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Dehumidification in Industrial and Building Applications

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Course Content

Introduction

Human comfort in conditioned spaces requires relative humidity in middle percentages, on the order of 30 to 60 percent. High relative humidity causes human discomfort due to perspiration and inability of the body to cool by evaporation of sweat while low relative humidity causes human discomfort due to nasal dryness and problems with static electricity generation. Dehumidification is required to control indoor relative humidity during warm and moist weather and humidification is required during cold and dry weather.

WHY DEHUMIDIFICATION?

There are many reasons to control humidity in indoor spaces:

1) Meeting the requirement of space usage
2) Providing the healthy and comfortable environment for occupants
3) Preserving the structural integrity of the building
4) Preserving furnishings, wall paper, flooring, paneling and valuable artifacts
5) Preventing rusting of pipes, metal items such as tools, window frames and other ferrous surfaces
6) Providing the specific environment for specific processes
7) Eliminating odors and foul smells

Dehumidification solves four common problems typically encountered in industrial applications:

Moisture Regains (clogging and sticking)

Virtually every substance has some affinity for moisture. Even plastic resins like nylon can collect six to ten percent of their dry weight in water vapor. In many cases, this presents no problem. In others, moisture regain can affect critical dimensions or make products stick together. The typical home salt shaker illustrates this point when moisture regain on humid days clogs the shaker holes.

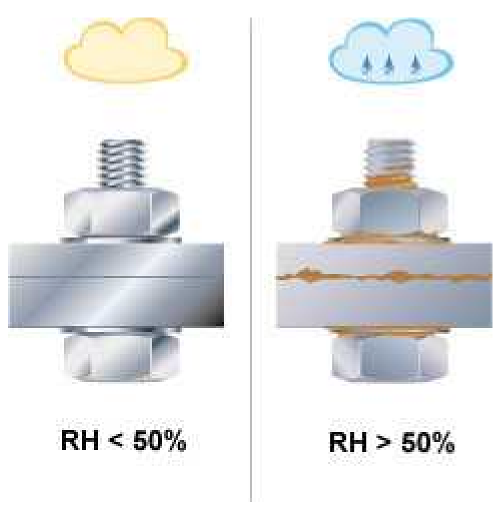
Protection against Condense

When cold surfaces are surrounded by moist air, water vapor will condense on the surface like “sweat”. This can lead to a surprising number of problems. For instance, consumers in a supermarket may not be able to see frozen foods in a refrigerated display case with a glass door. That may mean thousands of dollars in lost revenue. In other cases, the condensate dripping from the pipe or duct surfaces can spoil the ceiling, flooring, wall paper or discolor the furnishings. In all these cases, the solution is to lower the dew point of the air to at value inferior to the temperature of the coldest surface and the only possibility of lowering the dewpoint is by means of dehumidification, there are no other alternatives.

A surface will not have condensation or ice formation on it, if the air in contact with it has a dewpoint lower than the surface temperature.

Protection against Corrosion

The water in the air at a high relative humidity is sufficient for a corrosion process to take place. The limit lies at 45-50 %RH: if the humidity lies under that value, there is little risk of corrosion; if the atmospheric humidity lies above 60% the corrosion process will set in. From 60 to 100 %RH the corrosion process will be importantly accelerated.



Process/Product Drying- Drying heat-sensitive products

The quality of dry drugs, dry food, hard candy, sugar, chewing gums, capsules, pills and other hygroscopic materials can only be maintained, if it is kept in contact with air at a low relative humidity. Sensitive products can begin to deteriorate at just 30% relative humidity (RH). Many products require a constant RH of 10-15% or less for stability and many types of products must be dried to low moisture levels but cannot stand excessive heat. Enzymes, for instance, are destroyed by heat, and if yeast is dried with very hot air, it cannot work properly.

How to Remove Moisture

There are several methods of drying air:

1) Make-up air method - This method uses the principle of dilution, removing a portion of the moisture laden air from a space and replacing it with drier air. This method works well in parts of the country where outside air is drier. This method is difficult to apply in summer months and expensive to operate in winter due to heating costs.

2) Compression of air – As air is compressed beyond saturation, water vapor in the air begins to condense, and leaving air is drier. When air is compressed, the dew point is raised, that is, the temperature at which water vapor will condense is raised. Using compression to dry air is effective when small quantities are needed. This method has high installation and operational costs and is most common when less than 100cfm of dry air is required.

3) Heating or Evaporation - If the room (and thereby the air) is heated, the air will be able to hold more water, and the relative humidity will decrease. However, heating asks for a well-insulated building in order to meet the demands of building rules and regulations and in order to keep down the cost of energy. *Note that approximately 1052 Btu are needed to evaporate one pound of water.*

4) Dehumidification

   - Refrigeration dehumidification – As air is cooled, relative humidity increases; water begins condensing when the air is cooled below its dew point or saturation temperature. Air conditioners and mechanical dehumidifiers are both refrigeration dehumidifiers. This method is effective for desired conditions down to 45 percent RH for standard applications and has moderate capital and operating costs.

   - Desiccant dehumidification - Desiccant dehumidifiers use special materials that absorb or hold moisture. The material is unique in that it does not change its size or shape when acquiring the moisture and can be regenerated by applying heat. This technique is used effectively to dry air in the range of 0 to 50%RH. It has a relatively expensive capital expense as well as a high operational cost. Desiccant dehumidifiers are effective in controlling humidity at low temperatures, and have been widely applied in powder processing and product handling activities such as granulated sugar storage silos, packaging rooms, ammonium nitrate fertilizer storage buildings, and candy wrapping.

If you use too little dehumidification, materials dry too slowly, increasing the chances of secondary damage; if you use too much, you may waste your client's money. If the application is drying of products, it is usually best to first get rid of excess water followed by evaporation and dehumidification. Specifically for comfort and critical low temperature manufacturing applications, you have two possibilities: Refrigeration and Desiccant Dehumidification. A further discussion on “how to produce dry air” will follow later in the course.

SECTION – 2 PSYCHROMETRICS

The rudimentary basis for understanding any air conditioning, moisture control and dehumidification is psychrometrics, the study of the properties of air-water mixtures. Psychrometrics is an essential foundation for understanding how to change air from one condition to another. But before we discuss this further, let us familiarize ourselves with some of the often used terminologies and their definitions.

1) **Dry Bulb Temperature (DBT) -** The Dry Bulb Temperature refers to the ambient air temperature measured using a normal thermometer freely exposed to the air but shielded from radiation and moisture. It is called "Dry Bulb" because the air temperature is indicated by a thermometer not affected by the moisture of the air.

   The dry bulb temperature is an indicator of heat content of the air. As the DB temperature increases, the capacity of moisture the airspace will hold also increases. This reduces RH and allows for drying. The dry bulb temperature is usually given in degrees Celsius (ºC) or degrees Fahrenheit (°F). The SI unit is Kelvin (K). Zero Kelvin equals to -273°C.

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2) **Wet Bulb Temperature (WBT) -** The Wet Bulb temperature is the temperature measured by using a thermometer whose glass bulb is covered by a wet wick/cloth.

   The wet bulb temperature is indicator of moisture content of air. Wet bulb temperature is very useful in industrial drying processes as the difference between the dry bulb and wet bulb temperature is a measure of the drying capacity of the air. At 100% relative humidity, the wet bulb temperature equals dry bulb temperature.

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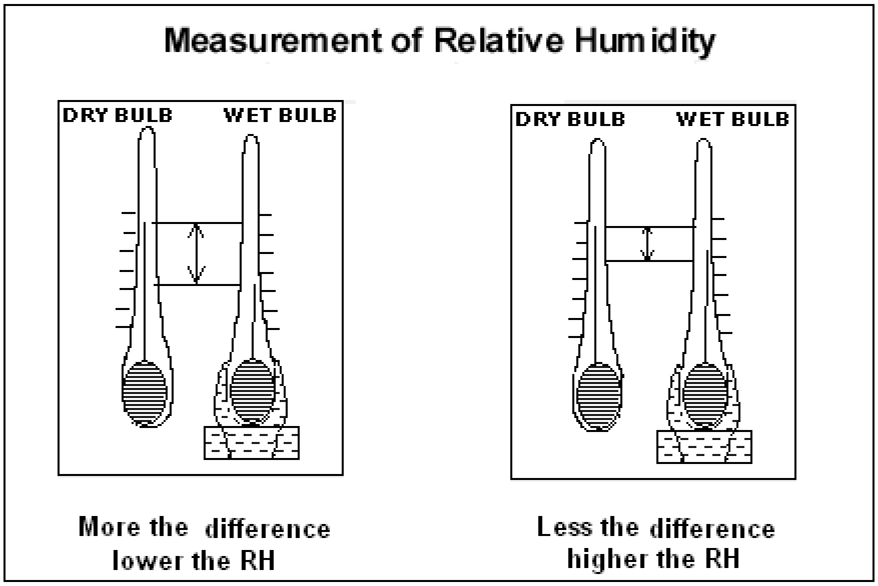
3) **Humidity-** Humidity is the quantity of water vapor present in air. It can be expressed as an absolute, specific or a relative value.

   - <u>Absolute humidity</u> is the actual mass of water vapor present in the air water vapor mixture. Absolute humidity may be expressed in pounds of water vapor (lbs).
   - <u>Specific humidity or (humidity ratio)</u> is the ratio between the actual mass of water vapor present in moist air - to the mass of the dry air. The humidity ratio is very useful in dehumidification because at normal pressures, this ration represents an actual measure of water mass relative to air mass. Therefore, the humidity ratio does not change with changes in air temperature. Specific Humidity is normally expressed in grains of water vapor /lb of dry air and may also be expressed in the units of pounds of water vapor/lb of dry air or grams of water vapor /kg of dry air.
   - <u>Relative Humidity or RH</u> is the actual amount of moisture in the air compared to the total or maximum moisture the air can hold at a given temperature. When air has 50 percent relative humidity (RH), we say it is 50 percent saturated (the terms are numerically so close that we use them interchangeably). Obviously, as air approaches 100 percent saturation, it can take on less and less water until at 100 percent RH, the air cannot hold more water.
     Relative humidity is determined by comparing the "wet-bulb" and "dry-bulb" temperature readings. Dry bulb and wet bulb temperatures are taken simultaneously and then plotted on a psychrometric chart. Relative humidity is determined by the value at the intersection of two temperature lines.



4) **Dew Point Temperature –** The Dew Point is the temperature at which water vapor starts to condense out of the air and becomes completely saturated. Above this temperature the moisture will stay in the air. When we find dew on grass in the morning, we know that the temperature of grass reached the dew point sometime during the night before. As air cools during the night, it can hold less and less moisture and the relative humidity will get very high. Finally, air becomes so cool and saturated; it releases some moisture as dew.

   The dew point temperature is an indicator of the actual amount of moisture in air. The local dew point temperature is sometimes given on weather reports and when the dew point temperature is not provided, the nighttime low temperature is a good estimate of the dew point temperature and the amount of moisture held in the air.

   The dew-point temperature is expressed in degrees and like humidity ratio; it represents an absolute measure of the moisture in the air at a constant pressure. *If the dew-point temperature is close to the air temperature, the relative humidity is high, and if the dew point is well below the air temperature, the relative humidity is low.*

   The Dew Point temperature can be measured by filling a metal can with water and ice cubes. Stir by a thermometer and watch the outside of the can. When the vapor in the air starts to condensate on the outside of the can, the temperature on the thermometer is pretty close to the dew point of the actual air.

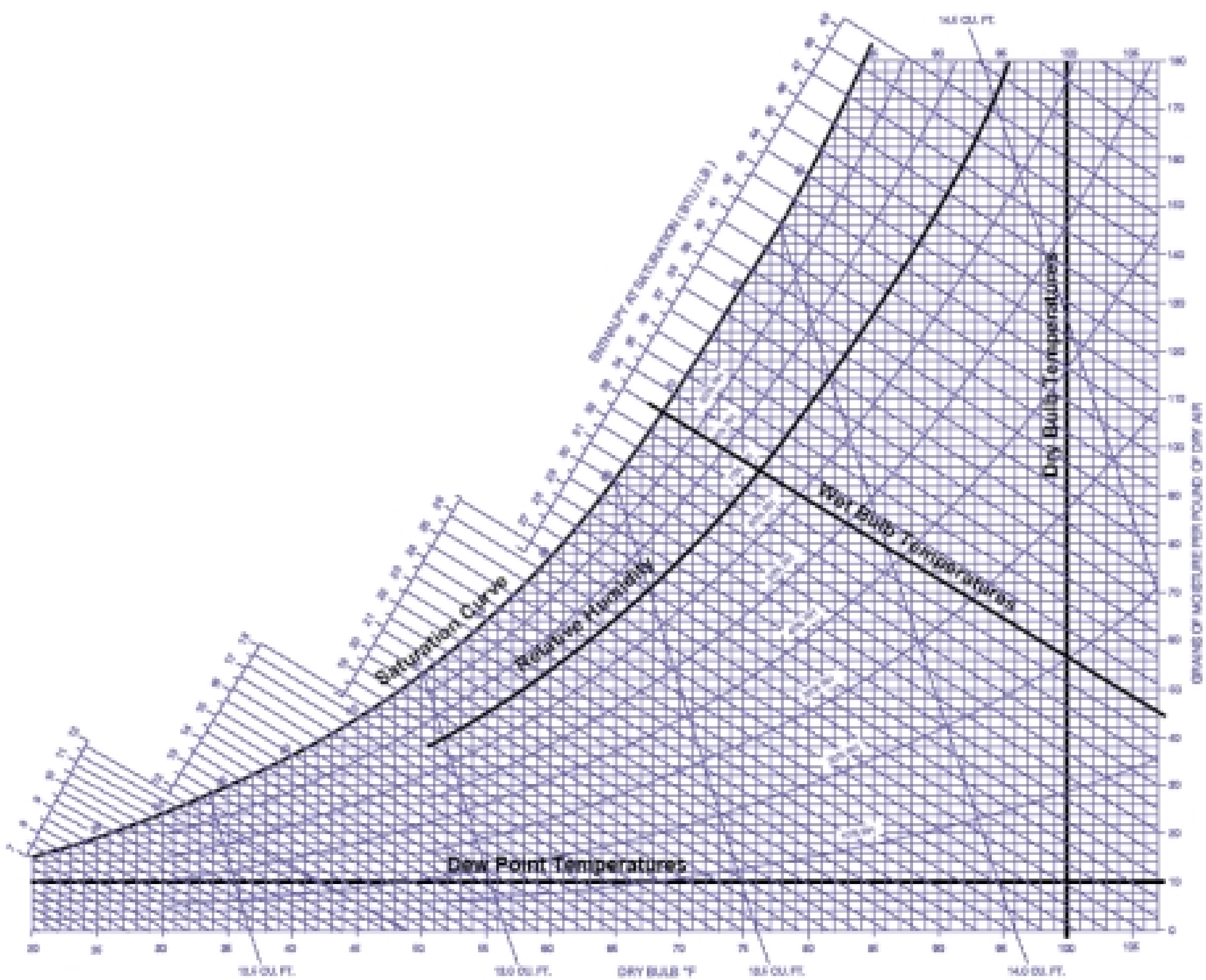
5) **Grains of Moisture:** Term used to express the weight of moisture per pound of air (14 cubic feet). 7000 grains is the most that can be held in one pound of air. Since water weighs 8.34 pounds per US gallon and since there are 8 pints in one gallon, 7000 grains is equal to about 1 pint of water. Air cannot hold much moisture as far as poundage of water. In indoor comfort applications, the "space-neutral" conditions are defined as 75ºF, 50%RH, which corresponds to 65 grains per lb of dry air.

