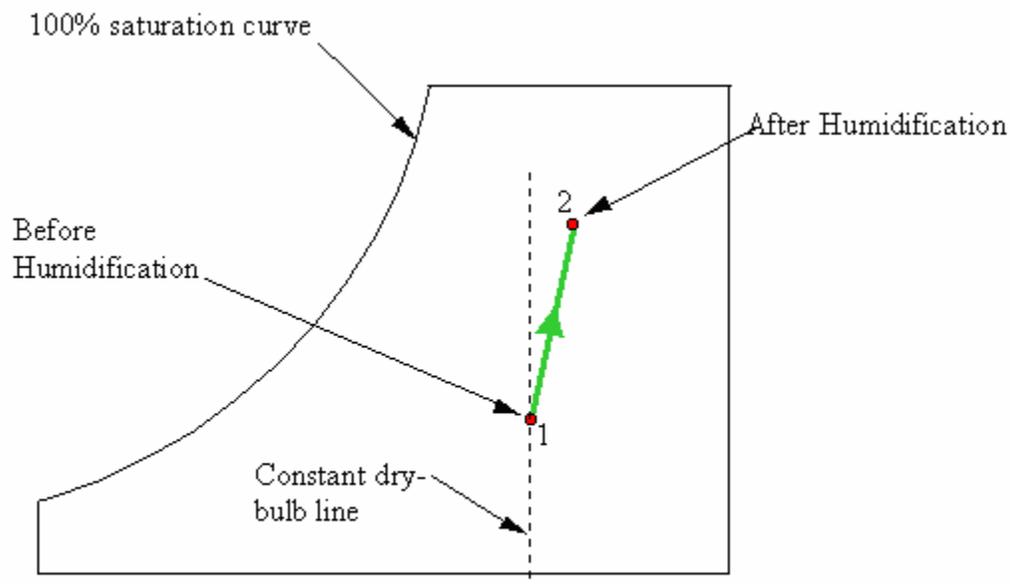
PSYCHROMETRIC CHART SHOWING HUMIDIFICATION

The humidifying capacity, W, in lb/min is given by:

$$W = V * \rho * (w_2 - w_1)$$

Where

- V= volume flow rate of supply air, cfm

- ρ = density of supply air, lb/ft^3

- w_2, w_1 = moisture content of air from psychrometric chart at final and initial states of an air-conditioning process, (lb/lb-dry air)

SUMMER MODE AIR CONDITIONING PROCESS

An air-conditioning cycle comprises several air-conditioning processes that are connected in a sequential order. An air-conditioning cycle determines the operating performance of the air system in an air conditioning system. The working substance to condition air may be chilled or hot water, refrigerant, desiccant, etc.

Each type of air system has its own air-conditioning cycle. Psychrometric analysis of an air-conditioning cycle is an important tool in determining its operating characteristics and the state of moist air at various system components, including the volume flow rate of supply air, the coil's load, and the humidifying and dehumidifying capacity.

According to the cycle performance, air-conditioning cycles can be grouped into two categories:

- Open cycle, in which the moist air at its end state does not resume its original state. An air conditioning cycle with all outdoor air is an open cycle.

- Closed cycle, in which moist air resumes its original state at its end state. Air-conditioning cycle that conditions the mixture of recirculating and outdoor air, supplies it; recirculates part of the return air, and mixes it again with outdoor air is a closed cycle.

A basic air-conditioning system is a packaged system of supply air at a constant volume flow rate, serving a single zone, equipped with only a single supply/return duct. A single zone is a conditioned space for which a single controller is used to maintain a unique indoor operating parameter, probably indoor temperature.

A basic air-conditioning schematic for summer mode is shown below:
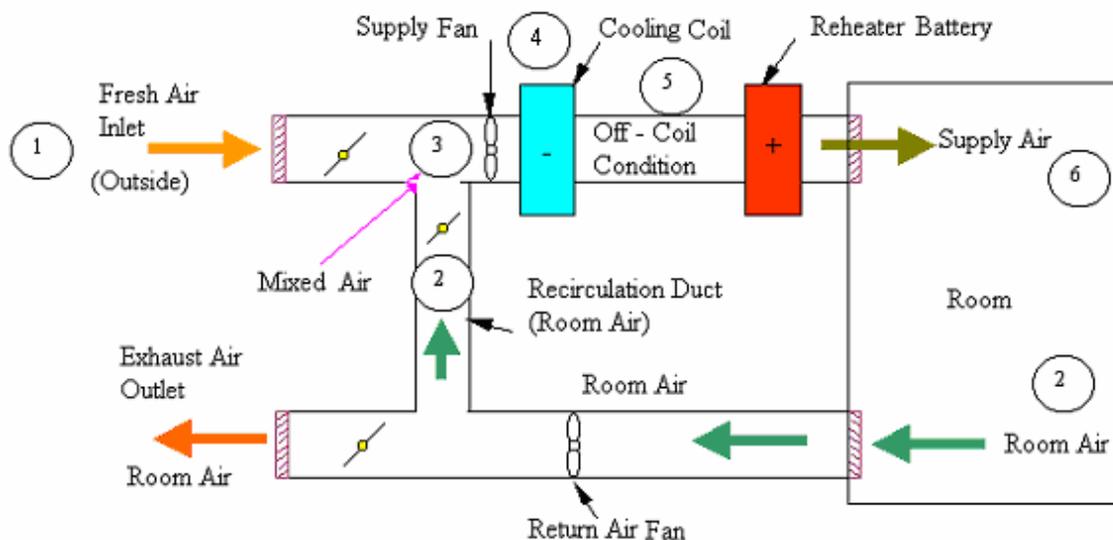
Here, return air from the conditioned space (2) mixes with required amount of outdoor air (1) at point (3) for acceptable indoor air quality and energy saving. The mixture is then cooled and dehumidified to ADP (4) at the cooling coil. The actual off-coil condition is represented as (5) due to inefficiency (bypass) of cooling coil. The conditioned air is supplied to the room through the supply fan, supply duct, and ceiling diffuser (6). Supply air then absorbs the sensible and latent load from the space, becoming the space air (2). Room air is returned back to the cooling

unit again and forms a closed cycle. Part of the return air is exhausted to balance the outdoor air intake and infiltration.

The psychrometric process is illustrated below for **summer** cycle.
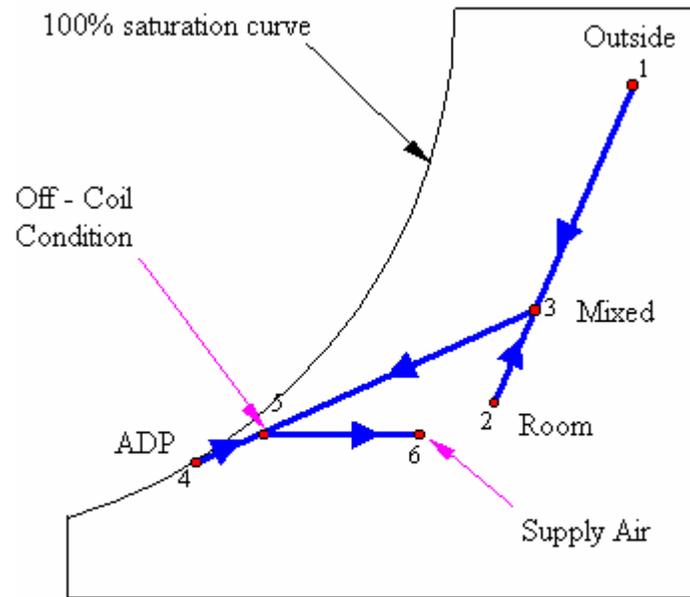
PSYCHROMETRIC CHART SHOWING TYPICAL
SUMMER CYCLE

You may notice that the off-coil condition is represented by node 5 away from ADP point 4. This is because the cooling coil is not 100% efficient and some of the air going through a cooling coil does not come into contact with the tubes or fins of the cooling coil and is therefore not cooled to the ADP temperature.

