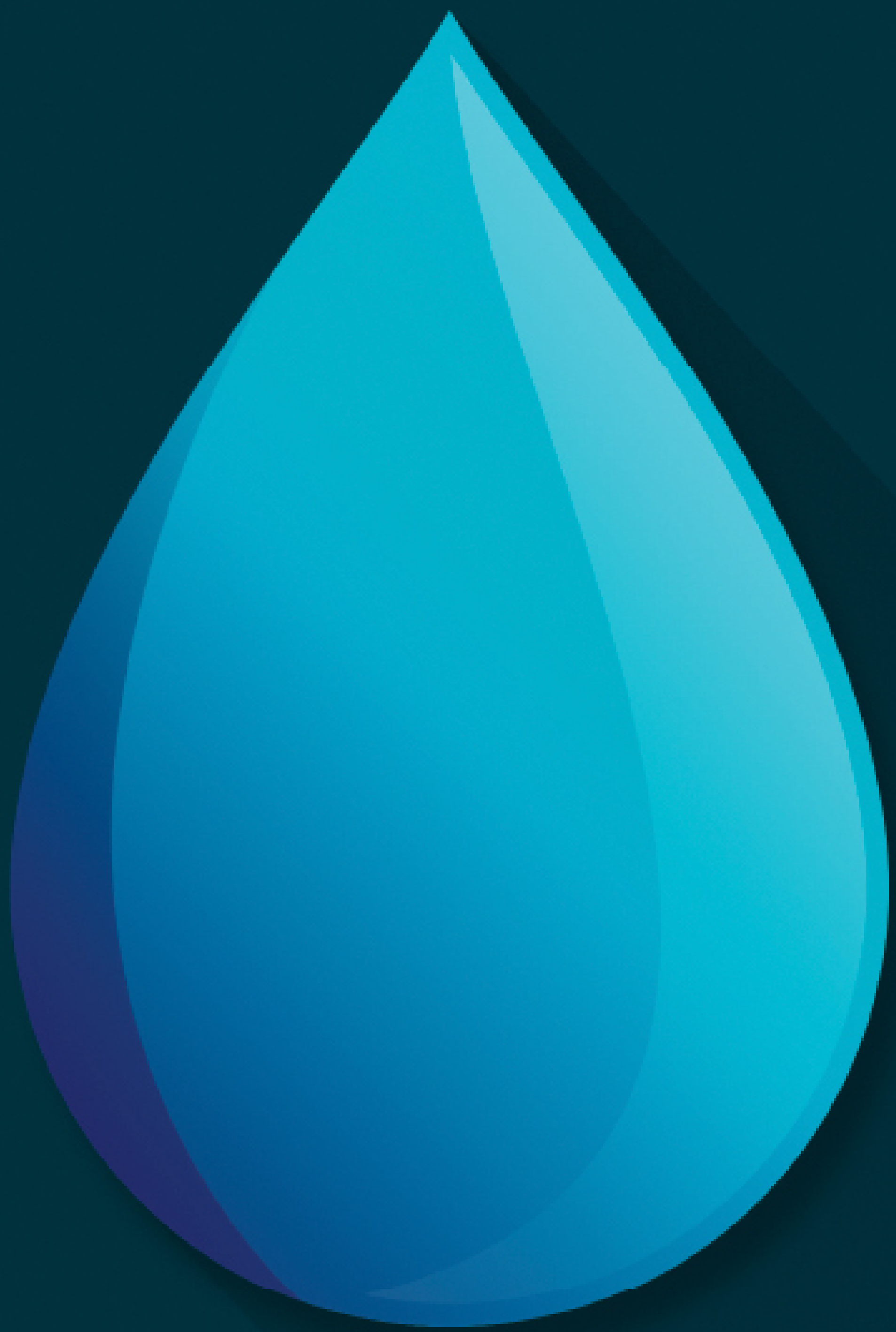
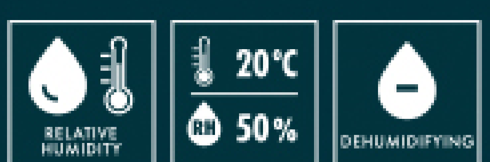
The Dehumidification Handbook

Third Edition





Munters

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This handbook explains how and why to dehumidify air. It is written for the engineer who has a basic understanding of heating and cooling systems, or one who operates a building or process which is influenced by humidity.

Drawing on the collected experience of hundreds of practicing engineers, the handbook guides the system designer by describing fundamental principles illustrated by concrete examples. This book helps the engineer balance the conflicting requirements of energy, construction budgets and maintenance considerations to design practical and economical dehumidification.

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CONTRIBUTORS

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1
INTRODUCTION

This handbook explains how and why to dehumidify air. It is written for the engineer who has a basic understanding of building heating and cooling systems, or who operates a building or process which is influenced by atmospheric humidity.

The text assumes the reader is interested in the technology, and has a need to apply it to gain some economic benefit. The text also assumes the reader is interested in specific examples as well as the theory of dehumidification. These assumptions guide the necessary compromises any book must make between accuracy and clarity, and between details which illustrate a specific case and those which illustrate general, abstract principles.