6) Vapor Pressure – This is the pressure exerted by the water molecules in air. Sometimes expressed in psia or in-Hg, it is very useful in sorption dehumidification because water vapor moves through air to equalize differences in vapor pressure. The operating principle of desiccant dehumidifiers is to move water vapor through difference in vapor pressure.

7) **Psychrometric Chart –** The psychrometric chart shows the relationship between the air volume, temperature and relative humidity and is used to calculate specific humidity, dew point (wet bulb temperature) and vapor pressure.

- On the psychrometric chart, the dry-bulb temperature is shown along the bottom axis and the constant dry bulb temperatures appear as vertical lines in the chart.
- Lines of constant wet bulb temperatures run diagonally from the upper left to the lower right in the psychrometric chart.
- Combining the dry bulb and wet bulb temperature in a psychrometric chart, gives the state of the humid air.
- The vertical axis represents humidity ratio, the moisture content of the air mixture, expressed in this chart as grains of moisture per pound of dry air. Some charts show humidity ratio as pounds of moisture per pound of dry air, where 7000 grains = 1 pound.

The laws of physics dictate that the air will hold a specific amount of water (measured in lbs or grains) based on the dry bulb temperature of the air. As air temperature decreases, the amount of water that can remain in the air as vapor also decreases. This means that if the temperature falls and the amount of water vapor in the air remains the same, the vapor must become liquid (visible droplets). The air at this point is “saturated” with water vapor. If the maximum humidity ratio for each dry bulb temperature is plotted, the curve connecting these points is called the Saturation Line or the 100% Relative Humidity Line. Any additional moisture in the air mixture would be visible droplets or fog.

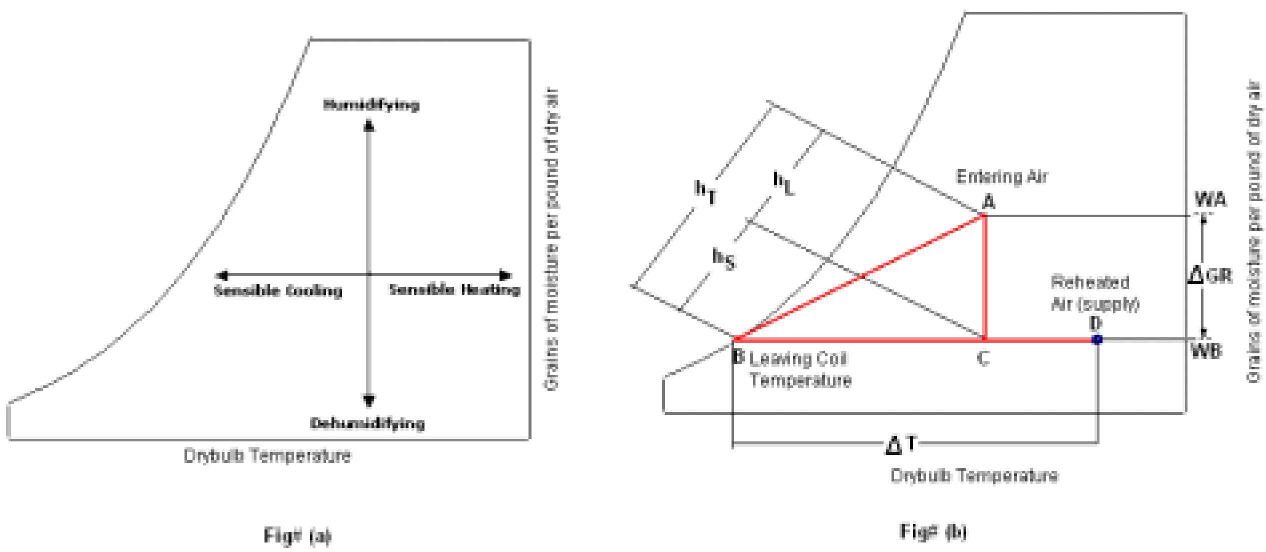


Psychrometric Processes

There are broadly four types of energy changes:

1) Sensible heat occurs when heat is added without the addition or reduction of moisture.
2) Sensible cooling is the reverse of above
3) Latent heat also known as humidifying, is the addition of moisture without changing the dry bulb temperature.
4) Latent cooling or dehumidification is the removal of moisture.

Fig# (a) below shows how these processes are displayed on the psychrometric chart. Rarely will these occur as shown but will be rather be a mixture of them. Fig# (b) shows dehumidification by cooling is a combination of sensible and latent cooling and sensible heating. First the system cools the air to reduce the dry bulb temperature to the dew point, then the latent cooling reduces the absolute humidity and finally the air is reheated increasing its dry bulb temperature before distributed to the space.



PSYCHROMETRIC PROCESSES

Fig# (b), Psychrometric Process

hT = Total Cooling, A-B

hS = Sensible Cooling, C-B

hL = Latent Cooling, A-C

WA = Specific humidity A (room air)

WB = Specific humidity C (supply air)

ΔT = Total temperature rise, B-D

ΔGR = WA - WB

Note an important aspect of dehumidification application:

Air is compressible fluid and its volume is represented by the following equation:

V = K (T/P)

Where

V = Volume

T = Temperature

P = Pressure

K = Constant

As the air temperature increases, its total volume increases and decreases on reduction of temperature. Pressure has the opposite effect; as pressure increases volume decreases. Water (moisture), however is not compressible. Therefore given a specific amount, it will always occupy the same amount of volume. The psychrometric chart illustrates this concept. *As moisture laden air is heated or cooled, the air volume changes but the moisture does not. Thus there is a change in relative humidity, without a change in actual water content.* This is important to understand because water damage occurs at an absolute humidity concentration regardless of its relative humidity. A further description is noted below:

What is a good measure for moisture control - Relative Humidity, Dew Point or Absolute Humidity

Relative humidity has been one of the most inexpensive means of humidity measurement and control and is widely used and understood, but there is fair amount of confusion on this point as well.

Relative Humidity - Relative Humidity measurements take into consideration both the sensible and latent component of air. For example relative humidity reflects the relative amount of water that air can hold at a given temperature. Change the temperature and the humidity will change even if net amount of moisture in the air remains the same. In fact a 1ºF change in room temperature can change the relative humidity by 2%. If outside air at 85ºF and 60% RH is cooled to 72 degrees without any moisture removal, the RH will increase to almost 90%. This temperature effect on relative humidity makes it useless as a moisture control parameter.

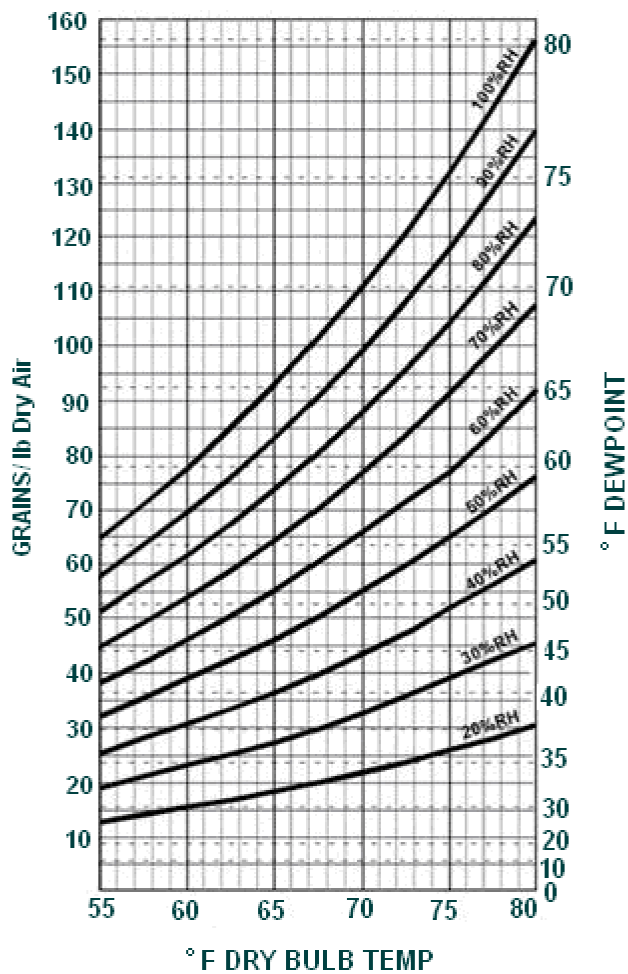
The mold growth is closely linked to relative humidity on the surface of the material. This material may be in the ductwork, in carpet, or on the inside of walls and at a very different temperature than what we would typically measure in the space. The space temperature and associated RH measured will likely be very different than on the surfaces of concern. So while RH on the surface of a material is a good indicator of mold growth, RH in the space cannot be used to predict or control the moisture content on mold vulnerable surfaces.

The key to controlling moisture in buildings is to directly measure the latent content (moisture) of air separate from its sensible (dry bulb) component. This is where measurements such as dew point and grains/lb of dry air come into play. Temperature will not affect either of these measurements.

Dew Point -Dew point is really a predictive measure that indicates the temperature at which moisture in the air will reach 100% and condense onto a surface. To ensure high moisture levels or condensation does not occur on those surfaces, dew point levels in the air should be controlled in the building to below temperature of the coldest surfaces in a space (e.g. slab on grade floor or un-insulated duct sections). This will ensure that moisture conditions never approach the point that would sustain mold growth on cooler surfaces. . Humidity Ratio (Grains)

Absolute Humidity -The measure of Grains/lb of dry air called the humidity ratio and is an indication of the mass of water vapor in air. This measure is most useful to designers and operators of dehumidification equipment because it can be used to predict the specific amount of energy necessary to remove a given amount of moisture from air. This measure is ideal for building control, where your control strategies are based on maintaining a given target level of moisture control.

Figure below shows the relationship between dew point, grains and relative humidity. This chart shows how RH can change with temperature but the latent content of air is unaffected by temperature. For a designer a chart like this could be used to select the ideal conditions desired in the space for comfort in terms of temperature and relative humidity. The control strategy for the space would then be to maintain the latent conditions in the space that are equivalent to that design target.



SECTION -3 REFRIGERATION & DESICCANT DEHUMIDIFICATION

REFRIGERATION DEHUMIDIFICATION

Principle: Lowering air temperature decreases the air's ability to hold moisture.

As air is cooled, relative humidity increases; water begins condensing when the air is cooled below its dew point or saturation temperature. The air becomes drier in absolute terms, but is now saturated i.e. its relative humidity [RH] is close to 100 %. If a low RH is needed in addition to a low absolute moisture level, the air can be heated after it leaves the cooling coil. This method is effective for desired conditions down to 45 percent RH for standard applications.

However cooling air just for drying is usually not practical as this method has significant capital and operating costs. Normally, this method is reserved for applications where cool air is needed anyhow especially in drier and humid locations requiring air-conditioning and secondly where high outdoor air needs drying only slightly lower than the incoming ambient.

Evaporator Optimization

Refrigeration dehumidifiers operate in many modes of operation in order to insure continuous moisture removal. To achieve this, the system's design incorporates many different critically sized components that must function seamlessly. A cooling coil (evaporator) must be designed to transfer the required energy and shall be selected to provide a minimum 52ºF supply air temperature, while still operating at standard chilled water temperatures. It is often necessary to utilize cooling coils very deep in the direction of air flow with wide fin spacing when circulating anti-freeze fluids at temperatures as low as 32ºF without icing the coil surface. An important distinction of conventional air conditioner cooling coil v/s refrigeration dehumidifier cooling coil is

1) A conventional air conditioner will generally use three (3) or four (4) rows with 400CFM per ton (12,000 BTU's) to be removed. This combination provides a high sensible-to-latent ratio, which is key to the system achieving a high EER (energy efficiency ratio). A dehumidifier will use six (6) to eight (8) rows with a reduced air volume of 200CFM per ton. This provides a high MRE (moisture removal efficiency) value. This becomes important in high humidity applications such as pool facilities, industrial plants and treatment of 100% outside air.
2) The second major impact is that the air conditioner type coil creates an approach temperature *(the difference between leaving air temperature across the coil and the actual refrigerant temperature in the coil)* of 12°F to 15°F (6°C to 7°C), while the dehumidifier design creates an 8°F to 10°F (4°C to 5°C) approach. This is important in part-load situations where the leaving air temperature limits are 52°F or lower for the air conditioning design.

Refrigeration Dehumidification Limitations

In drier parts of the country moisture removal by air conditioning is usually sufficient. However, air conditioners are designed to cool more than they dehumidify air. This can cause a problem in the warm, humid climates of coastal areas such as Florida, where air-conditioning alone can't keep up with the dehumidification and the relative humidity inside a space can get quit high. Summer dehumidification needs are **much** greater than the need for cooling air only and therefore many a time air conditioning and desiccant dehumidification may be both needed almost anytime of the year. As a passive measure, in warm, humid climates, it is very important to stop unwanted infiltration of air and limit air change by ventilation. By repeatedly returning air to the air conditioner, enough moisture can be "wrung out" of air to help keep indoor humidity levels below the mildew alert level. Drier air takes less energy to cool so energy is saved, too.

DESICCANTS

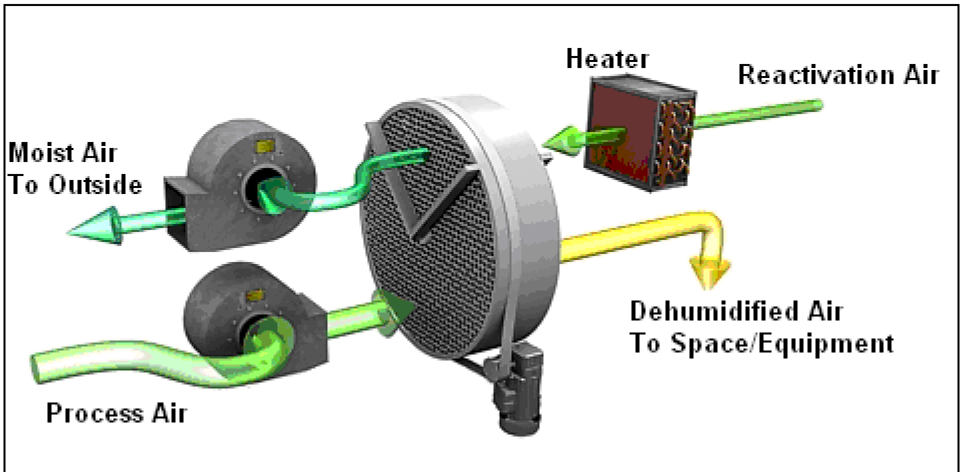
Desiccants remove water vapor by chemical attraction caused by *"differences in vapor pressure".* When air is humid, it has a high water vapor pressure. In contrast, the surface of dry desiccant has a very low vapor pressure so the water vapor moves out of the humid air onto the desiccant surface to eliminate the vapor pressure difference. Eventually, the desiccant surface collects enough water vapor to equal the vapor pressure of the humid air. Then, it must be dried (reactivated) by applying heat before it can be used to remove more moisture from the air stream. In general

- Temperature for reactivation: 300 - 350^{o}F (150 -175ºC)
- Heat required for reactivation: 2050 Btu/lb of water removed (4800kJ/kg)

Desiccant Wheel Construction & Operation

The desiccant structure is formed into the shape of a wheel. The wheel constantly rotates through two separate airstreams. The first air stream, called the process air, is dried by the desiccant. The second air stream, called reactivation or regeneration air dries the desiccant. The desiccant material is contained in the walls of thin air channels that extend through the depth of the wheel. Channel diameters vary, but typically are about 2 mm. This looks like a honeycomb which is open on both ends. Air passes through the honeycomb passages, giving up moisture to the desiccant contained in the walls of the honeycomb cells. Wheel diameter depends on how much air must pass through it. Large airflow volumes require a larger diameter wheel.