A mixing process therefore takes place as two air streams mix downstream of the cooling coil as shown below.
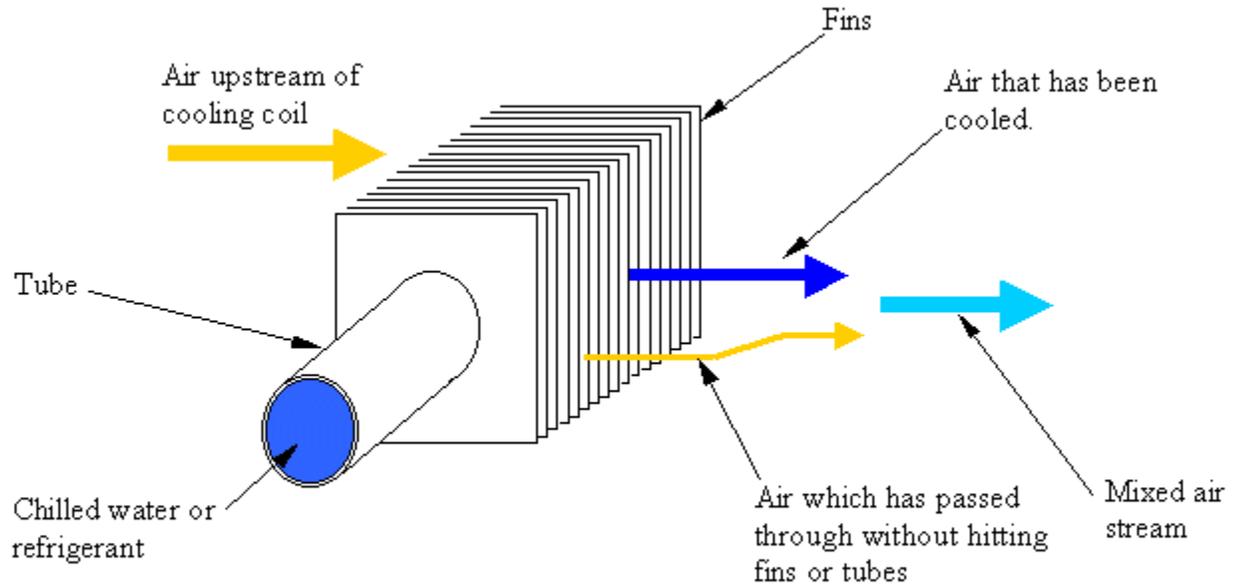
A SECTION OF COOLING COIL SHOWING AIR STREAMS

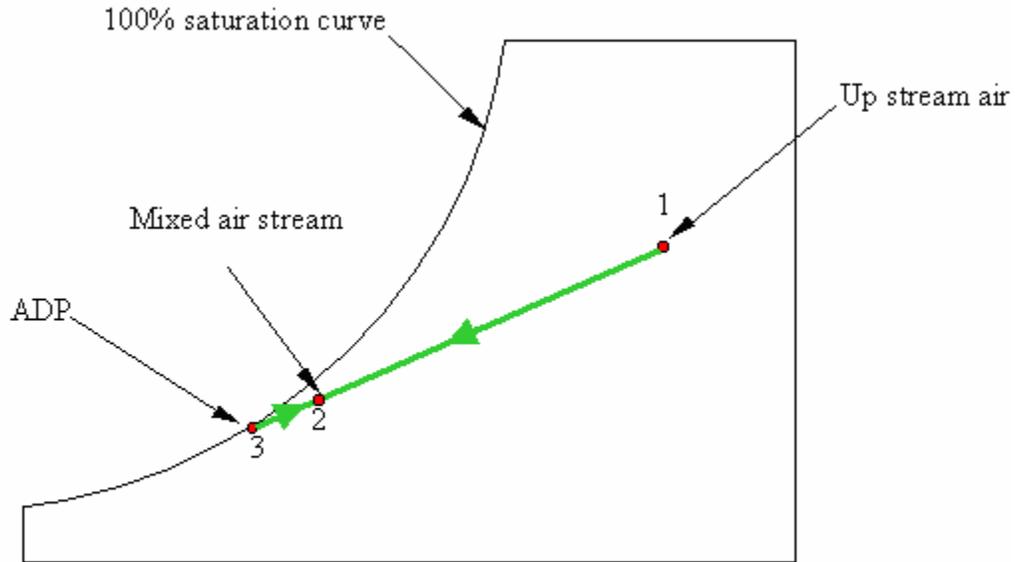One air stream is cooled down to the ADP and the other air stream by-passes the coil surfaces to give an off-coil air temperature (mixed air stream) a little higher than the ADP.

This may be looked upon as an inefficiency of the coil and is usually given as the cooling coil contact factor.

The process is shown on the psychrometric chart below.

PSYCHROMETRIC CHART SHOWING COOLING COIL
CONTACT FACTOR

The contact factor of a cooling coil may be found from;

$$\text{Contact Factor} = \frac{h_1 - h_2}{h_1 - h_3}$$
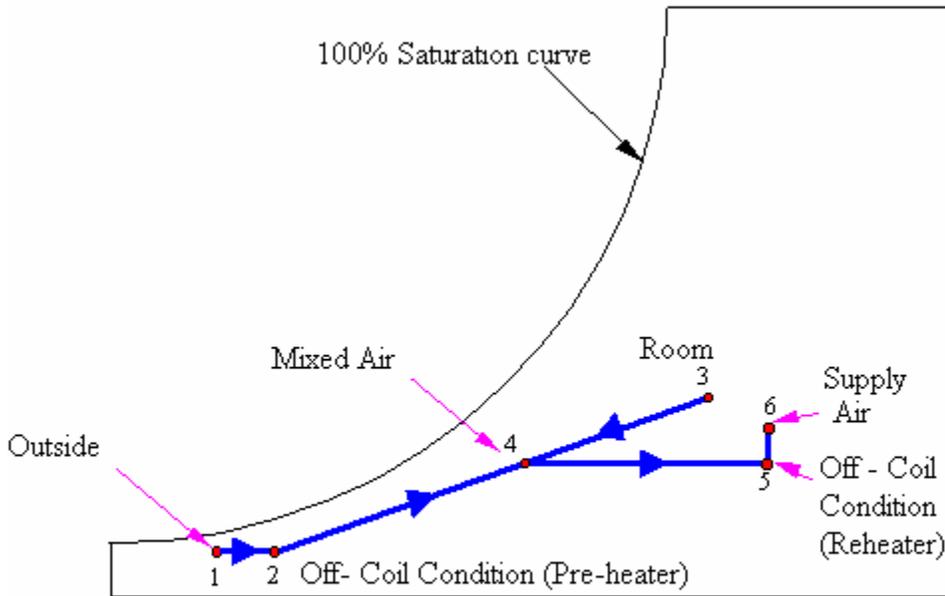
Another expression for contact factor is;

$$\text{Contact Factor} = \frac{\text{Distance 1 to 2}}{\text{Distance 1 to 3}}$$

WINTER MODE HEATING PROCESS

The schematic diagram below shows a typical plant system for **winter** heating.
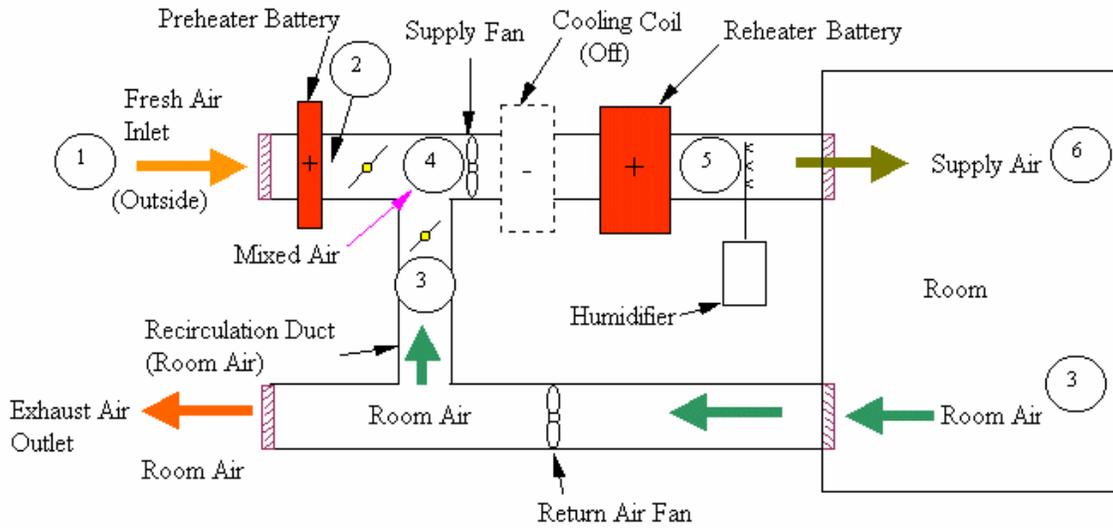
The psychrometric diagram below shows a typical **winter** cycle.

PSYCHROMETRIC CHART SHOWING TYPICAL
WINTER CYCLE

DESIGN SUPPLY VOLUME FLOW RATE

Design supply volume flow rate is estimated to determine the size of fans, grills, outlets, air-handling units, and packaged units.

For most comfort air-conditioning systems, design supply volume flow rate (V) in cfm is calculated on the basis of the capacity to remove the space cooling load at summer design conditions to maintain a required space temperature T_R:

$$V = \frac{Q_{Total}}{[60 * \rho * (h_R - h_S)]} = \frac{Q_{Sensible}}{[60 * \rho * C_p * (T_R - T_S)]}$$

Where

- V = design supply volume flow rate in CFM

- Q_{Total}, $Q_{Sensible}$ = Design space cooling load and design sensible cooling load, Btu/hr.

- C_p, ρ = Specific heat and air density. C_p is usually considered constant. Air density ρ may vary with the various types of air systems used.