Part of the information collected here comes from other technical references which deal incidentally with dehumidification in discussions of related topics. The American Society of Heating, Air Conditioning and Refrigerating Engineers (ASHRAE) has provided helpful information through its Handbook series. In addition, many industries have provided data which can assist the dehumidification engineer working in other fields.

The primary information source is the collected experience of hundreds of engineers who deal with dehumidification and humidity control on a daily basis. These individuals have been very generous with their time and hard won understanding of dehumidification technology. Every day they must balance the opposing pressures of optimum theoretical system performance against compelling limitations of available information, time and budget. The system design examples are fictitious, but every element contained in them has occurred in the field.

In the last few years, dehumidification technology has emerged from its industrial heritage to take an expanding role in commercial and institutional building heating and cooling systems. The entire field is changing rapidly. It has been a difficult decision to limit the information here to what is well understood by the contributors and in widespread use as of the publication date. Many new applications and new equipment designs will become available in the next few years.

Undoubtedly some applications are missing from this handbook because the contributors may not be fully aware of them or choose not to describe them at this time. For these omissions we must apologize. The contributors and the Editor hope the reader will be kind enough to inform the publisher of his or her own special knowledge and experiences, so that future editions of this handbook can be increasingly useful.

2
PSYCHROMETRICS

The practice of humidity control assumes a familiarity with the properties and behavior of moist air. This is the science of psychrometrics. While it is not necessary to have a deep understanding of the thermodynamics of moist air, it is useful to understand the terms used in the trade, and understand why particular concepts are essential in designing systems.

Psychrometric terms and equations are described in detail in the Handbook of Fundamentals published by the American Society of Heating, Air Conditioning and Refrigerating Engineers (ASHRAE). When an engineer needs to understand all aspects of psychrometrics, the ASHRAE Handbook of Fundamentals provides an excellent reference.

This chapter is not concerned with complete precision for all temperatures and pressures of air and water mixtures. We assume the designer is working at sea level air pressures, and is concerned with air-water mixtures at temperatures between -20 and +100°F. The information in the chapter describes the basic terms in a simple way, and shows how charts and graphs can be used to understand the overall pattern of air-moisture dynamics.

Early in the twentieth century, a German engineer named Richard Mollier invented a graphic method of displaying the properties of various mixtures of air and water vapor. This device has different names in different counties — *the i-x diagram, Mollier diagram or psychrometric chart* — but the names all refer to similar technical graphic displays.

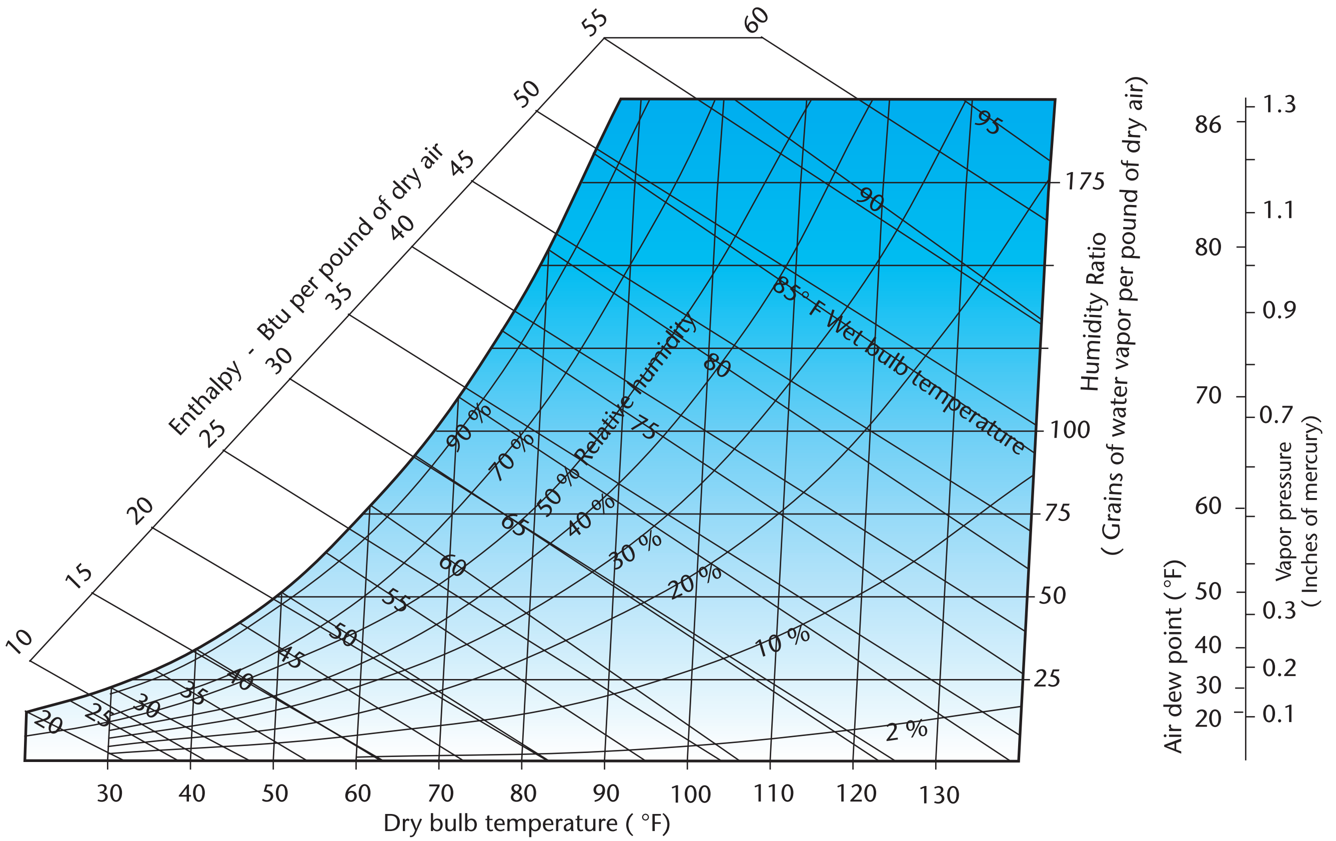
At first, the chart can be rather daunting, because it displays so much information in a small space. However, once the basic information elements are understood, the chart becomes an essential reference tool when designing temperature and humidity control systems.