The figure below illustrates the simple principle of desiccant wheel operation.



DESICCANT WHEEL OPERATION

The desiccant cycle follows four steps:

Step#1 is sorption, where the process air passes through a portion of the rotating desiccant wheel, giving off its water vapor to the desiccant. As the moisture is removed from the air, the desiccant surface vapor pressure and temperature rises as latent heat in the air is converted to sensible heat.

Step#2, the moisture content in the desiccant reaches equilibrium. The surface vapor pressure is equal to the vapor pressure of the surrounding air. Without a vapor pressure differential, there is no further moisture movement to the desiccant.

Step#3 is the reactivation of the desiccant, when the surface vapor pressure rises as the desiccant is heated. Because the vapor pressure of moisture in the desiccant is higher than that in the surrounding air, moisture leaves the desiccant. The moisture content is reduced, but vapor pressure remains high because of the higher temperature.

Step#4 is the cooling of the desiccant. The vapor pressure at the desiccant's surface reduces rapidly with cooler temperatures. When the desiccant is cool, the vapor pressure once again is lower than the surrounding air, so it can collect moisture again.

Because a desiccant dehumidifier removes water vapor rather than condensed liquid from the air, there is no risk of freezing. This type of equipment is most often used for applications requiring dewpoints below 50ºF.

Desiccant Key Facts

1) Desiccant wheels are designed to operate with either "a 25% area for reactivation and 75% area for process (25/75 split)" or with "50% area for reactivation and 50% for process (50/50 split)". Generally, the 25/75 split is used for industrial dehumidification, low dewpoint and compact desiccant cooling applications. The 50/50 split is more often used for commercial cooling applications or application where low temperature waste heat is available for reactivation.

2) Heat of reactivation energy can be supplied either with electric resistance, (indirect) steam, (indirect) hot water, or direct (or indirect) fired natural (or propane) gas. Additionally, solar or waste heat sources may be utilized. Desiccant wheels are designed for regeneration temperatures up to a maximum temperature of 350°F. In addition to the regeneration temperature, several other factors influence the performance of the desiccant wheel. Process and regeneration inlet humidity and temperature, regeneration to process flow ratio, face velocity, and wheel diameter & rotational speed all has an impact on performance.

3) Energy to drive the desiccant cycle is proportional to the mass that is heated and cooled. The more desiccant and water that is heated, the more energy is used. The cost of operation depends primarily on the cost of the reactivation energy plus the cooling energy.

4) Desiccant process is the reverse of evaporative cooling. When water is evaporated into air, the heat needed for evaporation comes from that same air and the sensible temperature falls. Conversely, when air is dehumidified, the heat needed to evaporate the water originally is liberated and the temperature of the air stream rises.

5) With desiccants, moisture removal occurs in the vapor phase. There is no liquid condensate. Consequently, desiccant dehumidification can continue even when the dew point of the air is below freezing. This is different from cooling- based dehumidification, in which the moisture freezes and halts the process if part of the coil surface is below 32°F.

6) *The ideal desiccant would have an infinitely high surface area for collecting moisture, and an infinitely low mass,* since the required heating and cooling energy is directly proportional to the mass of the desiccant and the mass of the machinery which presents the desiccant to the air-stream. The heavier the desiccant assembly compared to its capacity, the more energy it will take to change its temperature-which accomplishes dehumidification.

7) One aspect of desiccant wheel behavior can be confusing to the first-time user of the technology; air leaves a desiccant wheel dry, but leaving air is 30 to 40ºF warmer than inlet air. *For example, if air enters a desiccant wheel at 70°F and 50% RH, it will leave the wheel at about 100°F and 4% RH.* This non-intuitive behavior becomes easier to understand as the reverse of evaporative cooling. When water is sprayed into air, it evaporates by using part of the sensible heat in the air—so the dry bulb temperature falls as water vapor is added to the air. Desiccants produce the opposite phenomenon. As water vapor is removed from air, the dry bulb temperature of the air rises. The amount of temperature rise depends on the amount of water removed. More water removal produces a greater temperature rise.

   The initial user naturally asks: how can desiccant systems save cooling energy, if dehumidification adds sensible heat to the air? Part of the answer is that some heat is moved to reactivation by a heat exchanger. The rest of the answer depends on the application. For example, if air is dry, it may not be necessary to cool it if the space is already overcooled—as in a supermarket, where display cases cool the aisles as well as the product. Alternatively, dry air can be cooled using low-cost indirect evaporative cooling such as cooling towers, or with highly efficient vapor compression systems operating at high evaporator temperatures. In such cases, desiccants can save energy and energy cost.

Desiccants – Adsorption or Absorption

Desiccants do not absorb moisture, they *"adsorb"*. Absorption occurs when one substance is chemically integrated into another. Adsorption means moisture is held inside a desiccant by physical bonds; no chemical change actually occurs.

1) Adsorption is a physical process in where *moisture is held on the surface of the material* without any change of in the physical or chemical structure of the material. Water is adsorbed or held on the surface of the material and in the pores, Adsorbents are mostly solids. Typical examples of Adsorbents used for dehumidification are silica gel, molecular sieve and activation alumina.

2) Absorption involves a *change in the physical or chemical structure of the material* and it is in general not easy to reactivate the material. Absorbents are generally liquids or solids; solid sorption materials include sodium chloride salt (NaCl) and liquid sorption systems use calcium chloride solution, glycol or kathene fluids to absorb moisture from an air stream. Water is then boiled out before the sorption liquid is ready to absorb moisture again. Breweries, food processing and natural gas gathering facilities use liquid sorption equipment. Neither of the above systems is generally used for abrasive blasting operations.

Desiccant Classification

Desiccants can be either liquids or solids, and there are many different materials of both types. However, the great majority of systems built for commercial and industrial applications use solid adsorbents such as silica gel or activated alumina.

- **Silica gel - SiO_2** - is a hard, adsorbent, crystalline substance and very porous. Voids are about 50 - 70% by volume and adsorb water up to 40% of its own mass. The bulk density of silica gel is 480 - 720 kg/m^3. The specific heat capacity is 1.13kJ/kg-K.
- **Activated alumina** is about 90% aluminum oxide Al_2O_3 and very porous. Voids are about 50 - 70% by volume and adsorb water up to 60% of its own mass. The bulk density is 800 - 870 kg/m^3. The specific heat capacity is 1.0 kJ/kgK.

In weighing the various choices of desiccants, it is important to consider the ambient and the space environment in relation to the desiccant's individual capabilities. Key parameters include: initial humidity; amount of moisture acceptable in space; volume of product; Moisture Vapor Transmission Rate (MVTR); and the maximum level of RH permitted within the space.

COMPARING DESICCANTS TO REFRIGERATION DEHUMIDIFICATION

Both desiccant dehumidifiers and mechanical refrigeration systems can remove moisture from the air, so the question is - which type is best suited for a given application? There really are no simple answers to this question but in general cooling-based dehumidification handles the moisture load occurring at high dewpoints; desiccant-based dehumidification removes the moisture load at lower dewpoints. The mix of the two technologies depends on the characteristics of the specific application. The generally accepted guidelines which most designers follow are listed below:

1) **Evaluating the dewpoint control level:** When the required moisture control level is comparatively high (above 50ºF dewpoint), cooling-based dehumidification is economical, in terms of both operating costs and initial equipment cost. Below this level, costs increase because precautions need to be taken to prevent condensed water on the cooling coil from freezing.

   Although water does not freeze until temperatures fall below 32ºF, a dehumidification system may have to deliver air below that level to maintain a room below 50ºF dewpoint. Without protective measures, a cooling-based dehumidifier providing air at low dewpoints can freeze. Equipment fitted with freeze-protection devices costs more and has higher operating costs/kg of water removed. Under these circumstances, desiccants become more economical than cooling-based systems at low dewpoints.

   In general

   - If the dew point is above 50 degrees, consultants recommend refrigeration dehumidification.
   - If your dew point falls below 40 degrees, desiccant dehumidification provides the best solution.

2) **Gauging relative humidity sensitivity:** When a process needs low moisture level in absolute terms, but can tolerate a high RH, a cooling-based dehumidification system can be cost effective without the need for desiccants. For instance in food storage areas, an ideal temperature for fruit and vegetable storage might be 40ºF. Of course, the dewpoint must be lower than that. If the RH is below 90%, the fruit can dry out and lose value. Because the product needs both low temperature and high humidity, cooling-based systems are ideal. In contrast, other processes might demand a low RH and a low dewpoint.

3) **Determining temperature tolerance:** An application with a narrow temperature tolerance needs cooling and heating in addition to dehumidification. If the application can tolerate wide temperature variations (such as occur in unheated storage, for example), the dehumidification equipment alone may suffice.

4) **Gauging Seasonal Variations:** On a seasonal basis, desiccant dehumidifiers can generally be a good choice from January through May and September through December, when morning temperatures are below 65ºF. Air conditioners may be used from May through September when morning lows are above 70ºF. Mechanical refrigerant dehumidifiers offer value from March through November when morning lows are above 55ºF. Both desiccant-based and refrigeration-based dehumidification systems work

most efficiently when used together. The advantages of each compensate for the limitations of the other.

5) **Energy Performance:** The amount of energy consumed to remove moisture from inlet air varies by machine. *In general, energy requirements for desiccant dehumidifier are high because of reactivation heat whereas in comparison cooling based dehumidification uses 50% less power.* Based upon an inlet of 95ºF, 50% relative humidity; desiccant dehumidifiers remove about 1.5 pounds of water vapor per kilowatt of electricity used. Air conditioners remove 3 pounds and mechanical refrigerant dehumidifiers remove 5 pounds. *Desiccant performance is fairly independent of inlet conditions*; (1.5 pounds per KW) over a large range of climates whereas air conditioner and mechanical dehumidifier deliver varied performance based upon inlet conditions.

Summarizing…

Desiccant systems are applicable to existing or new HVAC systems for clean rooms, supermarkets, refrigerated warehouses, ice rinks, schools, restaurants, theaters, hotels, hospital/healthcare facilities, and situations where one or more of the following situations apply:

1) Low indoor humidity (dew point below 50°F)
2) High latent load fraction (greater than 25%)
3) High outside air fraction (greater than 20%)
4) High electrical cost and low gas costs
5) Available heat source from waste heat, steam, hot water or gas for regeneration of desiccant
6) Can remove up to 50 grains of moisture per lb of air processed

Refrigeration-based dehumidification systems are sometimes more economical than desiccant based dehumidifiers when higher temperatures and humidity in the conditioned space is acceptable. Mechanical refrigeration systems are seldom used for applications below 50% RH at about 22°C or for a dew point lower than 11°C. Desiccant-based systems are more economical than refrigeration systems at lower temperatures and lower moisture levels and these keep humidity lower than would be practical with conventional cooling-based systems. Typically, a desiccant dehumidification system is utilized for applications below 45% RH down to less than 1% RH. Thus, in many applications, a DX or chilled water pre-cooling coil is mounted directly at the dehumidifier inlet. This design allows for removal of much of the initial heat and moisture prior to entering the dehumidifier where the moisture is reduced even further.

SECTION -4 SYSTEM DESIGN

Industrial & commercial dehumidification systems are tailored for each project. Therefore, a near-infinite variety of possible components are available to serve a near-infinite variety of possible applications. This presents the need to make many decisions early in the design process, often before cost/benefit implications are clear. The engineer investigating the system should work closely with potential suppliers to compare costs and benefits of dehumidification with alternative solutions. Suppliers can be most helpful when key aspects of the project are well defined. The engineer should be able to provide or have on hand the following information.

1) Purpose of the project

The purpose of the project must be clearly understood and documented so that decisions can be properly ordered according to their importance. If the system must prevent mold growth on starch in a storage silo, a strict tolerance of ±1% RH is not needed. The only concerns are that humidity does not exceed 60% and that condensation does not occur. A simple, inexpensive configuration will do the job.

Conversely, if dehumidification is needed to prevent the corrosion of lithium, a control with a tolerance of ±5% RH is useless. Above 2% RH, lithium corrodes and gives off hydrogen which eventually explodes. A sensor with a tolerance greater than the critical control level could not start the system in time to prevent the explosion. Understanding the purpose of the project helps prevent both unnecessary expense and false economy.

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2) Establish control levels and tolerances

Next, the humidity and temperature control levels and tolerances needed to achieve the purposes of the project must be determined. Sometimes, these decisions require research. Other times the relationship between the process and moisture levels is well understood. For example, if a process bogs down in the summer, but not the rest of the year, humidity tolerance is likely quite wide. The dehumidification system need only remove summer humidity extremes. Or perhaps the supplier of the problem material can recommend optimal environmental conditions for processing the product. The control setpoint must be established to allow the peak heat and moisture loads to be calculated.

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3) Estimate Loads

Without loads, equipment sizes and costs cannot be estimated. For example, a system that holds humidity at 72ºF and 35% RH is much smaller than one that holds humidity at 72ºF and 25% RH, all other variables being equal. The lower the desired humidity level, the more costly is the system. Higher moisture loads also raise costs. For these reasons, calculating loads is a critical step in system design. Typical moisture loads come from ventilation air, air infiltration, miscellaneous openings, people, products and packaging, and vapor permeation. The following aspects must be carefully examined while calculating the moisture loads:

a) **Outdoor Design Conditions:** A peak design weather condition is an important element in the load calculations. The end user must decide how conservatively the system should be sized. If extreme weather data are used, the system will control humidity throughout the 8760 hr in a typical year. The system will also cost a lot. If some off-hours can be risked, costs may be reduced 20-30%. However, if all moisture loads peak at the same time in extreme weather, the humidity level may rise above the setpoint. These options are quantified in the Fundamentals Handbook published by the American Society of Heating, Refrigerating, and Air-Conditioning Engineers according to the percent of annual hours that weather conditions can be expected to be above certain values. For example, the 0.4% values are likely to be exceeded only 35 hr/yr. A less conservative design point would be 1% or 2.5% values, which may be exceeded for 70 hr and 219 hr respectively. The end user must decide which data to use because he is in the best position to assess the economic and safety consequences of being above specification for short periods.

b) **Ventilation air/air filtration:** Outdoor air contains large amounts of moisture, especially in summer. Ventilation air is typically introduced through air conditioning and heating units. Fresh air is required in most controlled spaces. Most building codes specify a specific amount of air per person (20CFM per person per ASHRAE 62) or per sq ft of occupied space.