- T_R, h_R = Room temperature (normally taken as 75°F for comfort applications) and room enthalpy at defined indoor DBT and WBT

- T_S, h_S = Supply air temperature leaving the cooling unit, supply air enthalpy at defined leaving DBT and RH

The following key points must be noted:

- A greater ρ means a smaller airflow rate (CFM) for a given supply mass flow rate.

- Greater the cooling load or sensible heat gain, $Q_{Sensible}$, the higher will be airflow rate (CFM).

- For a given $Q_{sensible}$, the supply temperature differential $\Delta T = (T_R - T_S)$ is an important parameter that affects V. Conventionally, a 15 to 20°F ΔT is used for comfort air-conditioning systems. The selection of temperature differential $(T_R - T_S)$ is determined by the laws of psychrometrics governing the performance of air system. As a rule of thumb, following provide an indicative relationship between the sensible heat ratio (SHR) and leaving air temperature and therefore $(T_R - T_S)$. Once the $(T_R - T_S)$ is known, the supply airflow rate (CFM) can be calculated:

Sensible Heat ratio V/s $(T_R - T_S)$			
SHR	**Ts**	**Room DB (T_R)**	**($T_R - T_S$)**
SHR < .80	54	75	21
.80< SHR>.85	56	75	19
SHR >.85	58	75	17

Where:

- The sensible heat ratio (SHR) is: = (Sensible load)/(Sensible load + Latent load)

- LAT is the leaving air temperature

- TD is the temperature difference between the room temperature and the leaving air temperature (LAT from the air handler).

[Note - Recently, a 28 to 34°F temperature differential $(T_R - T_S)$ has been adopted for cold air distribution. When $(T_R - T_S)$ has a nearly twofold increase, there is a considerable reduction in airflow rate (CFM) and fan energy use and saving in investment on ducts, terminals, and outlets].

- The summer cooling load is often greater than the winter heating load, and this is why Q_{Total} or $Q_{Sensible}$ is used to determine airflow rate (V) except in locations where the outdoor climate is very cold.

SECTION – 3   MODES OF HEAT TRANSFER

All the materials that are used in the construction absorb and transfer heat. By knowing the resistance of heat flow through a building component (R-value), you can calculate the total amount of heat entering the building. The basic equation to determine the heat loss or gain by conduction is known as Fourier's Law and is expressed as:

$Q = k \, x \, A \, x \, \Delta T \, / \, t$

Where

- Q = Hat transferred per unit time (Btu/hr)

- A = Heat transfer area (ft^2)

- k = Thermal conductivity of the material (Btu/ (hr $^\circ$F ft^2/ft))

- ΔT = Temperature difference across the material (°F)

- t = material thickness (ft)

For example, the heat transfer in 24 hours through 2 sq-ft. of material, 3" thick, having a thermal conductivity factor of 0.25, with an average temperature difference across the material of 70°F would be calculated as follows:

$$Q = \frac{0.25(k) \times 2 \text{ sq. ft} \times 24 \text{ hours} \times 70^\circ \, \Delta T}{3} = 280 \text{ BTU}$$

Before we go further, let's refresh few basic heat transfer fundamentals.

Modes of heat transfer

- Mainly involves conduction, convection, and radiation.

- Evaporation is less important when there is no moisture involved.

For HVAC load calculation purposes, conduction and radiation are primarily considered. Conduction is the transfer of heat through an object and radiation is the transfer of heat through electromagnetic waves, in this case sunlight. Heat travels from hot to cold and construction materials resist the flow of heat through them differently. For example, heat passes through glass much easier than wood siding.

Heat flow through solids - Terminology

- **Conductivity:** Designated by **k**. Represents BTUH that flows through 1 sq-ft of material, 1 in. thick, when temperature drop through this material is $1^{o}F$. Thermal conductivity is expressed in (Btu-in/hr ft^2 °F). Materials with lower k-values are better insulators. Insulation materials usually have k-factors less than one and are reported at what is called mean temperature. To determine the mean temperature, measure the surface temperatures on both sides of the insulation, add them together and divide by two. "As mean temperatures rises, so does the k-factor"

- **Conductance:** Designated by **C**. Represents BTUH that flows through 1 sq-ft of material of a given or specified thickness when temperature drop through this material is $1^{o}F$. C factor is similar to 'k', except it is the rate of heat flow through an actual thickness of material, where 'k' is a factor per inch. The C-factor is the k-factor divided by the thickness of the insulation. The lower the C value, the better the insulator.

- **Resistivity:** It is an index of the tendency of a material to resist heat flow per inch of its thickness. Reciprocal of **k**. Designated by **r**. Thermal Resistivity is expressed in (hr-°F ft^2)/ (Btu in).

- **Resistance:** It is an index of the tendency of a material of given thickness to resist heat flow. Reciprocal of **C**. Designated by **R**. Note the relationship between resistance, conductance, resistivity and thermal conductivity below:

  R = 1/C = rx = x/k, where x = thickness

  The higher the R-value, the higher (better) the insulating value.

  R Values change as the thickness of the insulating material changes.

  R-value for material only deals with conductive heat transfer. Since the total heat transferred by conduction varies directly with time, area, and temperature difference, and varies inversely with the thickness of the material, it is readily apparent that in order to reduce heat transfer, the 'k' factor should be as small as possible, and the material as thick as possible.

- **U-value:** Overall coefficient of heat transmission of an assembly of materials. This a single, composite heat transfer coefficient that is used to correlate the overall rate of heat transfer with surface area and temperature difference. R-value and U-factor are the inverse of one another: $U = 1/R$. For example, a wall with a U-value of 0.25 would have a resistance value of R = I/U = 1/0.25= 4.0. Materials that are very good at resisting the flow of heat (high R-value, low U-factor) can serve as insulation materials. The Overall Coefficient of Heat Transmission is expressed in Btu/ (hr ^{0}F ft^2).

Example

Calculate the U factor of a wall composed of 2" of material having a 'k' factor of 0.80, and 2" of insulation having a conductance of 0.16.

U value is found as follows:

R total = 1/C + X1/k1 or

R total = 1/0.16 + 2/0.80

R total = 8.75

$U = 1/R$ or $1/8.75 = 0.114$ Btu/hr ft^2 °F

Once the U factor is known, the heat gain by transmission through a given wall can be calculated by the basic heat transfer equation. Assuming an area of 100 square feet wall with an inside temperature of 85°F and an outside temperature of 115°F, the heat transmission would be:

Q = U x A x TD

Q = 0.114 x 100 x 30

Q = 342 Btu/hr

Series heat flow

Building envelope is typically composed of various elements. A wall may be constructed of hardboard (facing outdoors), plywood (facing indoors) and sandwich insulation in between. When a building structure is composed of various layers of construction elements having resistances R1, R2, R3…. Rn, the overall resistance value is sum of all individual resistances for whole wall, internal air spaces, insulation materials and air films adjacent to solid materials. Individual R-values are used in calculating overall heat transfer coefficients.

For layered construction, with paths of heat flow in series, the total thermal resistance of the wall is obtained by:

$R_{Total} = R_1 + R_2 + ...$

Or $R_{Total} = 1/C + x_1/k_1 + x_2/k_2\ldots$

Where

- C   is the conductance

- x_1 is the thickness of material one

- x_2 is the thickness of material two

- k_1 is the thermal conductivity of material one

- k_2 is the thermal conductivity of material two

And the overall coefficient of heat transmission is:

$U = 1/R_{Total}$

Or

$$U = \frac{1}{R_i + R_1 + R_2 + \ldots R_o}$$

Where:

- R_i = the resistivity of a "boundary layer" of air on the inside surface.

- R_1, R_2 …= the resistivity of each component of the walls for the actual thickness of the component used. If the resistance per inch thickness is used, the value should be multiplied by the thickness of that component.

- R_o = the resistivity of the "air boundary layer" on the outside surface of the wall.

The formula for calculating the U factor is complicated by the fact that the total resistance to heat flow through a substance of several layers is the sum of the resistance of the various layers. The resistance to heat flow is the reciprocal of the conductivity. Therefore, in order to calculate the overall heat transfer factor, it is necessary to first find the overall resistance to heat flow, and then find the reciprocal of the overall resistance to calculate the U factor.

NOTE:

Note that in computing U-values, the component heat transmissions are not additive, but the overall U-value is actually less (i.e., better) than any of its component layers. The U-value is calculated by determining the resistance of each component and then taking the reciprocal of the total resistance. Thermal resistances (R-values) must first be added and the total resistance (R-Total) divided into 1 to yield the correct U-factor.

Correct:

$$U = \frac{1}{R_1 + R_2 + R_3 + \ldots + R_n} = \frac{1}{R_{Total}}$$

Incorrect:

$$U = \frac{1}{R_1} + \frac{1}{R_2} + \frac{1}{R_3} + \ldots + \frac{1}{R_n} = U_1 + U_2 + U_3 + \ldots + U_n$$

The total R-value should be calculated to two decimal places, and the total U-factor to three decimal places.