This chapter describes each of the properties of moist air in turn, and then shows how these can be found quickly by using a psychrometric chart. The chart is useful both for the information it contains, and the relationships it shows between air at different conditions. It not only shows the "trees" in the psychrometric jungle, but shows the whole "forest" as well, allowing an engineer to gain a sense of how easy or difficult it might be to change the air from one condition to another.

FIGURE 2.1

The Mollier diagram, or psychrometric chart, provides a comprehensive overview of the thermodynamic properties of air-water mixtures. If any two properties of the air mixture are known, the chart allows an engineer to quickly determine all its other properties.

The relationships shown in the psychrometric chart change with total air pressure. When working with air at elevations above 2500 ft. — or with compressed air — the engineer must consult different charts for accurate data.



To help make the abstract properties of air more concrete, we will imagine a container filled with a pound of air. The air has been taken from a sea-level building kept at typical comfort conditions — 70 degrees Fahrenheit and 50% relative humidity. Knowing these two properties of the air, we can determine all others by using a psychrometric chart. But we will begin by defining what we know.

Dry Bulb Temperature *Degrees Fahrenheit*

FIGURE 2.2

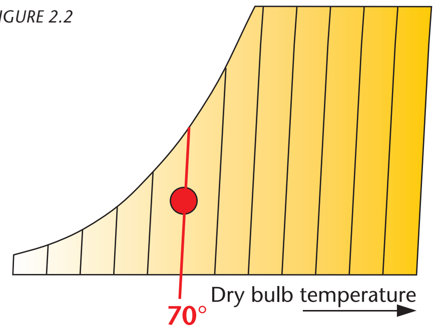
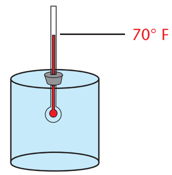


FIGURE 2.3



The air temperature is 70 degrees. When people refer to the temperature of the air, they are commonly referring to its dry bulb temperature — what can be read from a standard thermometer that has no water on its surface. This is also called the "sensible" temperature of the air — the heat which can be sensed by a dry thermometer. (The "wet-bulb" temperature is measured by a wet thermometer, as described later in this chapter.)

On a psychrometric chart, the dry bulb temperature of the air is displayed at the bottom, increasing from left to right.

Relative humidity *Percent of saturation*

FIGURE 2.4

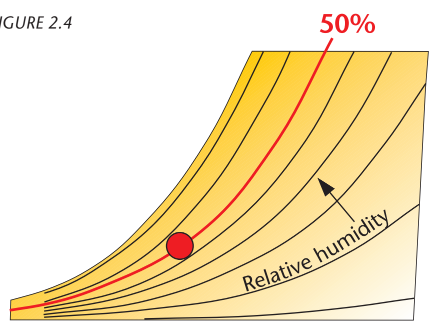
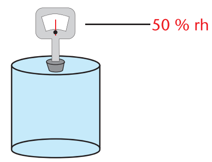


FIGURE 2.5



In our example, the air is at 50% relative humidity. Essentially, relative humidity expresses the moisture content of air as a percent of what it can hold when the air is saturated.[1] Like the name suggests, it is not a measure of the absolute amount of moisture in the air; it measures the moisture contained in the air relative to the maximum value at the dry bulb temperature of the air sample. Since that maximum increases with temperature, the term relative humidity has caused much confusion. When people refer to relative humidity, it is important to define the dry bulb temperature of the air they are referring to.

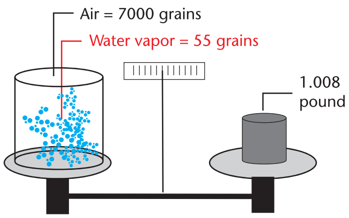
[1] This definition is accurate in concept, but strictly speaking, relative humidity is the ratio the actual water vapor partial pressure in moist air at the dew point pressure and temperature to the reference saturation water vapor partial pressure at the same temperature and barometric pressure. The difference between this definition and the one above is normally significant only outside of the human comfort range.

This is not to say that relative humidity is not useful. Quite the contrary. Most materials absorb moisture in proportion to the relative humidity of the surrounding air. Appropriately, many humidity control systems respond to sensors which measure relative rather than humidity ratio. However, in designing a system to control air moisture, it is important to define both the relative humidity and its concurrent dry bulb temperature range.