   The make up ventilation requirements are also influenced by the exhaust requirements. If the exhaust air is more, the space will need more make up air to replace. The problem is especially acute in large areas where exhausts may not be obvious. Engineers need to be fully aware of the effect of insufficient makeup air on humidity-controlled spaces. Higher ventilation rates particularly in the coastal humid areas will require larger dehumidification equipment. The most cost-effective adjustment to building operation is to minimize exhaust air, reducing the cost of dehumidifying the makeup air, however, note that the state or local codes compliance is especially important for hospitals and health care facilities or for any application that exposes humans to toxic fumes.

c) **Ventilation Rates for Drying Projects:** For moisture control, ventilation rates are dependent upon moisture loads and the moisture level of supply air. Using ventilation rates to size drying projects can result in inaccurate amounts of equipment for the job. When exhaust fans are used during a project, always match dehumidifier air flow with exhaust flow.

d) **Infiltration through Miscellaneous Openings:** The moisture-laden air “infiltrates” the building through opening and closing of doors and through cracks and crevices in a building envelope. The amount of infiltration varies with the age and integrity of construction and tightness of windows. Opening a door pulls in moist air. Observe and note the number of times a door is opened at busiest times.

Air locks greatly reduce moist air infiltration. As humidity control levels go lower, air lock doors become more economically advantageous. With an air lock, it is assumed that equilibrium is reached half way between the inside and outside conditions and that all the air enters the room each time the lock opens.

When product must enter or leave a humidity-controlled room on a conveyor, the opening must be considered as a possible infiltration source. The infiltration of moist air through large openings such as ducts can be reduced by supplying a slight overpressure of makeup air to force dry air out of cracks rather than letting moist air leak in. Sealing building cracks also reduces dehumidification costs for a modest investment in caulk.

e) **People:** People produce moisture and this varies with the level of activity. When workers exhale or perspire, moisture is given off, creating another load source. The rate depends on the exertion level. When room loads are calculated, allowances must be made for "visitors" entering and leaving the room. Doubling "people" estimates to allow for changes in room use is often recommended.

f) **Moisture Emission from Products/packaging/appliances:** The load from products and packaging varies greatly by application. In large storage applications, moisture released from product may be the single largest load component. The load equals the difference between a product's initial wet weight and the weight when at equilibrium with the lower humidity. In addition showers, pools, fountains and some commercial processes may also contribute to moisture generated from within a building.

g) **Vapor permeation:** Vapor permeation through building components is typically the smallest load component. It accounts for less than 2% of the total. This load should receive more attention if the building is very large and moisture permeates across a large surface area or if the control condition is very low. Below 5% RH, every leak is critical.

SECTION-5 CALCULATING MOISTURE LOADS

To properly apply a dehumidification system, the amount of moisture to be removed must be calculated.

This section provides an approximate yet very practical method for calculating the moisture load in various applications. There are many sources of moisture in a facility. A list of the common ones follows:

- Infiltration
- Permeation
- Ventilation and make-up air
- Door and window openings
- People
- Processes
- Product

INFILTRATION & PERMEATION

Space moisture load is a combination of permeation and infiltration; both of which are often considered the same thing.

1) Permeation is the migration of water vapor through materials such as brick and wood. Permeation is a straight line function of the difference in interior and exterior vapor pressure (determined by grains/lb).
2) Infiltration is the movement of water vapor through cracks, joints and seals. Infiltration is generally represented in air changes per hour and is not a straight line function because of the two factors involved:
   - Each pound of air entering the space will impose a moisture load determined by the difference in interior and exterior moisture content.
   - Since the vapor pressure differs as the moisture content, the vapor will move at a higher velocity than the air.

The combination of the two factors will be encountered in determining the moisture load. *As the difference between interior and exterior moisture content increases, there will be a corresponding rate of increase in the permeation and infiltration loads.* Moisture load in a space due to infiltration and permeation is not easily measured. Factors such as the actual moisture deviation, materials of construction, vapor barrier and room size all have an effect on the vapor migration. Suppliers have been using some basic equations derived from field studies to estimate moisture infiltration and permeation.

Calculating Infiltration and Permeation Load

The combined infiltration and permeation load can be approximated from the following equation:

Lb/HR Moisture = V x GR x F1 x F2 x F3 x F4 / (7000 x C)

Where

V = volume of controlled space in question - ft^3

ΔG = difference between the grains per lb of outside air and the grains per lb desired in the controlled space (Table – I)

F1 = Migration factor due to moisture difference (Table – II)

F2 = Air change factor (Multiplier from Table III)

F3 = Construction factor – Table IV

F4 = Barrier factor (Table V)

C = 13.5 (conversion factor for cu-ft / lb)

7000 = Conversion factor for grains/ lb

Grain Difference (ΔG)

The ΔGR (grain/lb) difference between the outside and inside conditions is obtained from the psychrometric chart. For most applications the only information available is the dry bulb and relative humidity or dry bulb and wet bulb temperatures. The psychrometric chart is used to plot these two values by finding their intersection and then following the horizontal line to the right to determine the moisture content in grains per pound.

Alternatively, the table below shows the grains of moisture per pound of dry air at standard atmospheric pressure at relative humidity ranging 10 to 90%. For other values the Psychrometric chart must be utilized.

TABLE – I

Water Vapor in Air (grains water vapor/lb dry air)

Relative Humidity (%)	Temperature (ºC / ºF)							
	-1ºC	4ºC	10ºC	18ºC	22ºC	27ºC	32ºC	38ºC
	30ºF	40ºF	50ºF	60ºF	70ºF	80ºF	90ºF	100ºF
10	3	4	6	8	11	16	21	29
20	5	7	10	16	21	30	42	58
30	7	10	14	22	34	46	65	87
40	9	14	20	30	44	62	85	116
50	12	16	26	39	55	78	108	147
60	14	18	32	48	66	92	128	176
70	17	20	38	54	78	108	158	208
80	19	22	42	62	88	125	173	
90	21	24	48	70	100	140	195	

The ASHRAE fundamental handbook provides the outside air moisture content for various locations. The table below provides some information.

OUTSIDE AIR MOISTURE CONTENT TO BE REMOVED

	City	Gr.		City	Gr.		City	Gr.		City	Gr.
AK	Anchorage	59	IN	Fort Wayne	121	NV	Las Vegas	82	TX	Lubbock	111
AL	Birmingham	126		Indianapolis	130		Reno	59		Odessa	109
	Mobile	137	KS	Wichita	120	NY	Albany	109		San Antonio	128
AR	Little Rock	132	KY	Louisville	125		Buffalo	108	UT	Salt Lake City	76
AZ	Phoenix	102	LA	Baton Rouge	136		New York	121	VA	Norfolk	124
CA	Long Beach	91		New Orleans	143		Rochester	116		Richmond	130
	Los Angeles AP	96		Shreveport	134		Syracuse	110		Roanoke	113
	Sacramento	72	MA	Boston	112	OH	Cincinnati	120	VT	Burlington	106
	San Diego	103	MD	Baltimore	120		Cleveland	116	WA	Seattle	71
	San Francisco AP	67	ME	Caribou	102		Columbus	119		Spokane	61
	Santa Barbara	85		Portland	106	OK	Oklahoma City	125		Yakima	63
	Stockton	72	MI	Detroit	114	OR	Eugene	73	WI	Green Bay	117
CO	Denver	78		Flint	117		Portland	72		Madison	115
CT	Hartford	111		Grand Rapids	116	PA	Erie	114		Milwaukee	115
DC	Washington Nat'l	129		Sault St. Marie	102		Philadelphia	124	WV	Charleston	120
DE	Wilmington	121	MN	Duluth	98		Pittsburgh	116		**CANADA**	
FL	Daytona Beach	137		Rochester	120		Scranton	114			
	Fort Myers	143		St. Paul	114	RI	Providence	114	AL	Calgary	69
	Jacksonville	134	MO	Kansas City	126	SC	Charleston	136	BC	Vancouver	76
	Miami	137		St. Louis	132		Columbia	122	MN	Winnipeg	97
	Orlando	136	MS	Jackson	136	SD	Sioux Falls	119	NB	Saint John	87
	Pensacola	143		Meridian	127	TN	Bristol	118	NF	St. John's	89
	Tallahassee	136	MT	Billings	70		Chattanooga	126	NS	Halifax	100
	Tampa	136	NC	Cape Hatteras	142		Knoxville	124	ON	Ottawa	101
GA	Atlanta	123		Charlotte	122		Memphis	132		Sudbury	93
	Augusta	128		Raleigh	126		Nashville	126		Thunder Bay	91
HI	Honolulu	117	ND	Fargo	109	TX	Brownsville	136		Toronto	108
IA	Des Moines	122	NE	Omaha	125		Corpus Christi	141		Windsor	115
	Davenport/Moline	127	NH	Concord	109		Dallas	121	QC	Montreal	106
ID	Boise	59	NJ	Atlantic City	123		El Paso	99		Quebec	100
IL	Chicago	118		Newark	121		Fort Worth	123	SK	Regina	80
	Rockford	119	NM	Albuquerque	80		Houston	135			

Migration Factor (F1)

The rate of infiltration is a function of the magnitude of imbalance between the outside absolute humidity and that inside the conditioned space. The greater the difference, the greater shall be the driving force to make the vapor pressures equal. The migration factor compensates for this influence. The Table-II below provides some industry standard values, which can also be approximated by $\Delta G \div 35$.

TABLE - II

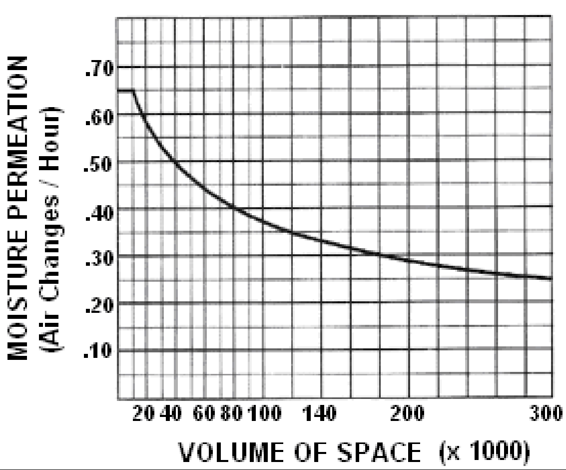
MIGRATION FACTOR

Gr/lb Difference	F_1 Factor
35	1.0
40	1.12
50	1.35
60	1.59
70	1.82
80	2.06
90	2.29
100	2.53
110	2.76
120	3.00

Air Change Factor (F2)

According to ASHRAE, the median number of air changes per hour is 0.5. The actual number of air changes is influenced by several factors; the most dominate being the size of the room. The larger the room the longer it takes to convert one volume. The Table – III below compensates for the reduction in infiltration /permeation on larger or smaller volumes.

TABLE - III



AIR CHANGE FACTOR (Approximate)

VOLUME (Cu-ft)	AIR CHANGES / Hr
Less than 10000	0.65
10001 - 20000	0.60
20001 - 30000	0.55
30001- 40000	0.50
40001 - 60000	0.45
60001 - 100000	0.40
100001 - 200000	0.35
> 200000	0.30

Construction Factor; (F3) and Barrier Factor; (F4)

Another primary factor is the amount of moisture that is allowed to permeate through the walls, floor and roof. The construction factor takes into account the effect good vapor barriers and construction materials will have on the moisture migration.

Table 4 gives factors for common construction materials. This factor will vary between 0.3 and 1.0. A composite wall must be modeled and a factor estimated.

TABLE - IV

CONSTRUCTION FACTOR

Construction Type	F_3 Factors
Masonry or Frame Construction	1.00
Sheet metal, steel welded	0.30
Module panel, caulked & sealed	0.50
Glass	0.30

TABLE - V

BARRIER FACTOR

Vapor Barriers	F_4 Factors
Laminated, mylar vapor wrap (metallic or polyethylene film)	0.50
Sheet metal seal	0.50
Two layers edge sealed moisture paper	0.67
Two coats vapor proof paint	0.75

Note: If the product of $F_3 \times F_4$ is less than 0.5, use 0.5. If the room is completely vapor proofed, with continuous vapor barrier under the floor (or of all-metal, welded material) the factor may be reduced to 0.3.

Example:

Find the amount of moisture that will permeate the room defined below.

1) Room with 12" masonry walls
2) Two coats of aluminum paint as vapor barrier
3) Volume of room - 22,000 ft^3
4) Outside Design: 90°F db 60% RH (Read specific humidity from psychrometric chart or alternatively from Table I, which shows 128Gr/lb)
5) Required - To hold in room - 38Gr/lb

Solution:

$\Delta G = 128 - 38 = 90$

F1 = 2.29 from Table II

F2 = 0.55 from Table III

F3= 1.0 from Table IV

F4 = 0.75 from Table IV

Therefore, amount of vapor able to permeate = 22000 x 90 x 2.29 x 0.55 x 1.0 x 0.75 / 13.5 = 138545Grs/hr

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MOISTURE THROUGH INTERMITTENT OPENINGS

Another source of moisture is the opening of doors and windows to the conditioned space or other openings such as conveyor passages. In these cases, the amount of moisture is directly proportional to the frequency of the opening, the difference in indoor and outdoor moisture content and the wind velocity at the opening. Obviously the first precaution is to assure that openings are adequately vapor-sealed. Then the drying equipment must deal with the moisture load that comes into a controlled space when the door is open.

The wind velocity will be the most difficult to take into account since it will vary depending on the location of the opening with respect to the wind source. Local weather stations can provide details on the normal prevailing direction and speed. However, a guideline is 12 CFM of outside air per square feet of opening. Assuming that the door is open only for short periods, the amount of moisture can be estimated by the following formula.

Moisture, lb/hr = O x A x ΔG x 12 / (C x 7000)

Where

O = time in minutes, the area is open per hour

A = area of the door opening in square feet

ΔG = Difference in specific humidity in grs/lb between controlled space and the ambient. See Table II for outside conditions and specific humidity

C = 13.5 (conversion factor for cu-ft / lb)

7000 = Conversion factor for grains/ lb

When this equation is used for a fixed opening such as a window, the minute's open/hr will equal 60 or alternatively if the door opening is specified in number of times each hour the door is opened, the amount of moisture can be calculated using equation:

Lb/hr = O x A x ΔG x F1/ (K x 7000)

Where

O = number of times each hour the door is opened. (If unknown assume personnel door to be opened 2 times/hr for every occupant)

A = area of the door opening in square feet

ΔG = Difference in specific humidity in grs/lb between controlled space and the ambient. See Table II for outside conditions and specific humidity

K = 7 (constant)

F1 = Factor from Table I for moisture difference

Example:

Door area - 3' x 7'

Door opens - 6 times each hour

Moisture difference - 90Grs/lb

Solution:

Grains per hour of additional load = 6 x 21 x 90 x 2.29 / (7 x 7000) = 0.53 lbs per hour

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MOISTURE ORIGINATING IN THE CONTROLLED SPACE

Moisture or vapor originating in the controlled space comes from any of several sources, depending on the intended use for the space. Three basic sources of moisture originating in space could be remembered as 3 P's:

- Population load, including people and animals
- Product load, brought in by the product

- Process load

Population Load: People working in an area add moisture to the air because of breathing and the evaporation of perspiration. When animals occupy the controlled space, moisture release is contributed by their excrement. How much moisture do people or animals add to a controlled space? This is a function of the number of people and their activity: a worker lifting boxes will generate 4 to 8 times the moisture of a worker at a lab bench.