Example

Determine the U-value for a layered wall construction assembly composed of three materials:

1) Plywood, 3/4-inch thick ($R_1 = 3/4 \times 1.25 = 0.94$)

2) Expanded polystyrene, 2-inches thick ($R_2 = 2" \times 4.00 = 8.00$)

3) Hardboard, 1/4-inch thick ($R_3 = 0.18$)

4) $R_i = 0.68$ ("still" air)

5) $R_o = 0.17$ (15 MPH wind, winter conditions)

Thermal Resistance of Composite Wall
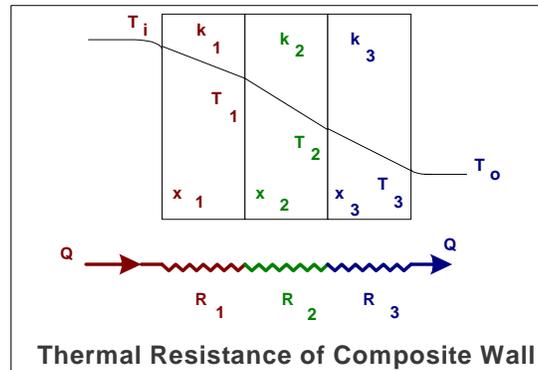
The U-values is:

$$U = \frac{1}{R_i + R_1 + R_2 + R_3 + R_{oo}}$$

$$= \frac{1}{0.68 + 0.18 + 8.00 + 0.94 + 0.17}$$

$$= \frac{1}{9.97} = 0.10 \frac{BTU}{hr. - sq. ft. - °F}$$

To calculate heat loss for say for 100 square feet of wall with a 70° F temperature difference would be:

$$Q = (.10)(100)(70) = 700 \ BTU/ HR$$

In the calculations above the TD is taken as 70°F, which is temperature difference between indoor and outside air. If the sun shines on a wall or roof of a building and heats the surface much hotter than the air (as typical in the summer), the heat

flow through the wall or roof would be greatly influenced by the hot surface temperature; hence, use a surface temperature rather than air to obtain a more realistic heat flow rate. Similarly, when calculating the heat flow through a floor slab resting on the ground, there will not be an air boundary-layer resistance underneath (Ro = 0) and the temperature (t_o) will be the ground temperature (not the outside air temperature).

Parallel Heat Flow

Average transmittances for parallel paths of heat flow may be obtained from the formula:

$$U_{avg} [A_A (U_A) + A_B (U_B) +...] / A_t$$

Or

$$U_{avg} = [1/ (R_A /A_A) + 1/(R_B/A_B)...]/A_T$$

Where:

- A_A, A_B, etc. = area of heat flow path, in ft^2

- U_A, U_B, etc.= transmission coefficients of the respective paths

- R_A, R_B, etc.=thermal resistance of the respective paths

- A_T= total area being considered (A_A+A_B+...), in Ft^2

Such an analysis is important for wall construction with parallel paths of heat flow when one path has a high heat transfer and the other a low heat transfer, or the paths involve large percentages of the total wall with small variations in the transfer coefficients for the paths.

Heat flow through air

- Wind outdoors has to be taken into consideration.

- Air films both inside and outside building envelope affects heat flow.

- A combination of dead air spaces and reflective surfaces produce very efficient insulation materials.

SECTION – 4   HEAT GAIN AND HEAT LOSS IN BUILDINGS

Each building has a characteristic exterior air temperature, known as the balance point temperature, at which the building in use would be able to support thermal comfort without the need for a heating or cooling system. At the balance point temperature, which is strongly influenced by internal loads and envelope design, building heat gains and losses are in equilibrium so that an appropriate interior temperature will be maintained naturally and without further intervention. When the outside air temperature falls below the balance point temperature, heat losses through the building envelope will increase – and interior air temperature will drop unless heat is added to the building to compensate. A system that provides such additional heat is called a space (or building) **heating system**. When the outside air temperature exceeds the balance point temperature, heat gain through the building envelope will upset thermal equilibrium and cause the interior air temperature to rise. A system that removes such excess heat is called a **cooling system.**

There are two distinct components of the air conditioning load; (1) the sensible load (heat gain/loss) and (2) the latent load (water vapor gain).

Sensible Loads - Sensible heat gain is the direct addition of heat to a space,which shall result in increase in space temperatures. The factors influencing sensible cooling load:

- Solar heat gain through building envelope (exterior walls, glazing, skylights, roof, floors over crawl space)

- Heat flow from warmer surroundings (partitions that separate spaces of different temperatures)

- Ventilation air and air infiltration through cracks in the building, doors, and windows

- Heat flow into the space from energy consuming objects within the space; these objects usually include: Lights, Electrical and electronic appliances, cooking or kitchen appliances, Occupants within the space etc.

Latent Loads - A latent heat gain is the heat contained in water vapor. Latent heat does not cause a temperature rise, but it constitutes a load on the cooling equipment. Latent load is the heat that must be removed to condense the moisture out of the air. The sources of latent heat gain are:

- Ventilation air and air infiltration through cracks in the building, doors, and windows

- Moisture generated within the space from moisture generating objects. These objects usually include: occupants within the space (breathing), moisture generated by cooking or warming appliances, industrial or production machinery which evaporates water, housekeeping, floor washing etc.

The total cooling load is the summation of sensible and latent loads.

Solar radiation

Heat gains from the sun can lead to increases in internal temperatures beyond the limits of **comfort**. It is therefore necessary to determine the amount of **solar radiation** that is transmitted into buildings through; windows, walls, roof, floor and by admitting external air into the building.

Classification of solar radiation: Total solar heat quantity falling on a surface consists of (1) unshaded direct radiation, (2) unobstructed diffuse radiation from sky, and (3) reflected solar radiation from adjacent surfaces.

- Direct radiation: Radiation that reaches earth's surface direct from the sun. On a clear day radiation intensity may range from 400 to 450 BTUH per square foot of surface area.

- Diffuse radiation: Radiation that has been scattered or re-emitted. As solar radiation, making use of electromagnetic waves, passes through the atmosphere, a portion of the energy is reflected, scattered, and absorbed by smoke, dust, gas molecules and water vapor. The sky looks blue because of scattering of the radiant energy corresponding to blue part of the electromagnetic spectrum, having shorter wavelengths; red at sunset results from the scattering of longer wavelengths corresponding to the red part of the spectrum.

- Solar heat gain through windows depends on its location on the earth's surface (latitude), time of the day, day of the year, and the direction it faces.

Formula for calculation of solar heat gain:

$$\textbf{BTUH = SHGF * A * SC}$$

Where

- SHGF = solar heat gain factor that specifies solar radiation in BTUH/sq-ft.

- A = glass area in sq-ft, and

- SC = shading coefficient that specifies the percentage of solar heat passing through the glass.

Some radiation may be absorbed by the ozone layer in the upper atmosphere and some by water vapor near the earth's surface.

HEAT FLOW THROUGH BUILDING ENVELOPE

Solar Load through Roof and Wall

Formula for calculation of sensible heat gain/loss through a building envelope (roof, walls and conduction through glass)

$$BTUH = U*A*TD$$

Where

- **Rate of heat flow (U)**

  o U-value for roof, wall, floor, etc. may be calculated by finding the resistance of component materials, air films, and internal air spaces.

  o For floors and walls in contact with the ground, use (1) a factor of 2 @temperature $10^{o}F$ or above, (2) a factor of 4 @temperature below $10^{o}F$.

  o Use overall **U**-values for different types of windows (residential or commercial) and not the center of the glazing.

[See 1997 ASHRAE Fundamentals, Chapter 24 or 2001 ASHRAE Fundamentals, chapter 25].

- **Area of construction (A)**

  o Calculate wall, roof, window, door, and floor areas from architectural drawings.

- **Temperature difference (TD)**

o Calculate the difference between inside and outside temperatures of the building envelope.

TD is OK for heat loss calculations but for cooling load calculations, another parameter CLTD = Cooling Load Temperature Difference is rather used to account for thermal mass dynamics. The actual equation for cooling load is

$$\textbf{BTUH = U * A * (CLTD)}$$

Where

o **CLTD** = Cooling Load Temperature Difference (in °F) for roof, wall or glass. For winter months CLTD is $(T_i - T_o)$ which is temperature difference between inside and outside. For summer cooling load, this temperature differential is affected by thermal mass, daily temperature range, orientation, tilt, month, day, hour, latitude, solar absorbance, wall facing direction and other variables and therefore adjusted CLTD values are used. Refer 1997 ASHRAE Fundamentals, Chapter 28, tables 30, 31, 32, 33 and 34.

The thermal storage concept is discussed further, later in this section.

Solar Load through Glazing

The treatment of heat transfer through window glass, skylights and plastic sheets is different. Heat transfer through glazing is both conductive and transmission. It is calculated in two steps:

Step # 1

The equation used for sensible loads from the conduction through glass is:

$$\textbf{BTUH = U * A * (CLTD)}$$

Where

o U = Thermal Transmittance for roof or wall or glass. See 1997 ASHRAE Fundamentals, Chapter 24 or 2001 ASHRAE Fundamentals, chapter 25. (Unit- Btu/Hr Sq-ft °F)

o A = area of roof, wall or glass calculated from building plans (sq-ft)

o CLTD = Cooling Load Temperature Difference (in °F) for glass. Refer 1997 ASHRAE Fundamentals, Chapter 28, tables 30, 31, 32, 33 and 34.