On the psychrometric chart, relative humidity is displayed as a series of curves, increasing from the bottom of the chart to the "saturation curve" which forms the left boundary. The saturation curve represents 100% relative humidity.

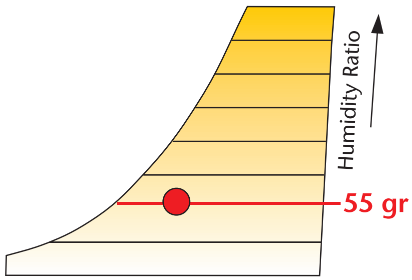
Humidity Ratio *Grains of water vapor per pound of air*

FIGURE 2.6



To define the amount of moisture in the air, we use its weight compared to the weight of dry air. This is like counting the water molecules and adding their weight together. The weight is measured in grains, and there are 7000 grains in a pound. In our example, when the air is at 70°F and 50% rh, its specific humidity is 55 grains of water per pound of dry air. In other words, in our sample, there is one pound of dry air and 55 grains water vapor, or 7055 grains of total weight.

FIGURE 2.7



The psychrometric chart allows us to determine the humidity ratio of our air sample. By starting at the 70°F, 50% point on the chart, we can read the humidity ratio by tracing a horizontal line to the chart's right edge, where the scale indicates the weight of the moisture in grains per pound of dry air.

Other psychrometric charts will display the humidity ratio as a small decimal fraction representing the mass of water divided by the mass of air. It amounts to the same thing. If you multiply the decimal fraction by 7000, you can convert the ratio to grains of water per pound of air, since there are 7000 grains in a pound. Multiply the ratio by 1000, and the value is expressed in grams per kilogram. In all cases, the vertical scale at the right edge of the chart indicates the amount of moisture in the air sample.

Vapor pressure *Inches of mercury*

Like all gas molecules, each water molecule exerts pressure on the surrounding environment. The amount of vapor pressure at a certain moisture content is the sum of the pressures of all the water molecules.

FIGURE 2.8

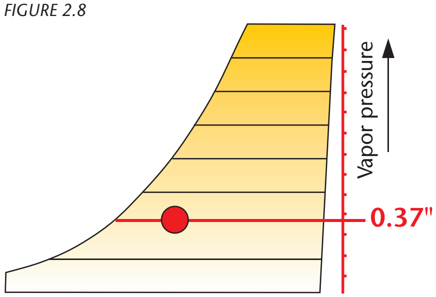
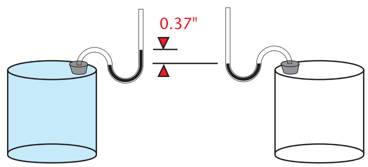


FIGURE 2.9



The unit of measure is inches of mercury — in other words, how high the water vapor can lift a column of mercury due only to its own partial pressure. In this example, the water vapor lifts a column of mercury 0.37 inches high.

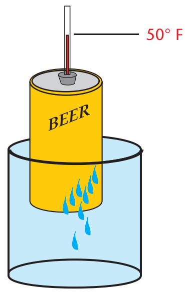
In conventional heating and cooling, engineers often measure air pressure in inches of water. Realizing that mercury weighs 13.6 times as much as water, it becomes apparent that a large amount of water vapor exerts a considerable force. The pressure difference is enough to peel paint off of wooden siding in the winter. There is more moisture inside the wood than outside, so the pressure difference forces the paint off the surface. As explained further in Chapter 3, desiccant dehumidifiers use differences in vapor pressure to attract water molecules out of the air and onto the desiccant surface.

The vapor pressure scale to the right of the chart increases linearly, just like humidity ratio.

Dew point temperature *Degrees fahrenheit*

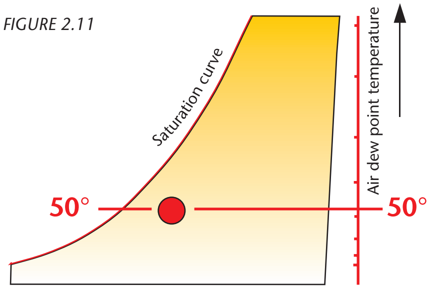
If moist air is cooled, it cannot hold the same amount of moisture. At some point, the moisture will condense out of the air onto any nearby surface. This point depends on the amount of moisture in the air, and is called the *dew point temperature.* The higher the amount of moisture in the air, the higher the dew point temperature.

FIGURE 2.10



In our example, the air has a moisture content of 55 grains. This amount of moisture will condense if the air temperature is dropped to 50°F. For instance, if a cold can of beer is taken from the refrigerator at a temperature of 50°F and placed in our air sample, the can surface will cool the air from 70° to 50°, and moisture in the air next to the can will begin to condense.

FIGURE 2.11



In other words, our air sample is "saturated" when cooled to 50°F — it has reached a condition of 100% relative humidity.

The left edge of the psychrometric chart is sometimes called the saturation curve. If you draw a line horizontally to the left from the 70°, 50% rh condition, it will intersect the edge — the saturation curve — at a temperature of 50°. This particular chart also repeats the dew point on a vertical bar to the right of the chart.