Evaporation Rates of People

Work Type	Evaporation Rate (lb/person/hr)
Seated in theater	0.10
Office work, light work	0.20
Medium factory work	0.475
Heavy factory work	0.965
Athletics	1.095

Product Load: Any material manufactured in a controlled area can bring moisture with it and then release the moisture into the work area. If the product has an affinity for water, then it may also release the water in the conditioned room. For example, wet wood brought into a conditioned warehouse will release the water at a specific rate. This can be determined by measuring the products weight loss over time.

Process Load: The manufacturing process itself may expel moisture into the atmosphere of a controlled space. Open tanks or cooking vessels will add to the moisture load. Other contributors include open stream exhausts, unvented combustion cycles, and aging or curing cycles.

In the case of, open water tanks, the evaporation rate can be calculated with the following equation.

$LB/HR = 0.1 \times A \times (VP_{H2O} - VP_{AIR})$

Where:

A = Surface area of water (square feet).

VP_{H2O} = Vapor pressure of water at water temperature

VP_{AIR} = Vapor pressure of air at its corresponding dew point.

The above equation assumes 10 to 30 FPM air velocity in room.

Vapor pressure table is indicated below and all other values can be obtained from technical publications.

FLUID TEMP. F°	VAPOR PRESSURE in. HG	FLUID TEMP. F°	VAPOR PRESSURE in. HG	FLUID TEMP. F°	VAPOR PRESSURE in. HG	FLUID TEMP. F°	VAPOR PRESSURE in. HG	FLUID TEMP. F°	VAPOR PRESSURE in. HG
35	.20356	49	.34937	63	.58041	77	.93589	91	1.46824
36	.21181	50	.36264	64	.60113	78	.96733	92	1.51471
37	.22035	51	.37636	65	.62252	79	.99970	93	1.56248
38	.22920	52	.39054	66	.64454	80	1.03302	94	1.61154
39	.23835	53	.40518	67	.66725	81	1.06728	95	1.66196
40	.24784	54	.42031	68	.69065	82	1.10252	96	1.71372
41	.25765	55	.43592	69	.71479	83	1.13882	97	1.76685
42	.26781	56	.45205	70	.73966	84	1.17608	98	1.82141
43	.27831	57	.46870	71	.76528	85	1.21445	99	1.87745
44	.28918	58	.48589	72	.79167	86	1.25388	100	1.93492
45	.30042	59	.50363	73	.81882	87	1.29443	101	1.99397
46	.31206	60	.52193	74	.84684	88	1.33613	102	2.05447
47	.32408	61	.54082	75	.87567	89	1.37893	103	2.11661
48	.33651	62	.56032	76	.90533	90	1.42298	104	2.18037

VENTILATION & MAKEUP AIR

Ventilation air typically introduced through air conditioning and heating units contribute to the moisture load.

This is especially important in the summer months when high humidity is common. The outside air can be takes as 20CFM per person multiplied by expected occupancy in accordance with the requirements of ASHARE 62 guidelines. State or local codes may supersede this figure. Code compliance is especially important for hospitals and health care facilities or for any application that exposes humans to toxic fumes. Refer to codes and/or the Industrial Ventilation handbook. The formula for calculating moisture load is:

LB/HR Moisture = N x CFM/person x ΔGR x 60/ (7000 x 13.5)

N = Number of persons

CFM = Volume of outside air introduced (20CFM per person, per ASHRAE 62 guidelines on indoor air quality)

ΔG = Difference in specific humidity in grs/lb between controlled space and the ambient. See Table II for outside conditions and specific humidity

60 = Conversion factor for min/hr

13.5 = Conversion factor for CU.FT./LB

7000 = Conversion factor for GR/LB

SECTION-6 SIZING THE DEHUMIDIFIER

Once the amount of moisture to be removed is calculated, the following equations shall be used to calculate air handling and dehumidifier sizes:

<u>Equation#1</u>

The amount of work done on an air stream to remove moisture:

W = 4.5 x q x (M1 – M2)

Where

W = work in grains (gr) of water vapor/hr, (Gr/hr) or divide by 7000 for Lbs/hr

q = *air volume flow (CFM, cubic feet per minute)*

M1 = moisture into the process, gr/lb of dry air

M2 = moisture out of the process, gr/lb of dry air

Equation #2

The amount of energy required to change the temperature of the air (with no change in moisture) is calculated from equation 2:

The sensible heat in a heating or cooling process of air can be expressed as:

$E = 1.08 \times q \times (T1 - T2)$

Where

E = sensible heat for temperature changes only (Btu/hr)

q = air volume flow (CFM, cubic feet per minute)

T1= Temperature into the process, (ºF)

T2= Temperature out of the process, (ºF)

Equation#3

The total amount of energy required to change the temperature and humidity of the air is calculated from equation#3. For refrigeration systems, 1 ton of refrigeration equals 12000 Btu/hr.

$E_t = 4.5 \times q \times (h1 - h2)$

Where

E_t = total energy or energy for temperature and humidity change, (Btu/hr)

q = air volume flow (CFM, cubic feet per minute)

h1 = enthalpy into the process (Btu/lb dry air)

h2 = enthalpy out of the process (Btu/lb dry air)

Equation#4

For refrigeration systems, 1 ton of refrigeration equals 12000 Btu/hr, thus refrigeration capacity

$E_{TR} = E_t / 12000$

Where

E_{TR} = Capacity in tons of refrigeration

E_t = total energy or energy for temperature and humidity change, (Btu/hr)

SAMPLE CALCULATION

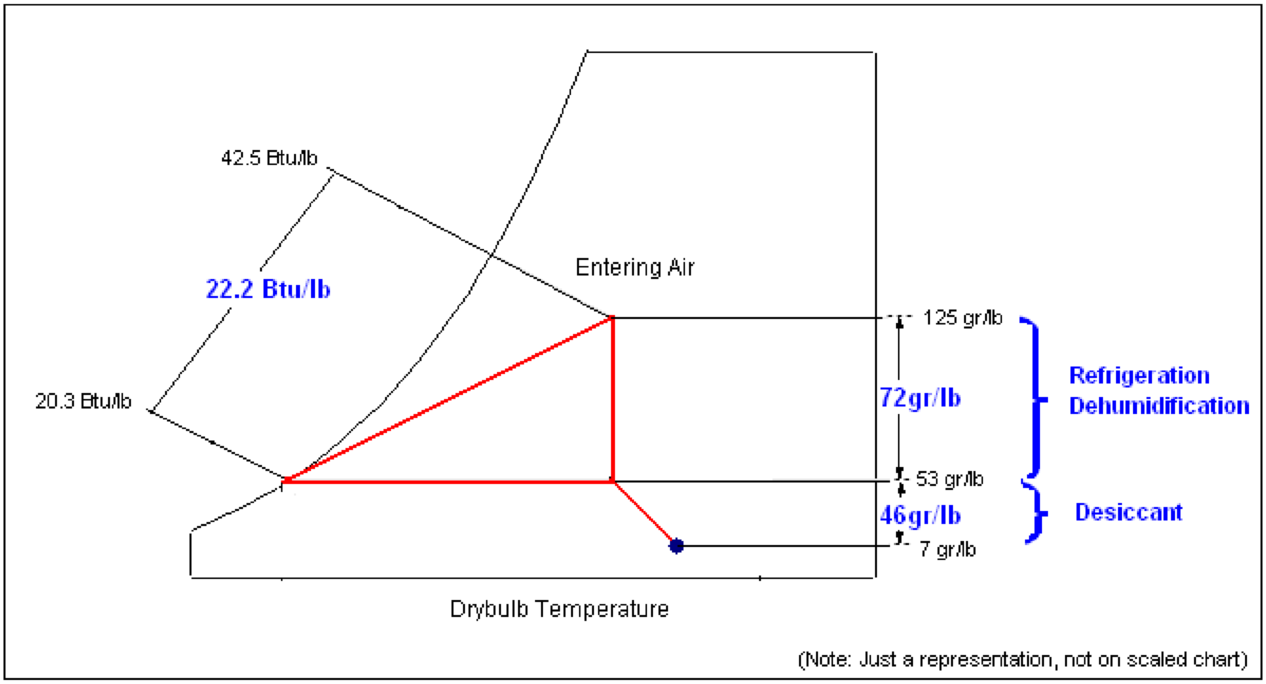
Consider a stream of air must be cooled from 95ºF at 50% RH to 50ºF at 100%RH. Calculate the amount of refrigeration capacity needed for the air stream on a per 1000-scfm basis and the amount of moisture to be removed.

Solution

From psychrometric chart, plot the plot these values by finding their intersection.

Follow the horizontal line to the right to determine the moisture content in grains per pound.

Follow the inclined line to the left to determine the energy contained in the air at each state point (enthalpy)



The results show:

M1 = 125 gr/lb

h1 = 42.5 Btu/lb

M2 = 53 gr/lb

h2 = 20.3 Btu/lb

Et = 4.5 x 1000 x (42.5 – 20.3)

Et = 99900 Btu/hr

Et = 99900/12000 = 8.3 tons of refrigeration

Amount of moisture removed; W = 4.5 x 1000 x (125 – 53) = 324000gr/hr or 46 lb of water /hr.

In some processes, the air must be dehumidified beyond what a refrigeration system can accomplish. Say in the example above, if the moisture in the air must be further removed to 7gr/lb, the moisture removed or work done by the desiccant is calculated by

W = 4.5 x 1000 x (53 – 7) = 207000 gr/hr or 29.6 lb of water /hr.

As a rule of thumb, the heat required for reactivation is approximately 2050 Btu/lb of water removed. Therefore, it would take approximately 29.6 x 2050 = 60680 Btu/hr of energy. Now the volume of air required for reactivation can be calculated using equation # 2 above

E = 1.08 x q x (T1 – T2)

Consider heating the reactivation air from the outside air condition of 95ºF to 250ºF, the volume of air required to regenerate and dry the desiccant is

q = 60680 / [1.08 x (250 – 95)] = 362 CFM

In the desiccant system, fewer pounds of water are removed, but the moisture is removed in a portion of psychrometric chart that refrigeration based technology could not have done effectively because 7gr/lb corresponds to a 7ºF dew point, which is far below the freezing point of 32ºF. This is why it is important to use a refrigeration system where it is efficient and a desiccant system to dry the air beyond a refrigeration system's capability when necessary.

SECTION-7 MOISTURE CONTROL IN BUILDINGS

Buildings in hot/warm and humid climates, with high space latent loads, high outside air ventilation rate, low interior space humidity requirement or stringent humidity control, will need dehumidification. These types of buildings include supermarkets, hospitals, labs, clean rooms and theaters.

Understanding the Functions of the HVAC System

In any climate, the HVAC system designer must consider the four basic functions of the HVAC system:

- Comfort control (temperature, relative humidity, dehumidification and air motion)
- Ventilation (proper distribution of makeup air to the interior spaces)
- Contaminant control (filtration)
- Pressurization (makeup and exhaust)

By design, these functions overlap. For example, ventilation helps provide contaminant control and pressurization. In humid climates, two of these functions are essential—providing proper dehumidification (a component of comfort control) and positive pressurization—to avoid moisture problems.

TEMPERATURE AND HUMIDITY CONTROL

It is important to understand that there are two components of performance to any air conditioning system. The first component relates to reducing the dry bulb temperature or "sensible energy" of the air. The second component of performance is the removal of moisture or the "latent energy component" in the air. In air-conditioning, the parameter of performance rating dry bulb/latent energy removal is called the sensible heat ratio (SHR). This term identifies that portion of sensible and latent removal that is provided by a particular system. Most systems today operate with a sensible heat ratio of 0.8 to 0.95 meaning that the larger portion of energy consumption of the system is devoted to sensible cooling (80 -95%) while the remaining 0.05 to 0.5 of the ratio (5-20%) is dedicated to latent heat removal or humidity control.

In modern buildings the need for dehumidification (latent heat removal) has considerably increased while less dry bulb cooling is needed today than they ever did before. This change in building SHR requirements is due to the great strides we have made in increasing energy efficiency in buildings. Initiatives like better wall and roof insulation, better window performance, more energy efficient lighting and appliances, have all reduced internal heat loads and therefore the sensible cooling requirements for buildings. At the same time, those factors that affect the moisture or latent load in buildings such as infiltration, higher outdoor ventilation requirements and the presence of people, have not changed. The reduced sensible cooling requirement for buildings has meant that air conditioning systems shall be designed for effective moisture removal and not just on dry bulb temperature control.

DESIGNING FOR PROPER DEHUMIDIFICATION

Dehumidification, or removal of moisture from the supply air, is accomplished in most buildings by cooling the air below its dew point and condensing moisture from the air. In a typical building, the supply air is cooled to about 52°F as it leaves the air handling unit and is warmed to a temperature of about 75F as it is diffused into the space. Cooling air to 52°F ensures RH in space to be 50-60%. HVAC systems are typically designed to ensure that they meet indoor comfort conditions at peak cooling loads but seldom does the air conditioning system operate at peak load. At most of the time the air-conditioning system operate at 50-60% loading implying that there is low demand for dry bulb sensible cooling, but the latent load is still at the peak. How to handle such situations?

Well there are number of ways but the conventional technique is to cool the air to 52°F as it leaves the air handling unit and provide terminal reheat before it is fed into the space. Reheat is very wasteful. First, excess energy is expended to cool the supply air, then, still more energy is expended to partially re-warm the same air before it enters the space. During low-load conditions, much more energy may be cancelled out in this manner than actually enters the space to provide cooling or heating.

These problems can be resolved by careful design, correct air pressurization in buildings, adequate dehumidification capability, proper control and operation of HVAC systems.

Research has also shown that many air conditioner designs, if turned on and off frequently, will actually evaporate moisture back into a space before condensed moisture can drain off cooling coils. This on/off cycling can further reduce the moisture removal capabilities of existing equipment. Never oversize the refrigeration equipment; oversized heating and cooling equipment will cycle on and off more frequently and for shorter periods, leading to poor latent removal performance and frequent temperature swings. In summer, this is particularly uncomfortable because short cycling prevents adequate dehumidification.

The common practice of setback control, which often means turning cooling equipment off on nights and weekends to conserve energy, is another serious contributor to excessive moisture problems in buildings in humid climates. When the cooling system is off, infiltration will carry in substantial levels of outside moisture that get absorbed in building components. This combined with low sensible cooling loads means that the system never runs long enough to remove all the moisture in the air and building components. If the dry bulb temperature is low enough during operating hours, then high moisture levels and condensation may be widespread on critical surfaces.

DESIGNING FOR PROPER PRESSURIZATION

Pressurization in buildings is the second, and equally important, issue for producing successful buildings in humid climates. A building must be positively pressurized relative to the outside; that is, some conditioned air from inside the building must exfiltrate through the envelope to the outside.