Step # 2

The equation used for radiant sensible loads from the transparent/translucent elements such as window glass, skylights and plastic sheets is:

$$\textbf{BTUH = A * (SHGC) * (CLF)}$$

Where

o A = area of roof, wall or glass calculated from building plans (sq-ft)

o SHGC = Solar Heat Gain Coefficient. See 1997 ASHRAE Fundamentals, Chapter 28, table 35

o CLF = Solar Cooling Load Factor. See 1997 ASHRAE Fundamentals, Chapter 28, table 36.

Partitions, Ceilings & Floors

The equation used for sensible loads from the partitions, ceilings and floors:

$$\textbf{BTUH = U * A * (Ta - Trc)}$$

Where

- U = Thermal Transmittance for roof or wall or glass. See 1997 ASHRAE Fundamentals, Chapter 24 or 2001 ASHRAE Fundamentals, and Chapter 25.

- A = area of partition, ceiling or floor calculated from building plans

- Ta = Temperature of adjacent space (Note: If adjacent space is not conditioned and temperature is not available, use outdoor air temperature less 5°F)

- Trc = Inside design temperature of conditioned space (assumed constant)

Outdoor Air

Outside air is needed for 3 main reasons:

- Required to maintain Indoor Air Quality for the occupants (see ASHRAE Standard 62 for minimum ventilation requirements). A minimum air exchange is necessary to decrease carbon dioxide buildup and remove odors. Such air exchange may take place either through infiltration or ventilation.

- Required to maintain space pressurization and prevent infiltration.

- Required as make-up air to compensate for what is compulsorily exhausted from restrooms, pantry, and kitchen and janitor areas.

Ventilation

- Ventilation refers to air brought into building by choice, intentionally.

- May either be provided mechanically or by natural means.

- The outside air that is allowed to enter the building through ventilation must also be conditioned (heated, cooled, humidified, dehumidified, or cleaned).

- Rate of ventilation determined by the requirements based on number of occupants and their activities.

- Recommended levels calculated either in terms of CFM per person or CFM per square foot.

- Mechanical ventilation is not usually required for small, residential buildings; infiltration (some quantity of air infiltrates, if the building is not hermetically sealed) is sufficient to furnish enough air change for such buildings.

Infiltration

- Leakage around doors and windows and around walls due to outdoor air pressure.

- Occurs in an uncontrolled manner. Winds on the windward side of the building blow outside air into the indoor spaces, displacing conditioned air that leaves through similar opening on the leeward side of the building where low outside air pressure creates suction.

- The outside air that enters the building must be conditioned (heated, cooled, humidified, dehumidified, or cleaned).

- Two methods for calculation of infiltration in buildings: crack length method and air change method.

- **Crack length method:** Expresses air flow rate per unit length of crack (e.g. a metal framed hinged window may have an infiltration of 0.35 CFM per

foot of crack); requires specific information about the dimensions and construction details of doors, windows, or other openings.

- o The above method can also be expressed in terms of air flow rate per unit area of door or window (e.g. an operable glass window has a rate of infiltration of 0.5 CFM per square foot of the window area; the rate is about 0.25 CFM per square foot for fixed glass windows).

- o The quantity of infiltration depends on the kind or width of crack and the pressure difference due to wind velocity. It is common to use a wind velocity of 15 mph for normal design conditions.

- **Air change method:** This method takes into account total replacement of inside air by outdoor air in terms of a unit called air change.

  - o When construction method and related crack lengths cannot be accurately evaluated, the air change method may be used as an alternative.

  - o An air change having a value of 1 per hour means that all air in a building is replaced by outside air every hour.

  - o Formula used for conversion of air change rate to CFM:

(Volume of building)*(Air change rate)/60 = CFM.

Calculation of Heat Gain / Heat Loss due to outside air (OA)

Heat gain/heat loss due to OA has two components: sensible and latent.

- Formula for calculation of sensible heat due to OA:

BTUH = 1.08*CFM*TD

- Formula for calculation of latent heat due to OA:

$$\text{BTUH} = 0.68 * \text{CFM} * \text{GD}$$

- Formula for calculation of total heat due to OA:

$$\text{BTUH}_{total} = 4.5 * \text{CFM} * \text{ED}$$

Explanation of variables:

Value of CFM to be used in calculation

- Outside air rate both due to infiltration and ventilation should be found out. The rate which is larger should be used in the calculation of heating/cooling load.

Value of TD to be used in calculation

- TD is the temperature difference of the outside air dry-bulb temperature (°F) and the dry-bulb temperature of air leaving the cooling coil (°F). It is also represented as $(T_o – T_c)$.

Value of GD to be used in calculation

- GD is the difference of the humidity ratio of outside air (grains of moisture) and humidity ratio of air leaving the cooling coil (grains of moisture). It is also represented as $(w_o – w_c)$.

Value of ED to be used in calculation

- ED is the difference in enthalpy of outside air (Btu per lb of dry air) and enthalpy of air leaving the cooling coil (Btu per lb of dry air). It is also represented as $(h_o – h_c)$.

Some explanations of the constants used in OA heat transfer models

- Specific Heat: Quantity of heat required to raise the temperature of one pound of a material by 1°F. Specific heat of air is 0.24 BTU/lb/1°F.

- Why a constant 1.08 in the sensible heat transfer model?

    o At a rate of 1 CFM airflow, there are 60 such units in an hour (the hourly rate of airflow is required to be known because heat flow is calculated in terms of per hour). 60 CFM of air = 4.5 lbs of air.

    o Heat required to raise the temperature of 4.5 lbs of air = 0.24*4.5 = 1.08 BTU/lb/^{o}F.

- Why a constant of 0.68 in the latent heat transfer model?

    o Heat released in condensation of one pound of water = 1060 BTU.

    o Heat released in condensation of one grain of water = 1060/7000 BTU.

    o Assuming a moisture difference of one grain per pound between inside and outside air, for 60 CFM of air there will be difference of 4.5 grains. Heat released in condensation of that amount of water = (1060/7000)*4.5 = 0.68 BTUH.

INTERNAL LOADS

People

The human body generates heat within itself through the process of metabolism, and releases it by conduction, convection, radiation, and evaporation from skin and clothing; and by convection and evaporation in the respiratory tract. While part of this heat gain due to evaporation of moisture is latent, the rest is sensible. The proportion of heat released in the form of sensible to latent heat depends on

the ambient dry bulb temperature and moisture content of the surrounding air. The total amount of heat gain due to this factor depends on number of people, duration of occupancy, and activity level. Both sensible and latent heat components should be calculated using following equations:

- The equation used in estimating sensible load due to occupants is:

$$\textbf{BTUH}_{\textbf{sensible}} = \textbf{N} * (\textbf{Q}_\textbf{S}) * (\textbf{CLF})$$

- The equation used in estimating latent load due to occupants is:

$$\textbf{BTUH}_{\textbf{latent}} = \textbf{N} * (\textbf{Q}_\textbf{L})$$

Where

- N = number of people in space.

- Q_S, Q_L = Sensible and Latent heat gain from occupancy is given in 1997 ASHRAE Fundamentals Chapter 28, Table 3

- CLF = Cooling Load Factor, by hour of occupancy. See 1997 ASHRAE Fundamentals, Chapter 28, table 37.Note: CLF = 1.0, if operation is 24 hours or of cooling is off at night or during weekends.

Lighting

Electric light fixtures convert electrical power into heat and light. Even light is eventually converted into its equivalent heat energy in the space; so, in effect, all electrical power entering a light fixture ends up as heat in a space. The lights results only in sensible heat gain and is given be equation:

$$\textbf{BTUH} = \textbf{3.41} * \textbf{W} * \textbf{F}_{\textbf{UT}} * \textbf{F}_{\textbf{BF}} * (\textbf{CLF})$$

Where

- W = Installed lamp watts input from electrical lighting plan or lighting load data

- F_{UT} = Lighting use factor, as appropriate

- F_{BF} = Blast factor allowance, as appropriate

- CLF = Cooling Load Factor, by hour of occupancy. See 1997 ASHRAE Fundamentals, Chapter 28, Table 38. *Note: CLF = 1.0, if operation is 24 hours or if cooling is off at night or during weekends.*

[Simple method of estimating heat gain due to lighting: BTUH = 3.4 * watts/sq-ft * floor area].