Cooling-based dehumidification systems remove moisture from air by cooling it below its dew point temperature. Cold objects in storage can be below the dew point temperature of the surrounding air, so moisture condenses on them just like the beer can in our example. The importance of these facts will become apparent in later chapters.

Enthalpy *Btu's per pound of air*

FIGURE 2.12

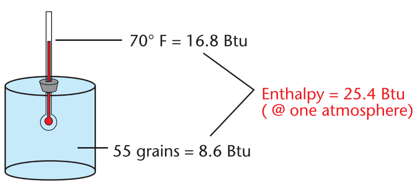
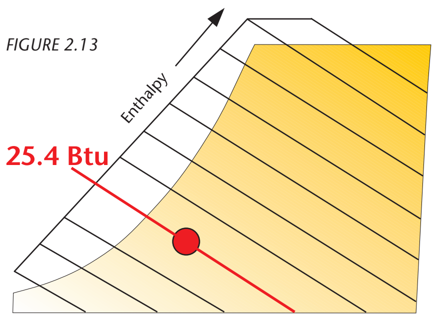


FIGURE 2.13



Enthalpy is a useful, but sometimes confusing concept — the measure of the total energy in the air. When air is hot, its enthalpy is high. Enthalpy is also high when air is moist. This is because it takes heat to evaporate moisture into the air — the more moisture in the air, the more heat was necessary to heat it and evaporate it. Every pound of water requires approximately 1061 Btu's for evaporation. (The actual amount of heat necessary reduces as the temperature rises. 1061 Btus is the value at 0°F). Therefore enthalpy — the total energy of the air — is a function of its sensible temperature, the absolute amount of moisture it contains and its total pressure.

In other references, enthalpy is defined as the sum of the *sensible and latent* heat in the air. The term "latent heat" is another way of expressing the amount of moisture in the air. Moisture represents the amount of heat required for its evaporation, but that heat is latent — it cannot be "sensed" by a dry bulb thermometer. On the psychrometric chart, the enthalpy scale is located to the left of the saturation curve. In our example, the enthalpy at 70°F and 50% rh is 25.4 Btu per pound of air.

The concept of enthalpy and the psychrometric chart allows the engineer to quickly and easily calculate the minimum energy to move from any condition to any other condition. This is very useful in designing systems to control air temperature and moisture. For example, If we needed to saturate the air in our container to a condition of 70° and 100% relative humidity, we would have to add moisture, and the energy required to evaporate the additional moisture would be at least 8.67 Btu's for every pound of air we saturate. This is the difference between the current enthalpy and the 34.07 Btu enthalpy of air at 70°, 100% rh.

Wet-bulb temperature *Degrees fahrenheit*

FIGURE 2.14

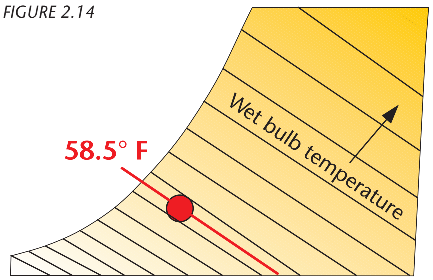
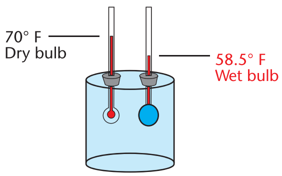


FIGURE 2.15



Where the dry-bulb temperature is measured by a dry thermometer, the wet-bulb temperature is taken by surrounding the sensor with a wet wick and measuring the reading as the water evaporates. As the water evaporates from the wick, it draws heat required for evaporation from the thermometer bulb, cooling the thermometer in proportion to the amount of evaporation. In our example, the wet bulb thermometer is cooled from 70°F to 58.5° by evaporation.

If the air was more saturated than 50%, it would not have been cooled so much. If the air was drier, the wet bulb temperature would be below 58.5°.

On the psychrometric chart, the wet bulb lines are very nearly parallel to the enthalpy lines. If we know both the wet bulb and dry bulb temperature of the air, its absolute moisture content can be found easily by using the chart. This is the basis of many simple moisture measurement devices like the "sling psychrometer".

The concept of wet bulb temperature is useful because it is a relatively simple and inexpensive way to measure air moisture. Also, the difference between the dry bulb and wet bulb temperatures provides a measure of the air's drying potential. If the dry bulb is high and the wet bulb low, the air can absorb a large amount of moisture, and it contains enough sensible heat to vaporize the water it can absorb. If both the dry and wet bulb temperatures are the same, the air is saturated and cannot absorb more moisture.