Positive pressurization is normally achieved by providing more outdoor, or makeup air to a building than is exhausted from the building. An HVAC designer typically provides 10 to 15 percent more outdoor air to a building through the air handling units than is exhausted through restroom, kitchen, or laboratory exhaust systems.

Areas that are positively pressured tend to exfiltrate excess outside air, and areas of negative pressurization have large amounts of infiltration. At best, this infiltration and exfiltration can cause a tremendous energy waste to cool and dehumidify the extra outside air. In the case of infiltration, when moist outside air comes into contact with cool building surfaces inside the wall cavity, condensation and moisture accumulation will occur. The amount of moisture accumulation can be quite large. For example, the infiltration of 1 cubic foot per minute (CFM) of outdoor air in Orlando, Florida, can bring in nearly 20 gallons of unwanted moisture per year. In its worst case, infiltration will create severe humidity problems, mold growth, and deterioration of building materials.

ASHRAE handbooks provide guidance on mechanical ventilation, and design and control of interior relative humidity conditions to control microbial growth, to minimize condensation potential, and to provide occupant comfort, in relation to air leakage.

CONTROL OF HUMIDITY

There are plenty of approaches that can be used to better control the latent energy or moisture inside buildings.

1) **Overcooling with Reheat:** Air traveling through the air conditioning equipment can be overcooled to remove additional moisture and then reheated. While some energy codes discourage this approach, there are now some innovative technologies using heat recovery methods to provide the reheat (e.g. heat pipes and rejected heat from cooling cycles).

2) **Airflow Reduction:** By reducing the flow of air through conventional cooling equipment, latent removal capability can be increased. However, too much flow reduction can cause freezing in the coil.

3) **Enthalpy Heat Recovery:** Enthalpy recovery heat exchangers can be used to dry incoming outside air with dry air being exhausted from buildings.

4) **Chiller Control:** In large buildings that utilize chillers, cooling systems can operated to vary the SHR in real time depending on need.

5) **Variable SHR Equipment:** Many manufacturers are now making unitary/ package cooling equipment for residential and commercial applications that can provide adjustable SHR output.
6) **Dedicated Outdoor Air Systems:** In many cases buildings are now using dedicated systems that dehumidify and cool outside air separately from the space conditioning system.
7) **Optimized Outdoor Air Control:** Outdoor air can be one of the major sources of moisture in a building. CO_2 based ventilation control can regulate the amount of air into a building to provide real time control of outdoor air ventilation based on occupancy. This ensures that excessive levels of air and humidity are not introduced to the building to add to the moisture removal load.
8) **Setback Control Based On Moisture Control:** Cooling systems should not be turned off during unoccupied hours or days. A moderate setback should be used that will control humidity in the space or the space cooling during setback periods should be controlled based on latent energy concentrations in the space.
9) **Economizer Control Based on Dew Point:** Controlling fresh air economizers based on temperature or enthalpy can cause excessive levels of moisture to enter a building. As a result many buildings, including most government buildings are switching to dew point control. The US GSA has established that fresh air economizers should only operate until a 51ºF dew point is reached.

SYSTEM DESIGN ALTERNATIVES

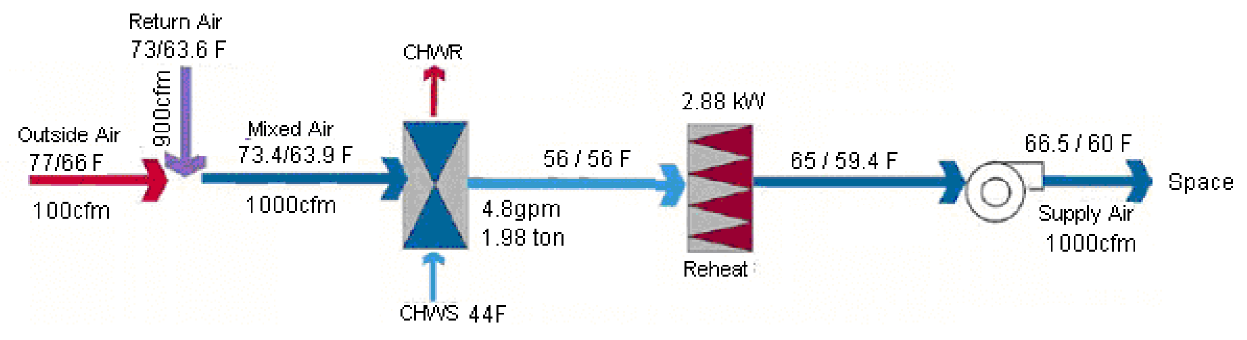
A variety of dehumidification systems have been developed to improve the energy efficiency of conventional reheat systems. Their target is zero reheat and zero overcooling in the dehumidification process by energy recovery, recycling, reuse and load reduction. These systems include:

1) Conventional reheat systems
2) Run-around coil systems
3) Heat pipe systems
4) Dual-path systems
5) Desiccant systems

CONVENTIONAL COOLING SYSTEMS WITH REHEAT

Install conventional systems in applications with low latent loads, with no requirements for indoor air quality or humidity control, and where low first cost is a high priority. Consider the use of cooling-coil face and bypass dampers or cold air distribution to reduce the need for reheat.

With conventional dehumidification technology (figure below), warm humid air, flows through a cooling coil where it is cooled and dehumidified. If required by indoor space thermostat setting, the dehumidified and cooled air is reheated through a heating coil prior to entering the conditioned space. The cooling coil can be powered by chilled water from central chiller plant or it can be a direct expansion refrigerant coil. Reheat may not be needed at peak load conditions but is often needed at low-load conditions with higher latent load fraction.



CONVENTIONAL COOLING WITH REHEAT

Applicability

Conventional reheat systems are most applicable in buildings with:

1) No requirements for humidity control of supply air
2) Dry climates where sensible cooling dominates
3) Low outside air ventilation rate and low space latent load
4) Space relative humidity settings of 55% and higher
5) Low- or no-cost waste heat, steam or hot water available for reheat use

Benefits

Benefits of conventional cooling with reheat include:

1) Simple system configuration
2) Good humidity control by adjusting the off coil air temperature
3) Low initial cost

Design Details

Conventional systems often double the total energy use because of overcooling and reheating of the supply air. For most climates, there will be long time periods when moisture removal is required but the internal sensible cooling load is significantly lower than the design conditions used to size the air conditioning. The dehumidifier with variable reheat can be set up to discharge cool air into the space when the room is occupied, often meeting the internal sensible cooling load without energizing the main air conditioning system. Variable reheat, with a narrower range of leaving air temperature, allows the designer to proceed with more flexibility and can achieve significant operational savings, and often a smaller capacity air conditioning system can be installed.

The need for reheat can be reduced by adopting variable reheat using any of the following approaches:

Dehumidifier with Face and Bypass Damper: Use a bypass damper in parallel with the cooling coil. The face and bypass damper system regulates the amount of air that flows over the reheat coil and so controls how much heat is transferred to the air stream. This allows a portion of the air to be cooled to a low temperature and dehumidified, and then remixed with the bypass air. The total moisture removal is greater than if all the air passes through the cooling coil but is not cooled to as low a temperature. This system controls the discharge temperature to a specific value by regulating the damper positions with a feedback controller.

Dehumidifier with Staged or Stepped Reheat: Stepped or staged reheat will use either a series of solenoid valves or an incremental stepped valve to adjust the amount of refrigerant allowed to the reheat condenser. Adjustments are made in specific increments, such as 20% of refrigerant volume flow. These systems provide an intermediate level of control of the reheat condenser. Leaving air temperatures can be held in a narrower range (±5°F) than with a fixed reheat system, but can not be held at a specific level.

Dehumidifier with Variable Hot Gas Reheat: A variable hot gas reheat system relies on an infinitely controlled valve to allow specific amounts of hot gas to the reheat condenser. This system very closely matches the leaving air temperature to the user's setpoint. A sensor providing leaving air temperature information to the controller is required. The controller typically uses proportional/integral control logic to position the hot gas throttling valve.

Use cold air distribution and adjust supply air volume with variable air volume controllers: Another means to reduce reheat requirements is to use variable air volume (VAV) boxes along with variable frequency drive (VFD) fan motor at the air handler. The supply air is delivered at 50ºF or lower, instead of the typical 55ºF and the supply air volume is adjusted instead. Additional benefit is that the lower airflow results in less fan energy however the downside is that lower chilled water temperatures are necessary and chiller energy consumption may increase. This approach requires careful selection of diffusers to maintain comfort and is also more susceptible to condensation on ductwork.

Costs

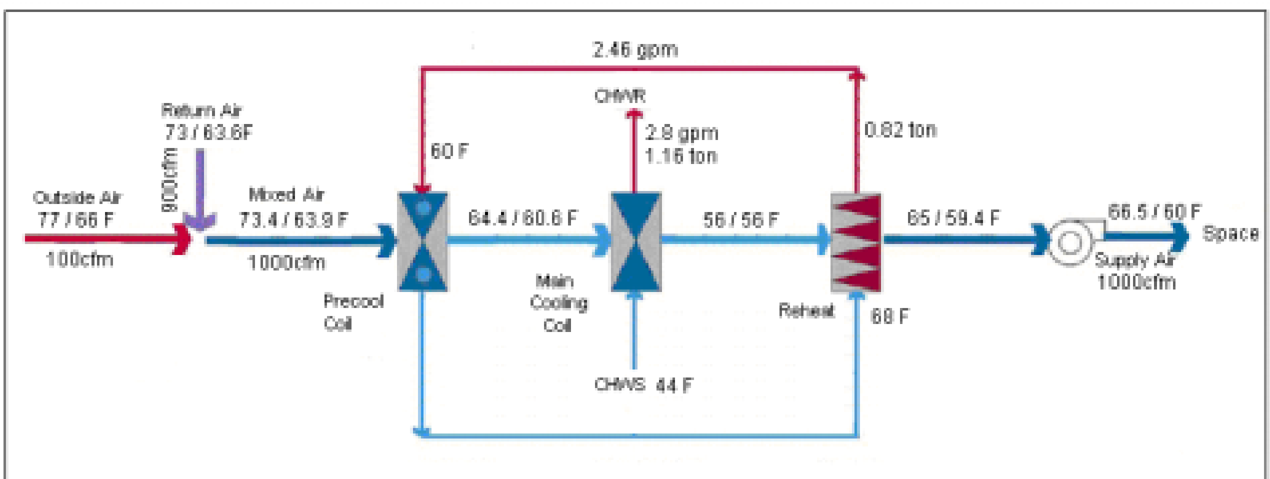
Conventional air handling systems with reheat cost approximately $4.00 to $5.00/cfm. Conventional reheat systems may have a lower initial cost but their operating costs are much higher because energy is wasted in overcooling and reheating the supply air.

RUN-AROUND COIL SYSTEMS

Install run-around coils in applications with large dehumidification requirements where the air must be reheated after passing the cooling coil.

A run-around coil system is a simple piping loop with an upstream precooling coil and a downstream reheating coil that sandwiches the main cooling coil. The circulating fluid is pumped to transfer heat from the warm mixed air to the off coil cold supply air. The run-around system reduces the cooling load on the main cooling coil; reheat is provided by the heat picked up by the circulating fluid in precooling coil instead of by an external source of expensive energy.

In new building designs and retrofits, a run-around system can reduce peak heating and cooling loads as well as total heating and cooling energy. The run-around system can have a significant impact on the heating and cooling capacity in new HVAC designs.



RUN - AROUND LOOP COOLING SYSTEM

The heat recovery effectiveness of the run-around loop is defined as the ratio of the actual heat transfer to the maximum possible heat transfer between the air streams. This is equivalent to the ratio of the difference between the mixed air temperature and the air temperature off the precool coil to the difference between the mixed air temperature and the air temperature off the main cooling coil. The effectiveness ranges from 50% for a normal loop to 65% for a high performance loop. Because of the relatively small temperature differences between the energy exchange coils, low approach cooling coils should be used. Designers must account for the additional pressure drop from the added coil.

Run-around coil systems are most applicable in situations requiring substantial dehumidification.

Benefits

Benefits of run-around coil systems include:

1) Lower cooling load contributes to a smaller cooling system and less pumping energy use, but fan energy increases due to extra air pressure drop through the run-around coils.
2) Reheat energy is saved
3) Lower total energy use

Design Details

The increased dehumidification capacity provided by runaround coils allows for a smaller cooling system. However, the addition of coils will increase the pressure drop, and fan power must be adjusted accordingly.

The run-around loop can either be applied to existing systems or can be installed at the factory. The run-around loop requires a fractional horsepower pump, a 120V–60HZ single-phase electrical circuit, and a three-way valve or a variable-speed drive (VSD) for the pump. For bigger systems, an expansion tank with air vent may be needed.

Costs

The initial cost of a run-around system is about double that of a conventional system, but if the downsizing of the chiller and cooling tower is counted, the total initial cost will be very close. The total installation cost is approximately $4.50 to $5.00/cfm.

The cost effectiveness of a run-around system depends on the system it is replacing. When used instead of a dehumidifying system requiring reheat, the simple payback is about two to three years. However, when the system replaces a system without reheat (no humidity control), there are additional benefits including increased comfort and enhanced indoor air quality, which are difficult to quantify.

Operation & Maintenance

Run-around systems require extra maintenance for the two coils and the loop. Air trapped in the coils, pump and piping must be vented upon initial startup to ensure effective fluid flow and heat transfer. The pre-cooling and reheating function can be controlled by adjusting the pump speed with VSD, cycling the pump on-off, or using valve control and bypass.

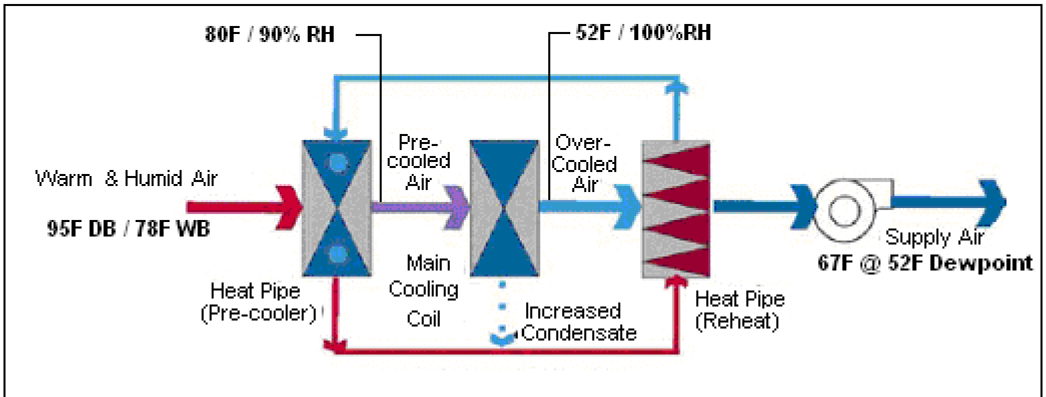
HEAT PIPE SYSTEMS

Install heat pipes in applications with large dehumidification requirements where the air must be reheated after passing the cooling coil. Heat pipes increase the effectiveness of air conditioning systems by helping to decrease the total cooling load of the air. The heat pipe system is hermetically sealed, uses a wicking action, and requires no pump.