Equipment

There is almost an infinite variety of equipment that contributes to heat gain to the conditioned space. The designer has to be thorough in identifying all the heat-producing equipment in a given space. In order to calculate heat gain due to the effect of equipment, the formula for lighting may be used as an approximation. Three different equations are used under different scenarios:

- Heat gain of power driven equipment and motor when both are located inside the space to be conditioned

$$Q = 2545 * (P / Eff) * F_{UM} * F_{LM}$$

Where

- P = Horsepower rating from electrical power plans or manufacturer's data

- Eff = Equipment motor efficiency, as decimal fraction

- F_{UM} = Motor use factor (normally = 1.0)

- F_{LM} = Motor load factor (normally = 1.0)

- Note: F_{UM} = 1.0, if operation is 24 hours

- Heat gain of when driven equipment is located inside the space to be conditioned space and the motor is outside the space or air stream

$$Q = 2545 * P * F_{UM} * F_{LM}$$

Where

- P = Horsepower rating from electrical power plans or manufacturer's data

- Eff = Equipment motor efficiency, as decimal fraction

- F_{UM} = Motor use factor

- F_{LM} = Motor load factor

- Note: F_{UM} = 1.0, if operation is 24 hours

- Heat gain of when driven equipment is located outside the space to be conditioned space and the motor is inside the space or air stream

$$Q = 2545 * P * [(1.0\text{-}Eff)/Eff] * F_{UM} * F_{LM}$$

Where

- P = Horsepower rating from electrical power plans or manufacturer's data

- Eff = Equipment motor efficiency, as decimal fraction

- F_{UM} = Motor use factor

- F_{LM} = Motor load factor

- Note: F_{UM} = 1.0, if operation is 24 hours

Kitchen appliances

The total residential equipment loads can be approximated by the major appliances in the kitchen. A value of 1200 BTUH may be used as heat gain from residential kitchen appliances. (Note: for commercial buildings, you do not have to calculate heat gain from kitchen appliances separately if you have already calculated the heat gain from equipment.)

HEAT GAIN/LOSS IN BUILDINGS: OTHER FACTORS

Outdoor Design Conditions

The capacity calculations for the HVAC systems shall be based on the climatic information contained in chapter 26 of 1997 ASHRAE Handbook of fundamentals. Design conditions for the United States appear in Table 1a and 1b; for Canada in Tables 2a and 2b, and the international locations in Tables 3a and 3b.

Table A-Winter Conditions:

- The information provided in table 1a, 2a and 3a are for heating design that include:

  o Dry bulb temperatures corresponding to 99.6% and 99% annual cumulative frequency of occurrence,

  o Wind speeds corresponding to 1%, 2.5% and 5% annual cumulative frequency of occurrence,

o Wind direction most frequently occurring with 99.6% and 0.4% dry-bulb temperatures and

o Average of annual extreme maximum and minimum dry-bulb temperatures and standard deviations.

Note that the percentages reflect annual cumulative frequency of occurrence of outdoor weather conditions. No humidity data is provided for winter. Assume 100% RH

Table B-Summer Conditions

The information provided in table 1b, 2b and 3b are for cooling and humidity control that include:

o Dry bulb temperature corresponding to 0.4%, 1.0% and 2.0% annual cumulative frequency of occurrence and the mean coincident wet-bulb temperature (warm). [This implies that 0.4%, 1.0% and 2.0% of the time in a year, the outdoor air temperature will be above the design condition.] These conditions appear in sets of dry bulb (DB) temperature and the mean coincident wet bulb (MWB) temperature since both values are needed to determine the sensible and latent (dehumidification) loads in the cooling mode.

o Wet-bulb temperature corresponding to 0.4%, 1.0% and 2.0% annual cumulative frequency of occurrence and the mean coincident dry-bulb temperature

o Dew-point temperature corresponding to 0.4%, 1.0% and 2.0% annual cumulative frequency of occurrence and the mean coincident dry-bulb temperature and humidity ratio (calculated for the dew-point

temperature at the standard atmospheric pressure at the elevation of the station).

o Mean daily range (DR) of the dry bulb temperature, which is the mean of the temperature difference between daily maximum and minimum temperatures for the warmest month (highest average dry-bulb temperature). These are used to correct CLTD values.

Selecting Appropriate Outdoor Design Conditions

We have three Cumulative Achieved Periods for summer, two for winter. We have to decide which are appropriate for design. Statistically we expect these design conditions to be exceeded as follows:

0.004*365 day/yr = 1.46 day/yr = 35 hrs/yr.

0.01*365 day/yr = 3.65 day/yr = 87.6 hrs/yr.

0.02*365 day/yr = 7.30 day/yr = 175.2 hrs/yr.

In effect, we are asked to make a cost/benefit choice in any design. Normally we do not design a building with a cooling/heating so large that cooling/heating loads can be met for temperatures which may occur only once in a hundred years. This would provide an expensive design and with operational capabilities which are unlikely to be needed.

In section – 1 "Comfort", we noted that no single temperature/humidity combination satisfies everyone. Based on the studies on Predicted Mean Vote (PMV) studies, the designer should select conditions that will be considered tolerable for particular area of application. Here are some examples:

o If we are designing a 1 room apartment for college students low costs may be the major consideration. In this case we may choose to design

so that the system fails to meet normal comfort conditions 2% of the time. (7.3 days/yr.)

- o If we are designing a house for a middle income family, cost is still a factor but the family may be less willing to compromise comfort. Here we may choose to design which fails to meet design conditions 1% of the time (3.65 days/yr)

- o If we are designing a computer room for an expensive main-frame computer, we may want to provide maximum "comfort" for the system. Here we would design for the 0.4% design condition. (1.7 days/yr.)

- Winter design conditions shall be based on the 99.6 percent column dry bulb temperature in the ASHRAE Fundamentals Volume.

- For water-cooled heat rejection equipment (such as cooling tower) use ASHRAE 1% wet bulb and for air-cooled use ASHRAE 1% dry bulb conditions.

- Base the selection of evaporative equipment on the 2.5% wet bulb temperature. For applications where maintaining indoor temperature or humidity conditions is critical, the designer may use the corresponding 1.0% temperatures.

- For the selection of condensers and condensing units that will be subjected to unusually high radiation heat gain, add 3°C (5°F) to the dry bulb temperature specified above.

- Outdoor design temperature for heating to prevent freezing conditions should be based upon the 99% dry bulb temperature.

- Special purpose facilities will require special definition of appropriate interior design conditions.

Indoor Design Conditions

Indoor cooling and heating design temperatures should be based upon local energy codes. Design conditions for specific occupancies are specified in ANSI/ASHRAE 55-1992.

For most of the comfort systems, the recommended indoor temperature and relative humidity are:

- Summer: 73 to 79°F; The load calculations are usually based at 75ºF dry bulb temperatures & 50% relative humidity

- Winter: 70 to 72°F dry bulb temperatures, 20 - 30 % relative humidity

These are design conditions and not operating limits. All thermostats shall be adjustable between 60 to 85°F).

Thermal mass dynamics

Heat Gain: The heat gain for a building is a simultaneous summation of all external heat flows plus the heat flows generated inside the building. The heat gain varies throughout the 24 hours of the day, as the solar intensity, occupancy; lights, appliances etc keep varying with time.

Cooling load: The cooling load is an hourly rate at which heat must be removed from a building in order to hold the indoor air temperature at the design value. In other words, cooling load is the capacity of equipment required to account for such a load.

Theoretically, it may seem logical to address that the space heat gain is equivalent to space cooling load but in practice **"Heat gain ≠ cooling load."**

The primary explanation for this difference is the time lag or thermal storage affects of the building elements. Heat gains that enter a building are absorbed/stored by surfaces enclosing the space (walls, floors and other interior elements) as well as objects within the space (furniture, curtains etc.) These elements radiates into the space even after the heat gain sources are no longer present. Therefore the time at which the space may realize the heat gain as a cooling load is considerably offset from the time the heat started to flow. This thermal storage effect is critical in determining the instantaneous heat gain and the cooling load of a space at a particular time.

Here are few definitions related to Heat absorption

Radiant heat: A large portion of heat gain by an envelope is radiant that does not immediately heat up a space; it must first strike a solid surface and be absorbed and then radiated back into the space.

Temperature difference: As radiant heat strikes a solid surface, some of it is absorbed, raising the temperature at the surface above that inside the material and above that of the air adjacent to the surface. This temperature difference causes heat flow into the material.

Thermal storage: Heat conducted into the material is stored, and heat convected into the air becomes heat gain to the space. The proportion of radiant heat being stored depends on (1) the ratio of the thermal resistance into the material to the thermal resistance of the air film, and (2) the temperature difference. When the temperatures of the air and solid material start out equal, the majority of the absorbed heat is stored in the material. As the process of storing radiant heat

continues, the inside of the material becomes warmer, which slows down the rate of heat transfer and storage of heat.