Additional references

There are subtle differences between the definitions we have used here and definitions which are completely precise for all temperatures and pressures. For instance, the relative humidity of the air is actually "the ratio of the mole fraction of water of the sample compared to the mole fraction of air saturated at the same temperature and pressure". In most situations, the difference between this value and the value arrived at by dividing the sample moisture in grains by the saturated moisture is insignificant. But in calculations for very low humidities, or for product drying at high temperatures and high moisture contents, the error is larger and may be important. When engineers need to know the precise definition of psychrometric terms, they should consult references that treat the subject in more depth than space allows here. Some references include:

Handbook of Fundamentals. 2001. American Society of Heating, Air Conditioning and Refrigerating Engineers (ASHRAE) 1791 Tullie Circle, N.E., Atlanta, GA 30329

Hyland, Richard W. and Wexler, Arnold. 1983. *Formulations for the thermodynamic properties of dry air from 173.15 K to 473.15 K and of saturated moist air from 173.15 K to 473.15 K, at pressures to 5 MPa.* ASHRAE Transactions 89(2A) : pp 520-535. American Society of Heating, Air conditioning and Refrigerating Engineers (ASHRAE) 1791 Tullie Circle, N.E., Atlanta, GA 30329

3

METHODS OF DEHUMIDIFICATION

There are three ways to remove moisture from air: by cooling it to condense the water vapor, by increasing its total pressure — which also causes condensation — or by passing the air over a desiccant, which pulls moisture from the air through differences in vapor pressures.

Information concerning pressure-based dehumidification can be found in references addressing compressed air. Since this handbook deals with ambient pressure humidity control, Chapter Three will discuss cooling and desiccant dehumidification techniques — how they work, how they differ from one another and where each can be applied with advantage.

Cooling-based Dehumidification

Most people are familiar with the principle of condensation. When air is chilled below its dew point temperature, moisture condenses on the nearest surface. The air has been dehumidified by the process of cooling and condensation. The amount of moisture removed depends on how cold the air can be chilled — the lower the temperature, the drier the air.

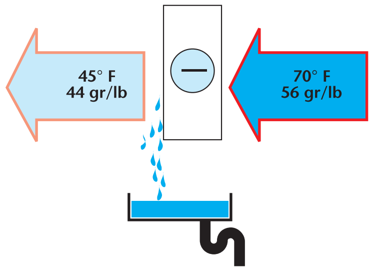


FIGURE 3.1

Cooling-based dehumidification

By chilling air below its dew point, moisture condenses and air is dehumidified.

This is the operating principle behind most commercial and residential air conditioning systems. A refrigeration system cools air, drains away some of its moisture as condensate and sends the cooler, drier air back to the space. The system basically pumps the heat from the dehumidified air to a different airstream in another location, using the refrigerant gas to carry the heat.

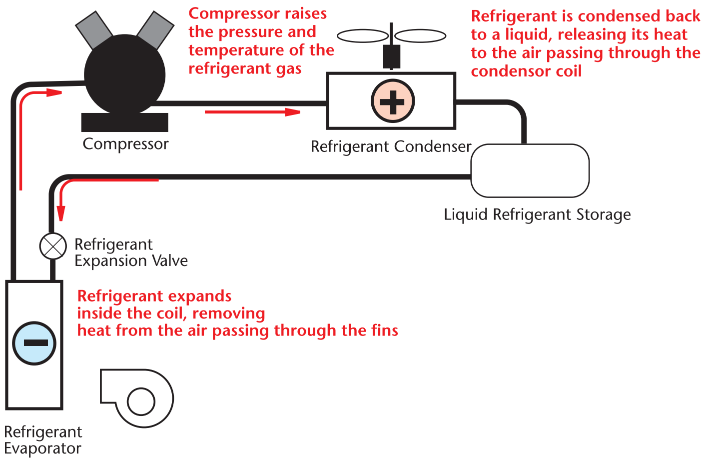


FIGURE 3.2

Mechanical cooling system

Refrigeration systems transfer heat from one airstream to another very efficiently — cooling one and heating the other. This is the basis for most cooling-based dehumidification systems.

Heat is removed from the dehumidified air by first transferring its thermal energy to an expanding gas — a refrigerant — which is inside the cooling coil that chilled the air. This coil is called the *evaporator*, because inside the coil, the refrigerant is evaporating and expanding from a liquid to a gas. For that gas to expand inside the coil, it needs heat, which it gets by cooling the air passing through the coil.

From the cooling coil, the refrigerant gas is sent to a compressor, where its pressure is increased substantially — 5 to 10 times greater than when it left the evaporator coil. The gas is therefore a much smaller volume, but compression has raised its temperature. For instance, the gas may have been at 60°F after it absorbed the heat from the air on the other side of the evaporator coil, but after compression, the refrigerant gas may be 200°F or higher. That heat — and the heat from the process of compression itself — must now be removed from the refrigerant. This is accomplished by running the gas through a second coil.

This coil — called the *condenser* — is located outside the conditioned space, in a place where the heat can be rejected to the air without causing problems. These units are often located outside a building or on a rooftop. The compressed, hot refrigerant condenses back to a liquid inside the coil, and its heat — which started out in the air being dehumidified — is transferred to the air on the other side of the condenser coil. The cooled refrigerant liquid can now return to the coil cooling the original airstream. As the liquid expands again back to a gas inside the evaporator coil, it gathers more heat from that airstream and the cycle repeats.

The process can be very efficient. The common measure of efficiency is the coefficient of performance, which is the energy removed from the dehumidified airstream divided by the energy invested to accomplish the transfer to the condenser airstream. This transfer energy consists of the compressor energy plus the fan energy that pushes air through the two coils. Many electrically-driven refrigeration systems enjoy coefficients of performance of 2.0 to 4.5, which is to say the system moves two to four and a half times as much thermal energy as it consumes in electrical energy — a very favorable ratio.