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

Heat pipes may be described as having two sections: 'Pre-cool' *and* 'Reheat'*.* The first section is located in the incoming air stream. When warm air passes over the heat pipes, the refrigerant vaporizes, carrying heat to the second section of heat pipes, placed downstream. Because some heat has been removed from the air before encountering the evaporator coil, the incoming air stream section is called the pre-cool heat pipe. Air passing through the evaporator coil is assisted to a lower temperature, resulting in greater condensate removal. The "overcooled" air is then reheated to a comfortable temperature by the reheat heat pipe section, using the heat transferred from the pre-cool heat pipe.

Activated by temperature difference and therefore consuming no energy, the heat pipe, due to its pre-cooling effect, allows the evaporator coil to operate at a lower temperature, increasing the moisture removal capability of the air conditioning system by 50-100%. With lower relative humidity, indoor comfort can be achieved at higher thermostat settings, which results in net energy savings. Generally, for each 1ºF rise in thermostat setting, there is a 7% savings in electricity cost. In addition, the pre-cooling effect of the heat pipe allows the use of a smaller compressor.



HEAT PIPE SCHEMATIC (DEHUMIDIFICATION)

Heat pipe performance depends on the temperature difference between the cooling and the reheat coil banks. As outdoor air temperature decreases, the heat transfer across the heat pipe also is reduced so that eventually little or no reheat is available. Temperatures leaving the air conditioner with heat pipe added may vary significantly as the heat load changes.

Benefits

Benefits of heat pipe systems include:

1) Removes 50% to 100% more moisture than systems without heat pipes.

2) Saves energy compared to systems that provide similar amounts of dehumidification.

3) Simple system with no moving parts or external connections makes it basically maintenance free.

Design Details

The increased dehumidification capacity provided by heat pipes allows for a smaller cooling system. However, the addition of heat pipes will increase the pressure drop, and fan power must be adjusted accordingly.

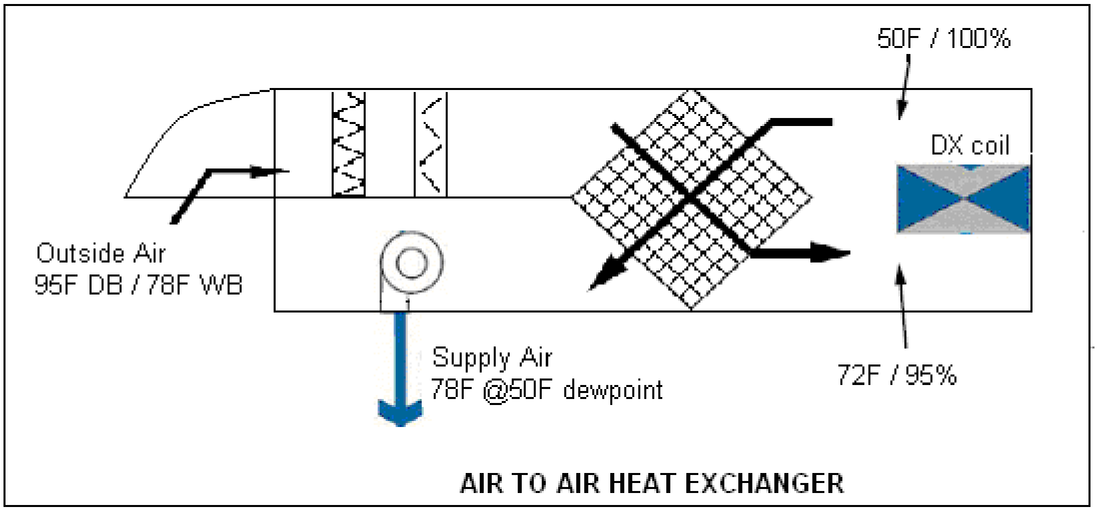
Heat pipes can either be applied to existing HVAC systems or can be installed at the factory. The heat pipe loop is usually controlled by cycling on-off or modulating the refrigerant flow with a control valve.

Costs: The installation cost of a heat pipe loop for a cooling system is approximately $2.50/cfm.

AIR TO AIR HEAT EXCHANGER

Another form of passive energy transfer when using an air conditioning unit for dehumidification of 100% outside air applications is the use of an air to air plate heat exchanger. Heat is transferred from outdoor air coming into the air conditioner to the cold air leaving it. Again, the goal is to boost the portion of air conditioning capacity used for removal of latent heat by decreasing the need to remove sensible heat. To a limited extent, the exchanger itself will remove latent heat by condensing moisture on the entering air side.

Outdoor air is introduced into one side of the heat exchanger, and is partially cooled. It then flows over the cold refrigeration coil for moisture removal and additional sensible cooling. The cold, saturated air then passes through the other side of the exchanger for warming before being introduced to the air handler or the space.



The efficiency of the heat exchanger is a function of the size and spacing of the aluminum fins and the temperature difference. As with the heat pipe, and for the same reasons, the amount of reheat and therefore the leaving air temperatures can vary significantly.

DUAL-PATH SYSTEMS

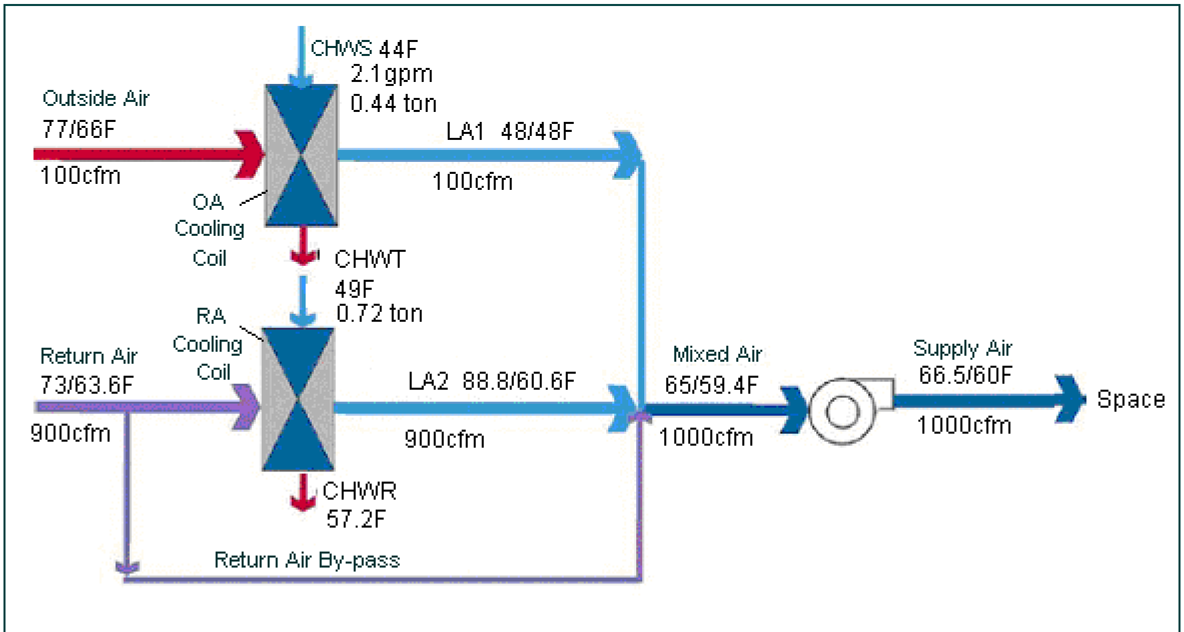
Install dual-path systems in applications with return air and large dehumidification load due to high outside air ventilation rate.

A dual-path system uses two coils (either chilled water or DX) to separately cool the incoming outside air and return air. The hot and humid outdoor air is cooled by a primary coil to 42°F to 45°F for dehumidification. The secondary coil furnishes the sensible cooling of part of the relatively cool and dry return air.

A portion of the return air may bypass the secondary coil and mix with the cooled return air stream. These two air streams are then mixed into supply air with appropriate temperature and humidity.

In systems where the fraction of outside air is low and the space latent load is high, the outside air alone may not be enough to handle the total latent load of the supply air, which requires some moisture to be extracted from the return air stream. This means a portion of the return air needs to be overcooled to extract moisture and additional reheat may be necessary to increase the air temperature for comfort supply.

One zero-reheat solution is to direct a portion of the return air to mix with the outside air before dehumidification.



DUAL PATH COOLING SYSTEM

In chilled water dual-path systems, the outdoor air (OA) coil can use cold chilled water at 40°F to 42°F for latent cooling, while the return air (RA) coil can use warmer chilled water at

50°F to 60°F for sensible cooling, thus improving chiller efficiency. Dual-path systems decouple the sensible cooling and latent cooling of the supply air, thereby improving control of temperature and humidity.

Dual-path systems are best in HVAC applications where the moisture load arises primarily from the outdoor air. These applications include commercial buildings in humid climates, schools, clean rooms, theaters, supermarkets, hotels and motels. For larger systems, separate air-handling units for outside air and return air can be used.

Benefits

Benefits of dual-path systems include:

1) Reduces the installed cooling tons over a conventional single-path system
2) Provides low operating cost with efficient cooling and no reheat
3) Provides direct control of ventilation air quantity for improved indoor air quality
4) Provides good humidity control at all times, including part load, as moisture is removed at its source, regardless of building load

Design Details

Dual-path systems avoid overcooling and reheating the supply air, thus reducing the size of cooling and heating systems. The sensible cooling of the return air can use chilled water with higher temperature to improve chiller efficiency. The additional costs of coil, duct and pipe work, and damper or VSD control must be adjusted accordingly.

Dual-path systems can be installed separately or integrated with additional HVAC/R equipment. They are currently available in factory package units for indoor and outdoor installation. The OA cooling coil should be sized for peak latent load, while the RA cooling coil should be sized for peak sensible load. The OA path controls the humidity of the supply air by modulating the chilled water flow, while the RA path controls the supply air temperature by adjusting the bypass damper position.

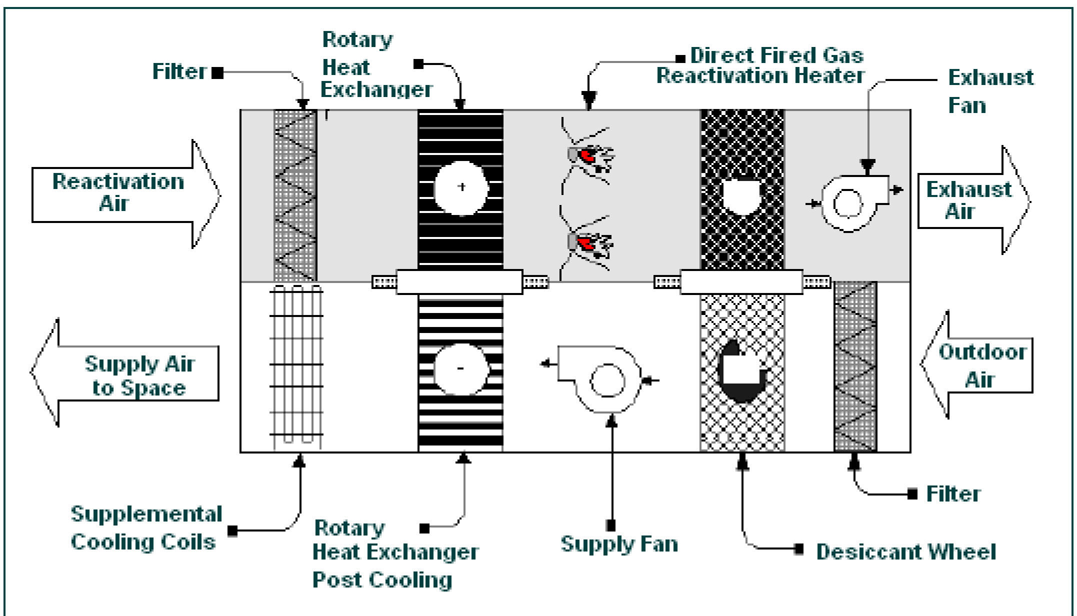
Costs: The installation price of a dual-path system varies between \$5– \$6/cfm. Dual-path systems are energy efficient while assuring an acceptable humidity level at all ventilation air volumes. Its use can also reduce demand and energy charges sufficiently to offset the higher first cost.

DESICCANT SYSTEMS

Install desiccant systems in applications requiring large dehumidification and low space humidity levels that would be difficult to achieve with cooling-type dehumidification.

Desiccant materials can absorb between 20% and 40% of their dry weight in water vapor from humid air. Both solid and liquid desiccants are used in cooling systems, but solid desiccants are much more common in commercial buildings. Liquid desiccants employ solutions such as glycol or salts such as lithium chloride ($LiCl_2$).

In solid desiccant systems, desiccant is formed in place in a honeycomb matrix wheel mounted between the process air stream and the reactivation air stream; air seals separate the air streams from each other. The desiccant wheel rotates slowly (6 to 20 rotations per hour) between the two air streams. The process airflows through the wheel, gives up its moisture to the desiccant and increases dry-bulb temperature (up to 120°F), and finally is cooled by coils for comfort supply. After drying the process air, the desiccant wheel is saturated with moisture and rotates slowly into the reactivation air. The hot reactivation air (with temperature up to 250°F typically required) heats the honeycomb, absorbs moisture released by the hot desiccant, and is released as exhaust air from the building. Desiccants are also available that can be regenerated at temperatures as low as 120ºF, allowing a greater range of options for heat sources such as heat pumps or solar sources.



DESICCANT WHEEL OPERATION

Desiccant systems often incorporate heat recovery equipment. If exhaust air is available, it can be used to cool the warm air (leaving the desiccant section) before it passes through the cooling coil. When there is no exhaust air, outside air can be used to cool this warm air, and the heated outside air can be used to reactivate the desiccant.

According to the manufacturers, a well maintained desiccant wheel will last for approximately 100,000 hours of operation (10 to 15 years) before they need replacement. Costs of desiccant systems are typically given in terms of $/CFM. For large commercial systems, the cost is approximately $5/cfm, while smaller units (less than 1000 CFM) may cost up to $8/cfm. Desiccant systems should use low-cost surplus heat, waste heat or solar heat for desiccant reactivation and where not possible, desiccant systems usually use heat from natural gas as their primary energy source. Desiccant systems use very little electric energy and can save money when the cost of power is high during the peak demand periods of summer. In large buildings where non-electric heat is available for reactivation of desiccants, desiccant systems can reduce HVAC electricity use by 30% to 60% and peak electricity demand by 65% to 70%.

The maintenance requirements of desiccant systems can be modest compared to conventional cooling-based dehumidification systems. Filters located in the inlet of process air and reactivation air needs to be cleaned or replaced every two months. About 90% of reported problems related to desiccant systems can be traced to clogged filters. The wheel can be vacuumed to remove dust from the wheel face. The drive belt around the heat wheel needs to be tight enough to turn the wheel without putting excessive load on the drive motor shaft bearings. No regular maintenance is required for the desiccant materials.