Thermal capacity: Heat is stored in building materials and furnishings the same way that sponge soaks up water. The capacity of a material to store heat depends upon the quantity (mass) present and its characteristic specific heat (Btu/lb-°F). As mass gets warmer, its capability to store heat decreases, just as a sponge cannot absorb any more water when it is saturated. The magnitude of the heat storage potential of a material depends on its thermal or heat-holding capacity. The thermal capacity of a material is its weight times its specific heat, and thus is directly proportional to its weight:

$$\textbf{Thermal capacity = specific heat * weight}$$

Initial cooling load: In an air-conditioned space, the accumulation of heat during the day is stored in the building envelope. When the cooling system is shut off during the night, transmission of heat through the building envelope continues either into or out of the building, depending on the outside air temperature. If the night time outdoor temperature is cooler than the indoor temperature, the heat may be lost to the surrounding environment, so heat stored in the building envelope will be gradually released outside. When the outside temperature is warmer than the indoor temperature, heat continues to build up in the space, some of it flowing into the solid materials, where it is stored. This represents an initial cooling load of the air-conditioning equipment when it is turned on again.

Initial heating load: During heating season, heat is discharged from the solid materials overnight when the thermostat is set back. The mass must then absorb heat during a warm-up period to raise its temperature. The bulk of the heating load during the warm-up period is attributable to the collective mass.

Mass arrangement of buildings: The magnitude and configuration of the mass are also important factors that affect the heat storage capacity of buildings. The building can be so designed as to store heat from the sun and the occupants that can be used to provide heat at night. In commercial buildings, the thermal lag can be designed to last overnight until morning so that the building doesn't have to be warmed back after setting back the thermostat at night. The heat storage and subsequent rate of release into or out of the building depend on the thermal storage capacity and the magnitude of the indoor-outdoor temperature difference.

- Thermal lag: If the building envelope contains a large quantity of mass, it will store a large quantity of heat and cause a delay in heat transmission. This delay is called thermal lag. The more the mass, the longer the delay. Thermal lag can be of several hours or even several days.

Determination of the need for thermal storage: The desirability of high or low thermal storage mass depends on the climate, site, interior design condition, and operating patterns. For example, high thermal storage mass is advantageous when outdoor temperature swings widely above and below recommended indoor temperature.

Moisture Transfer in Building Envelope

In most comfort air-conditioning systems, usually only the space temperature is controlled within limits. A slight variation of the space relative humidity during the operation of the air system is often acceptable. Therefore, the store effect of moisture is ignored except in conditioned spaces where both temperature and relative humidity need to be controlled or in a hot and humid area where the air system is operated at night shutdown mode. In most cases, latent heat gain is considered equal to latent cooling load instantaneously.

Moisture problems

Air leakage and its associated water vapor infiltrate or exfiltrate through the cracks, holes, and gaps between joints because of poor construction of the building envelope. The driving potential of this air leakage and associated water vapor is the pressure differential across the building envelope.

Types of problem: Problems involving moisture may arise from changes in moisture content, the presence of too much moisture, or the problems associated with its change of state. Of particular interest is the change of the vapor to either liquid or to solid state, called condensation. Specifically, there are three types of moisture problems in building construction: (1) visible condensation, (2) concealed condensation, and (3) extremely low humidity.

Visible condensation: Occurs when any interior surface is colder than the dew point of the nearby air. Once the air temperature drops below its dew point, moisture is released on the surface. The loss of moisture from the air causes a reduction of vapor pressure which then draws moisture in from the surrounding air. This moisture also condenses, causing a continuous migration of moisture toward the cold surface. Possible solutions to visible condensation problem are:

- Insulate interior surfaces, cold water pipes and ductwork

- Provide enough air motion

- Reduce the moisture content of air

- Ventilate moist air out of the space

- Reduce ventilation rates when the dew point is higher outside than inside

- Artificially warm the cold surfaces

- Mechanically dehumidify the air

Concealed condensation: Is caused by a higher concentration of water vapor inside than outside. When there is a higher concentration of water vapor inside then outside, the vapor pressure is also higher on the inside. This represents a driving force causing a diffusion of moisture through the building envelope. The build-up of vapor pressure within the building is a function of the amount of vapor in the air, its inability to escape, and the air temperature.

- **Effects:** Condensation within the envelope construction leads to deterioration of materials, paint peeling, and insulation.

- **Remedy:** 1) dehumidify the interior space; 2) provide tightly sealed vapor barriers (should be applied to the heated side); 3) ventilate the wall cavity.

Extremely low humidity: An extremely low RH can cause physical discomfort and may also cause damage to materials (e.g. shrinkage of porous materials, de-lamination of furniture and paneling, cracks in wood, loosening of joints, etc.). The minimum RH that should be normally maintained is 20 percent.

SECTION – 5   **HVAC NOISE & VIBRATION**

Noise can have a significant impact upon the comfort and welfare of occupants. Unwanted noise makes a workplace uncomfortable and less productive. When people are surveyed about workplace comfort, their most prevalent complaints involve the heating, ventilating and air-conditioning (HVAC) systems. The problems they cite most frequently, aside from temperature control, have to do with excessive noise and vibration.

The HVAC industry has established noise criteria (NC) values for evaluating the acceptability of sound levels. NC values for different types of buildings range from 30 to 40 decibels. A decibel is a unit of comparative sound measurement (a whispered conversation at a distance of 6 ft. from the ear, for example, has a sound pressure level of 30 decibels). The noise level and vibration transmission can be partially controlled by mechanical isolation, shields, baffles, and acoustical liners. Massive concrete pads and vibration isolators are placed between the equipment and the building to avoid transmission of vibration and noise through the structure. Equipment rooms can be acoustically isolated if necessary.

Architectural acoustics

Acoustics is the science of sound. It relates to recorded music, to speech and hearing, to the behavior of sound in concert halls and buildings, and to noise in our environment. It is the technology of designing spaces and systems that meet our auditory needs. Architectural acoustics deals with sound in and around buildings of all kinds. Good acoustical design ensures the efficient distribution of desirable sounds as well as the exclusion of undesirable sound. All acoustical situations consist of three parts: *(1) source, (2) Path, and (3) Receiver*.

SOUND

Sound may be defined as vibrations or pressure changes in an 'elastic' medium, such as air, water, most building materials, and earth. The terms noise and sound are often used interchangeably, but the term sound is generally used to describe useful communication or a pleasant sound; whereas the term noise is generally used to describe an unwanted sound which interferes with speech.

In examining building acoustics we are mainly concerned with sound vibrations through the air, whereas in the subject of noise control we are equally concerned with vibrations transmitted through solid materials such as pipe work, concrete plinths and building structures.

Cycle, period, and frequency of sound: A full circuit by a particle of a medium displaced by vibration is a *cycle*. Time required to complete one cycle is called the *period*. Number of complete cycles per second is the *frequency* of sound. Unit of frequency is *Hertz (Hz)*.

Wavelength: The distance a sound wave travels during one cycle of vibration. *Wavelength = Velocity of sound/Frequency of sound*.

Sound intensity: Sound travels freely in all directions (i.e. spherically). Sound intensity is the strength of sound per unit area of a spherical surface.

The Decibel Scale

Sound power (generating from an equipment say a fan) and sound pressure (received by the ear) are measured in dB (decibels). The decibel scale is used to measure sound intensity. In decibel scale,

- Minimum intensity of perceptible sound is given a value of 0

- Whole numbers are used and

- An increase of every ten units equals a doubling of loudness.

It is a logarithmic scale. When trying to calculate the additive effect of two sound sources, use the approximation as below (note that the logarithms cannot be added directly).

- o With two sound sources of equal dBs, the equivalent dB is the original plus 3 dB.

   (<u>Example</u>: 60 dB + 60 dB = 63 dB)

- o With two sound sources of unequal dBs, if difference is 10 dB or more, use higher dB.

   (<u>Example</u>: 70 dB + 60 dB = 70 dB)

When specifying noise criteria for HVAC equipment reference is made to the sound power level and not the sound pressure level.

Sound Power and Sound Pressure

The difference between sound power and sound pressure is critical to the understanding of acoustics. To understand the concept of sound power and sound pressure, an analogy can be made between a noise source and a light bulb. A light bulb is rated to dissipate a particular number of watts of power. The bulb will always dissipate the same amount of power independent of its surroundings or the environments in which it is located. As an example, a 60W light bulb consumes 60W no matter where it is screwed in. However, the same bulb may appear to brighten a room covered with shiny reflecting walls more than another with dull black walls. The effects of distance, volume of space, absorbing and reflecting surfaces etc. will combine to determine the resulting lighting level at any point.

This is an exact parallel to the acoustical situation. *Sound Power is the amount of acoustical power a source radiates in a non-reflective environment and is*

measured in Watts. Sound power is a fixed property of a machine irrespective of the distance and environment. On the other hand, Sound pressure is related to how loud the sound is perceived to be, and depends upon the distance from the source as well as the acoustical environment of the listener (room size, construction materials, reflecting surfaces, etc.). Thus a particular noise source would be measured as producing different sound pressures in different spots. Theoretically, the sound pressure "p" is the force of sound on a surface area perpendicular to the direction of the sound. The SI-units for the Sound Pressure are N/m^2 or Pa.