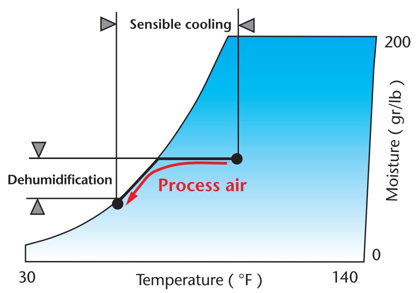


FIGURE 3.3

Dehumidified air path

Cooling systems first chill the air to its dew point — 100% relative humidity.
After that point, further chilling removes moisture. The more the air is cooled, the deeper it will be dried.

Dehumidification through air cooling can be illustrated on a psychrometric chart, using air at the same condition we used for the example in Chapter Two — 70°F and 50% relative humidity.

As air is cooled from 70°F to 51°, no moisture is removed. But when the air is at 51°, it is saturated — 100% relative humidity — and if it is cooled further, its moisture will have to condense out of the air. If we cool the air from 51° to 45°, we will remove 11 grains of moisture through condensation — the air has been dehumidified.

The actual hardware that accomplishes cooling dehumidification is exceptionally diverse. Literally thousands of different combinations of compressors, evaporators and condensers are in use throughout the world. But there are three basic equipment configurations of interest to designers of humidity control systems, which include:

- Direct expansion cooling
- Chilled liquid cooling
- Dehumidification + reheat

Direct expansion systems use the system configuration outlined in the previous example. The refrigerant gas expands directly into the air cooling coil, removing heat from the airstream. Residential air conditioners and commercial rooftop cooling packages are generally direct expansion — sometimes called "DX" — type units as shown in figure 3.2.

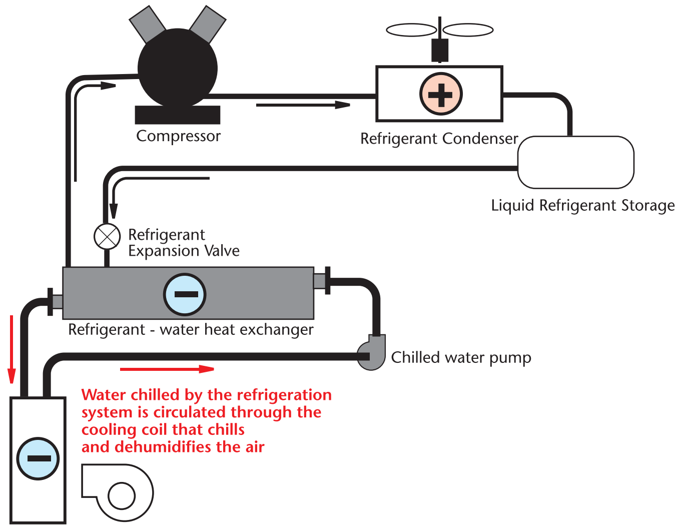


FIGURE 3.4

Chilled liquid cooling system

Evaporating refrigerant can cool liquid rather than air. The liquid is then used to cool air. The design can cool air close to 32°F without freezing the condensate, and has the advantage of equalizing the load on the compressor and condenser when many different air streams must be cooled by a single refrigeration system.

Chilled liquid systems use the refrigerant gas to cool a liquid, which is then circulated through a cooling coil to cool the air being dehumidified. Such machinery is often called chilled water, glycol chiller or brine chiller systems, according to the fluid cooled by the refrigerant gas. This is the same basic configuration that operates the water coolers that are so common in commercial and institutional buildings. Although there are hundreds of thousands of small chiller systems like water coolers, in air conditioning applications, these tend to be more complex and expensive than alternatives. As a result, chilled liquid systems are more often used in large installations where they can gain advantages of installed cost and operating efficiency over DX systems.

Dehumidification-reheat systems can use either direct expansion or chilled liquid for cooling the air, but following cooling, the air is reheated before it is returned to the space. Most residential dehumidifiers use this configuration. They are sold in appliance stores for use in basements and moist houses. Commercial and industrial versions of the dehumidification-reheat system are used in swimming pools, lumber kilns and locker rooms — high temperature, high moisture environments.

Dehumidification-reheat systems can use the fundamental operating characteristics of cooling systems to achieve great efficiency. If all other variables are constant, the mechanical cooling process is more efficient when:

- The condenser air temperature is low.
- The cooling coil air temperature is high.

A typical configuration of a dehumidification-reheat system places the refrigerant condenser coil immediately downstream of the cooling coil. This is ideal since the low air temperature after the cooling coil makes the refrigerant condenser very efficient. The reheat energy is essentially free, since it is heat rejected by the process of cooling. Expensive extra energy from outside the process is minimized.

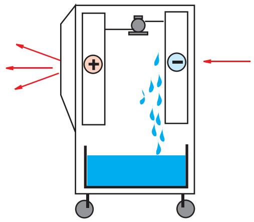


FIGURE 3.5

Dehumidification-reheat system

Most small, residential dehumidifiers use the cooling-reheat schematic to remove moisture from air. For high temperatures and moisture levels, the arrangement is especially efficient because it places the refrigerant condenser down stream of the evaporator. The reheat energy is essentially free, and the condenser is most efficient in the low temperature air that comes from the evaporator coil.