ENERGY RECOVERY ENTHALPY WHEEL

Energy recovery wheel can be incorporated into a ventilation system to transfer both the sensible and latent energy between outdoor and exhaust air streams. The core of an enthalpy wheel can be made from a variety of materials such as paper, metal or plastic, which is coated with a desiccant. This coating enables the wheel to transfer both sensible and latent energy between air streams.

The enthalpy wheel provides first stage cooling or heating. For instance, in the summer the wheel removes moisture from incoming air and pre-cools it to lighten the load on the evaporator coil. Likewise, in colder temperatures (winter application) the wheel will inject moisture into supply air and pre-warm it to reduce the load on the reheat coil. In the summer and winter modes, the wheel saves energy.

There are three general types of wheels being used today. They are sensible, enthalpy and regenerative.

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

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

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

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

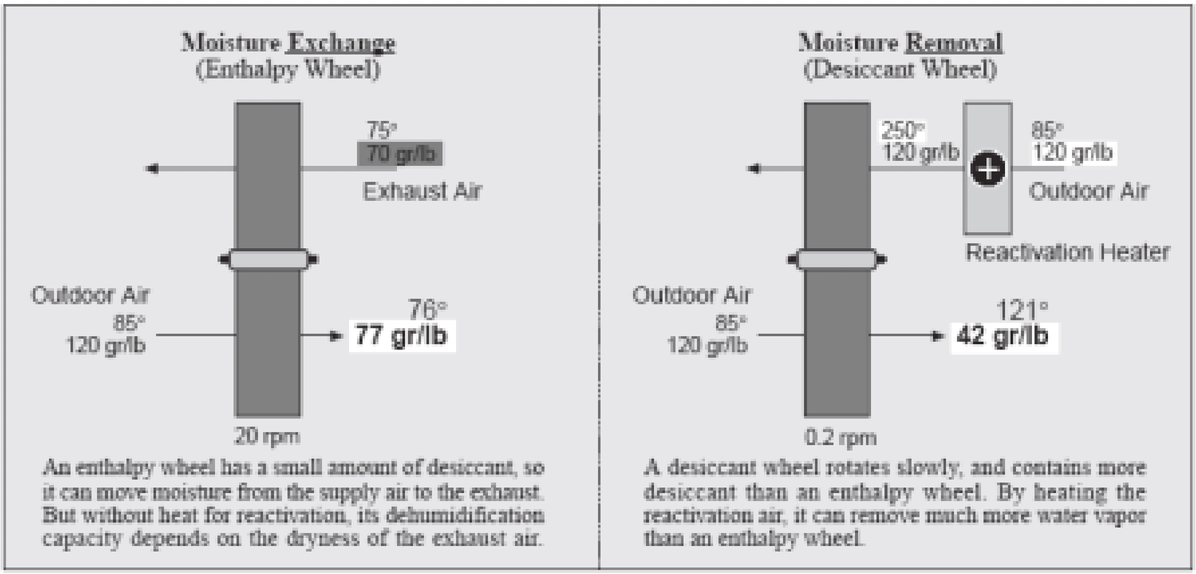
ENERGY RECOVERY V/S DESICCANT WHEELS

Desiccant wheels are often confused with energy recovery wheels. The confusion is understandable. Both devices look nearly the same, and both types are constructed with honeycomb media and contain desiccant. Also, sensible- only heat wheels are sometimes used as post-coolers in desiccant systems. But there are important functional differences between these devices which appear so similar.

Heat wheels are optimized to transfer sensible heat between two air streams, while desiccant wheels are optimized to remove moisture. These different purposes lead to differences in materials and in wheel rotation speed. An energy recovery wheel rotates at a comparatively high speed (20 rpm), to maximize the heat transfer between air streams. A desiccant wheel rotates 60 times more slowly (10 to 20 rotations per hour). The slow rotation speed allows the desiccant to adsorb more moisture, and it minimizes the amount of heat carried over from the hot reactivation air into the cooler process air.

If the exhaust air is dry, an energy recovery wheel can transfer some moisture out of the incoming air. But energy recovery wheels contain less desiccant than desiccant wheels. Also, the honeycomb material, air seals and support structure of an enthalpy wheel cannot endure the temperature and moisture differences typical of desiccant wheel operation.

Consequently, the wheels perform quite differently.



Desiccant wheels compared to energy recovery wheels

As seen in figure above, thermal energy used for reactivation allows desiccant wheels to remove much more moisture than energy recovery wheels. Desiccant wheels can deliver air below the moisture condition of the exhaust air. That level of dryness cannot be reached with energy recovery wheels.

Conclusions:

Some important observations and field results:

1) A standard cooling system does not provide enough dehumidification at low load when the supply air volume remains constant. Standard cooling with reheat provides good humidity control at the cost of higher energy consumption, which is discouraged by model energy codes. Variable air volume systems instead provide a suitable alternative to this problem.

2) Run-around coil systems and dual-path systems are energy efficient in dehumidification applications. They reduce coil loads and avoid reheat for most load conditions. The dual path system design uses a separate cooling coil for outside air.

3) Run-around systems and dual-path systems may need reheat at very low load conditions when space latent load dominates the total cooling load and the ventilation rate remains constant. A special case occurs in supermarkets where refrigerated display cases cool the store during unoccupied hours or when the store is cool due to cool weather. In this case, dual-path systems are equipped with reversing valves that allow the return air circuit to provide heating.

4) Pretreatment of outdoor air for moisture removal can provide a cost-effective way to prevent moisture from creating indoor air quality problems. When pretreatment is used, various approaches for energy recovery are available when recovery from exhaust air is not a possibility. Heat pipes, air-to-air heat exchangers, and some dehumidifiers recover energy via fixed reheat. While these methods can provide sufficient moisture removal capacity and do recover at least some energy, the level of temperature control is rather limited. The air handler must usually heat or cool to adjust the temperature.

5) Dehumidifiers with variable reheat both remove moisture and regulate the amount of hot gas reheat to control leaving air temperature. The annual energy savings over fixed reheat are significant, because the need for further heating and cooling is much decreased. The most precise control, and therefore the greatest energy savings, is provided by a dehumidifier with variable hot gas reheat.

To achieve an energy-efficient dehumidification system design, consider the following factors:

1) Size and select cooling coils with enough cooling capacity to handle the peak sensible cooling load and peak latent cooling load that occur at different load conditions. Use low approach cooling coils and low temperature water.

2) Design systems considering various load conditions rather than only the peak load condition. For conventional cooling with reheat systems, size reheat equipment to handle a higher reheat requirement in lower load conditions, especially for constant-volume systems.

3) Integrate heat recovery equipment into conventional cooling systems to reduce cooling loads and reheat energy. Runaround loop systems are much more energy efficient than conventional cooling systems, especially when operating in part-load conditions.

4) Dual-path systems offer competitive energy efficiency with run-around loop systems, and provide better control of the outside air ventilation rate. Dual-path systems decouple sensible cooling and latent cooling for easy control of the supply air temperature and humidity. Equipment is available to provide both cooling and reheat.

5) Desiccant systems are more competitive when a low supply air dew-point temperature is required, latent load fraction is high, low- or no-cost reactivation heat from steam, hot water or waste heat is available, and electricity costs are high when compared to gas costs.

6) Lay out equipment correctly. Place filters upstream of coils. Place fans downstream of coils (draw-through mode) to provide a small amount of reheat.

7) Select low face velocity coils to reduce air pressure drop and improve dehumidification performance. . *Where humidity loads are high, consider deeper (high row) evaporator coils.*

SECTION-8 CONSTRUCTION OF CONTROLLED SPACE

As much as 90 percent of all water intrusion problems occur within one percent of the total building exterior surface area. The one percent of the structure's façade contains the terminations and transition detailing that all too frequently lead to envelope failures. Construction detailing is a critical component for the success of any wall assembly. The designer must create details that effectively:

1) control rain water penetration that may occur via gravity flow, capillary action and pressure differential due to the effects of wind pressure, stack effect or mechanical ventilation that create pressure differences across the building envelope, and drive water through cracks or openings

2) control condensation that may occur via air leakage and diffusion

It is very important to take certain precautions in preparing space for the humidity control, regardless of the type of air drying equipment or the method used to do the drying. Satisfactory moisture control-better known as customer satisfaction-depends on many variables.

The Nature of Water Vapor

Consider two closed rooms, adjacent to one another. If the partial pressure of the water vapor in room #1 is greater than the partial pressure of the water vapor in room #2, then the water vapor will travel through the wall into room #2 regardless of the composition of the wall.

Let's take the hypothetical example a step further. If the absolute humidity of the air in room #1 is greater than that of the air in room #2, then the water vapor pressure will be higher in room #1. Therefore, when drying room #2, the problem of new water coming through the wall from room #1 must be considered.

A vapor barrier can slow down the passage of vapor from wet to drier areas, but it cannot keep water out; it can only slow the rate of penetration. Commercial vapor barriers-moisture resistant construction material including paints and other coatings offer a variety of design alternatives. The choice of vapor barrier is based on the degree of dryness required in the controlled space, the efficiency of the equipment being used for drying, and the cost of construction. When preparing space for the humidity control (permeation of water vapor), the following considerations are important to effectively control condensation and prevent moisture penetration:

Design Considerations

- Air permeability
- Continuity with other air barrier materials
- Structural integrity
- Durability
- Water penetration resistance
- Water vapor permeability
- Code compliance
- Climate

Air Permeability: The layers of material that make up a wall assembly have different air permeability. Energy codes in the United States have begun to require air tightness of the building envelope, but they are not specific about levels of air permeability for air barrier materials. The generally accepted level based on National Building Code of Canada requirements is 0.004 cfm/ft^2 at 1.57psf. While many common building materials like plywood and gypsum wallboard meet this standard, a sheathed wall assembly will not perform well as an air barrier unless the joints are treated with an air barrier material.

Air Barrier Continuity: Any vapor barrier must be continuous, without breaks or tears. In cold climates the breaks can allow significant amounts of warm moisture-laden air to escape from the interior environment and condense on a cold surface in the wall assembly. Conversely, in hot, humid climates, breaks in the air barrier permit moisture-laden air from the exterior environment to infiltrate the building envelope and potentially condense on a cold surface in the wall assembly.

Air Barrier Structural Integrity: Structural integrity of air barriers is important because if the materials tear or displace with wind loading, they lose their effectiveness as air barriers. Some building wraps have low air permeability, but they do not perform well when commonly installed because they have many seams. All lap joining must be tightly closed (this is particularly critical when mechanical or caulked joints are used).

Air Barrier Durability: While capable of resisting wind loads without compromise in performance, air barrier materials must also demonstrate durability in a number of other ways, particularly if the air barrier is concealed and inaccessible for maintenance. Durability criteria include:

- Resistance to puncture
- Resistance to pests—rodents, termites, carpenter ants, and other insects
- Resistance to low but sustained negative pressures from building stack effect and HVAC fan effect

- Ability to withstand stress from thermal and moisture movement of building materials, and stress from building creep
- Resistance to UV degradation (during the construction period)
- Resistance to mold growth
- Resistance to abrasion

Water Penetration Resistance: The traditional moisture protection used in wall construction is asphalt-saturated felt or kraft waterproof building paper. Building wraps are often used in place of asphalt felt in wall construction, often with the same perceived purpose. Seamless fluid applied moisture protection provides a significant improvement over traditional moisture protection and building wraps. *In fact, these can be 10 times more resistant to water penetration than building wraps and nearly 200 times more resistant to air leakage than asphalt felt*.

Water Vapor Permeability: The generally accepted definition of a vapor-retarding material is one that has a water vapor permeance of 1.0 perms or less. Table below provide Water vapor permeance of common moisture barrier and building materials.

Building Material	Water Vapor Permeance (Perm)
4 mil Polyethylene	0.08
6 mm (1/4 inch) Plywood3 (ext glue)	0.7
101mm (4 inch) Brick3	0.8
203mm (8 inch) Concrete Block3	2.4
25mm (1 inch) Expanded Polystyrene1	5
Type 15 Building Felt2	5.6
Fluid Applied Air Moisture Barrier Waterproof Coating	5.7
19mm (3/4 inch) Plaster on Metal Lath3	15
Fluid Applied Air Moisture Barrier Joint Treatment	17.3
9.5mm (3/8 inch) Gypsum Wallboard3	50

The purpose of a vapor retarder in wall construction is to minimize water vapor diffusion through the wall assembly and thus reduce the risk and the amount of condensation on cold surfaces in the wall assembly.

Whether or not a vapor retarder should be placed in a wall assembly and where it should be placed must be carefully evaluated in relation to climate, the physical characteristics of other components of the wall assembly, and interior relative humidity conditions.

1) In cold climates the predominant water vapor diffusion direction through most of the year is from the inside to the outside, as warm, humid air from the interior environment moves in the direction of cold, dry outside air.
2) Conversely, in hot, humid climates, the predominant water vapor diffusion direction through most of the year is from the warm, humid outside environment towards the cooler, dryer, air-conditioned interior environment.

Based on these general conditions, a vapor retarder is customarily placed on the interior of wall construction in cold climates and on the exterior in hot, humid climates. A vapor retarder should not be placed on the interior in hot, humid climates, since it will potentially cause condensation by restricting vapor diffusion to the interior. The use of interior vapor retarders has been shown to be a contributing cause in many cases of moisture problems and IAQ problems in buildings in hot, humid climates. Refer ASHRAE Handbook—Fundamentals, chapters 21 and 22) for further information.

Insulation between vapor barriers can be a potential problem: if construction occurs in humid weather, water can be "sealed in" between the two vapor barriers. Sealed-in vapor will travel into the controlled space and impose an extra drying load on the drying equipment. This extra load lasts only until the insulation dries out, but meanwhile, humidity control is difficult.

Code Compliance

Model building codes and state and municipal codes in the United States do not address air barriers, moisture barriers and vapor retarders in a uniform way.

1) Energy codes in the United States, including the IECC (International Energy Conservation Code), the State of Massachusetts Building Code, and ASHRAE 1999 energy conservation standard (ANSI/ASHRAE/IESNA Standard 90.1-1999, Energy Standard for Buildings Except Low-Rise Residential Buildings, an energy conservation standard which is required to be adopted by state building energy codes under the Federal Energy Conservation and Production Act) require air tightening of the building envelope. Although codes in the United States do not always provide specific limits for air leakage of air barrier materials, the generally accepted limit is 0.004 cfm/ft^2 at 1.57psf pressure [0.02 L/(s·m^2) at 75 Pa pressure] based on National Building Code of Canada requirements.
2) Most model codes generally require the use of a water-resistive barrier in wall construction and prescribe asphalt saturated felt (IBC Chapter 14, paragraph 1404.2). They often require the use of vapor retarders in wall construction (IBC Chapter 14, paragraph 1403.3) unless other means are provided to avoid condensation.
3) Southern Building Code Congress Public Safety Testing and Evaluation Services, Inc. publishes an Evaluation Guide on Floor, Wall, and Roof Systems (Testing for Moisture Protection Barriers—SBCCI PST & ESI Evaluation Guide 119), which lists specific performance criteria for air and moisture barriers, including fluid applied air/moisture barriers.

By constructing an airtight building envelope, the risk of moisture problems—decay, corrosion, loss of insulation value, mold growth and indoor air quality (IAQ) problems—which can occur because of air leakage and condensation, are minimized.