Inverse-square law: Sound intensity decreases at a rate inversely proportional to the square of the distance from the sound source. The relationship can be expressed as:

$$I = W/4\pi r^2$$

Where

- I = sound intensity in watts per square centimeter

- W = sound power in watts

- r = distance from the sound source in centimeter.

If the distance is measured in feet, $4\pi r^2$ has to be multiplied by 930 (because 1 square foot equals 930 square centimeters).

SOUND PROPAGATION

Direct: Reaches the receiver directly from the source.

Reflection: Occurs when sound waves bounce off a surface at the same angle at which it was incident on the surface.

Diffraction: It is the bending or flowing of a sound wave around an object or through an opening.

Diffusion: Scattering or random distribution of sound from a surface.

Reverberation: Persistence of sound after source of sound has ceased. Results from repeated reflections. Some reverberation is good (particularly for musical performances), but not always desirable. Intelligibility and subjective quality of sound is rated by reverberation time (RT).

Reverberation Time (RT): It is the time required for sound to decay 60 dB after the source has stopped producing sound. *Reverberation time = 0.05 * volume of room/total absorption of sound*. (Average ceiling height in spaces with upholstered seats and absorptive rear walls is approximately related to mid-frequency reverberation time. *Ceiling height ≈20 * Mid-frequency Reverberation Time in Seconds*.)

Echo: Distinct repetition of original sound clearly heard above the general reverberation. A reflected sound can be perceived as discrete echo if the reflected sound wave is heard 0.05 second or later after it was heard as a direct sound.

SOUND ABSORPTION

When sound energy strikes a surface, part of the energy is absorbed. Reverberation and echoes may be controlled by effective use of sound absorption quality of a surface. Acoustic absorption is defined in terms of an absorption coefficient. It is the ratio of absorbed sound intensity by a material to the intensity of the sound source.

Absorption coefficient = absorbed sound intensity/total intensity of sound source.

Total absorption by a surface = surface area * absorption coefficient

Unit of sound absorption is *Sabin*.

RAY DIAGRAM

Ray diagram is analogous to specular reflection of light. Analysis of ray diagrams can be used to study the effect of room shape on the distribution of sound and to identify surfaces that may produce echoes. A ray diagram shows both reflected and direct sound paths. The difference between these two paths is called path difference (Path Difference = Reflected Path - Direct Path). A path difference in excess of the distance that can be traveled by a sound wave in 0.05 seconds indicates that the reflected sound can be perceived as discrete echo.

WEIGHTING NETWORKS

A-weighting network: Generally, the sensitivity of human hearing is restricted to the frequency range of 20 Hz to 20,000 Hz. The human ear, however, is most sensitive to sound in the 400 to 10,000 Hz frequency range. Above and below this range, the ear becomes progressively less sensitive. To account for this feature of human hearing, sound level meters incorporate a filtering of acoustic signals according to frequency. This filtering is devised to correspond to the varying sensitivity of the human ear to sound over the audible frequency range. This filtering is called **A-weighting**. Sound pressure level values obtained using this weighting are referred to as A-weighted sound pressure levels and are signified by the identifier dBA. Simply speaking, it may be defined as a frequency-response adjustment of a sound-level meter that makes its reading conform, very roughly, to human response.

C-weighted network: The C-weighted network provides un-weighted microphone sensitivity over the frequency range of maximum human sensitivity (over 1000 Hz).

SOUND TRANSMISSION CLASS (STC) OF MATERIALS

STC is a single number rating of the air-borne transmission loss (TL) of a construction. It measures the sound transmission loss (TL) of a construction at one-third octave band frequencies.

- For measurement, analysis, and specification of sound, the frequency range is divided into sections or bands. One common standard division is into ten octave bands identified by their mid-frequencies: 31.5, 63, 125, 250, 500, 1000, 2000, 4000, 8000, and 16000.

The STC of a given material is determined by comparing its measured TL values against a standard STC contour using the following criteria:

- The maximum deviation of the test curve below the standard contour at any single test frequency shall not exceed 8 dB.

- The sum of deviations below the standard contour at all frequencies of the test curve shall not exceed 32 dB.

When the contour is adjusted to the highest value (in integral dB) that meets the above requirements, the STC of the material would be the TL value corresponding to the intersection of the standard STC contour and 500 Hz frequency ordinate.

Noise from Ventilation and AC Systems

The HVAC noise that ultimately reach living/ working quarters are made up of

- Low-frequency fan noise - fans generally produce sound in 16-Hz through 250-Hz octave bands; Variable-air-volume (VAV) boxes noise is usually in the 125-Hz through 500-Hz octave frequency bands.

- Mid-frequency airflow or turbulence-generated noise - Velocity noise from airflow and turbulence in a duct ranges from 31.5 Hz through 1,000 Hz.

- High-frequency damper and diffuser noises - Diffuser noise usually contributes to the overall noise in the 1,000-Hz through 4,000-Hz octave bands.

Typical mechanical system noise is made up of a variety of noise components. Therefore the HVAC designer has to deal with noise control at various frequencies. Noise in any HVAC system can be measured as a function of frequency in decibels, typically divided into eight octave bands. That result is then compared against a criteria curve, such as the noise criterion (NC) family of curves. "Private offices are typically designed for NC 30 to 35, while open offices are designed for NC 40 to 45. Experience has shown that sound levels in offices above NC 45 can start to create complaints.

HVAC Design Tips for Acoustically Sensitive Spaces

- **Construction Materials:** Construct wall and ceiling assemblies to absorb noise and lower reverberation time. You have to at least double the absorption in a space before there is a noticeable difference. Every time you double the absorption, the reverberant noise field is reduced by 3 dB, which is classified as "just perceptible".

  Use high-STC (sound transmission class) materials to keep noise out. When the mass of a barrier is doubled, the isolation quality (or STC rating) increases by approximately 5 dB, which is clearly noticeable.

- **Penetrations:** Make sure all penetrations through sound isolation walls and ceilings are sealed resiliently and airtight.

- **HVAC Equipment Location:** Locate equipment that generates noise and vibration as far away as practical from acoustically sensitive spaces. If an

air handler is too close to a critical space, the risk of excessive noise or vibration via every other potential path is greatly magnified.

- **Variable Speed Options:** Make sure fan motor can handle the speed change. Lowering fan speeds and fan modulation provides opportunities to adjust noise levels.

- **Duct Silencers:** Sound attenuators are an effective means of reducing broadband noise as it travels down a duct system, and have the advantage of predictable performance. Locate silencers far enough upstream of any acoustically sensitive space to ensure that the noise they generate is adequately attenuated before it reaches the occupied room.

- **Duct-Borne Noise:** The duct work that connects a fan or air handler to a room is a contained system that will also connect the equipment noise and vibration to the room unless adequate precautions are taken to attenuate the noise before it gets there. Add internally lined supply and return path ductwork to unit or at least lining the last 7 feet of the duct work to the room.

- **Return Air Path:** Don't forget that the return-air path is an equally important contributor to noise problems. In fact, because return-air systems sometimes employ common plenums above corridor ceilings, there may be less duct work in the return-air path to attenuate the noise, and the transfer of return air from one space to another may be a significant breach of the sound isolation between them.

- **Velocity Noise:** As conditioned air travels from a fan to an occupied room, it is subjected to acceleration, deceleration, changes in direction, division and a variety of surfaces and obstacles. Each of these effects disturbs the uniformity of the airflow and causes turbulence, which in turn creates

noise. Keep low airflow velocities throughout the duct systems serving acoustically sensitive spaces.

- **Ducts Shape:** Rectangular ducts, particularly those with high aspect ratio can transmit excessive noise if not properly supported. Round ducts are stronger and have better aerodynamic characteristics, and therefore experience fewer noise problems.

- **Terminal Units:** Registers and diffusers should be selected to minimize noise output. Equipment spaces should be separated as far as possible from spaces with demands for low background sound levels. Avoid locating air terminals in close proximity to air-handling units. Provide at least one change in direction between air terminals and air-handling units.

- **Vibration Isolation:** Rotating or motor-driven machinery generates vibration energy that can travel through a building's structure and radiate from the walls, floors and ceilings in the form of airborne noise. Vibration is best controlled by decoupling the vibrating equipment from the surrounding structure. This can involve spring mounts, elastomeric mounts, inertial bases, floating floors and/or structural isolation joints. Any ducting, piping and conduit that are connected to vibrating equipment must be isolated with flexible connections.

- **Volume Dampers:** Volume dampers are needed to limit the amount of air that is allowed down the duct path. Unfortunately, dampers accomplish their volume control by pinching down the air stream, increasing the pressure and consequently the noise wherever they occur. Never use face dampers on the diffusers to adjust the air volume. The face dampers are essentially exposed directly into the room and in acoustically sensitive

spaces; they can generate audible noise even if face dampers are left wide open. Adjust the volume upstream using opposed-blade-type dampers.

HVAC noise and vibration control is a specialized field. Even with the best of intentions, there are hundreds of ways to make acoustical blunders that can render a technical space virtually unusable. There is no substitute for getting qualified help for the mechanical and acoustical design of HVAC systems in a technical facility.