When the entering air is both warm and very moist, and the required leaving air dew point is also high, dehumidification-reheat systems are efficient and cost-effective methods of removing moisture from air. The designer is well-advised to use this method where possible. However, some limitations of the technology appear when the cooling process freezes moisture in the air rather than simply condensing it to a liquid.

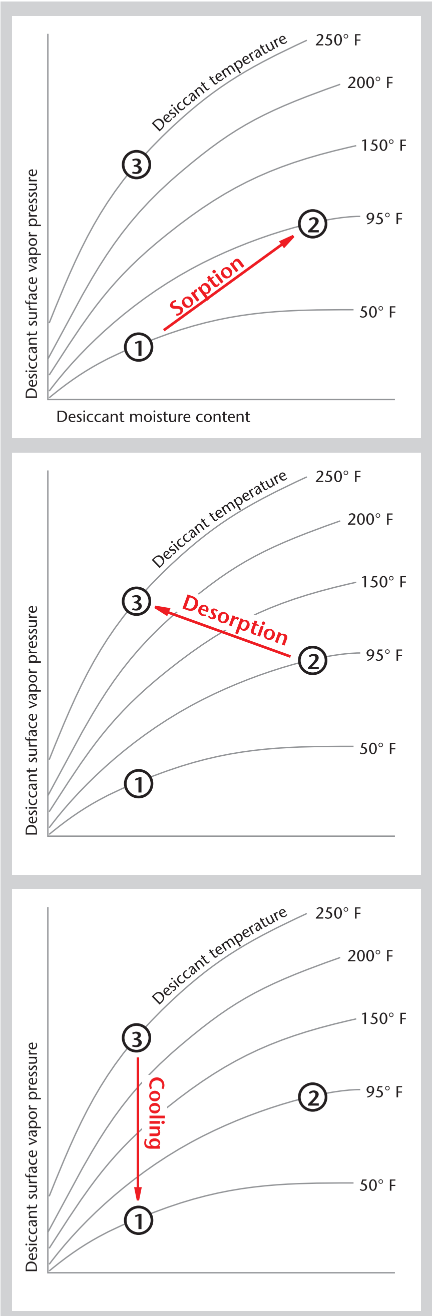
Frozen condensate causes two problems for a cooling system. First, the frost insulates the refrigerant from the air passing through the coil, which reduces heat transfer. Secondly, the frost physically clogs the coil, reducing the airflow. Eventually the frost blocks the airflow all together and dehumidification ceases. Systems that cool air below 32°F include defrost systems to melt the frozen condensate from the coil. Dehumidification and cooling stops while the coil defrosts.

Specially designed DX dehumidification systems cool air to levels between 43 and 45°F. Below that point, frost begins to form on parts of the coil, spreading slowly through the coil as the airflow becomes restricted. Some design techniques can also extend cooling below 43°F without frost, but the system can become very difficult to control at part-load conditions. The difficulty stems from the tendency of the system to continue cooling the air by a fixed amount. For instance, air entering at 75°F might be cooled to 55° — a 20° temperature differential. But if the system is not carefully controlled, part of the air entering at 50°F will be cooled to 30°, which freezes condensate in the coil.

Chilled liquid dehumidification systems are easier to control at low temperatures than DX cooling systems, because there is a smaller difference between the fluid entering and leaving the cooling coil. For example, the chiller may supply 32°F liquid to the air cooling coil, which may return to the chiller at 40°F. This can provide an average leaving air temperature of 35°. A direct expansion system, by contrast, might have to create a temperature of 20°F as the refrigerant expands into the coil in order to achieve that same 35°F average leaving air condition. This means part of the coil surface will be below 32°F, and frost will form at that point.

The specific dehumidification capacities of cooling systems are highly dependent on hardware specifics, and generalizations can be misleading. It is enough for the engineer to realize that while cooling systems dehumidify efficiently at high temperatures, special precautions are necessary when using cooling systems to dry air below a 40°F dew point.

FIGURE 3.6



Desiccant Dehumidifiers

Desiccant dehumidifiers are quite different from cooling-based dehumidifiers. Instead of cooling the air to condense its moisture, desiccants attract moisture from the air by creating an area of low vapor pressure at the surface of the desiccant. The pressure exerted by the water in the air is higher, so the water molecules move from the air to the desiccant and the air is dehumidified.

Actually, most solid materials can attract moisture. For instance, plastics like nylon can absorb up to 6% of their dry weight in water vapor. Gypsum building board can also store a great deal of water vapor, and the oxide layer on metals attracts and holds a small amount of water vapor under the right conditions. The difference between these materials and commercial desiccants is capacity. Desiccants designed for water vapor collection attract and hold from 10 to over 10,000 percent of their dry weight in water vapor, where other materials have much lower moisture capacity.

The essential characteristic of desiccants is their low surface vapor pressure. If the desiccant is cool and dry, its surface vapor pressure is low, and it can attract moisture from the air, which has a high vapor pressure when it is moist. After the desiccant becomes wet and hot, its surface vapor pressure is high, and it will give off water vapor to the surrounding air. Vapor moves from the air to the desiccant and back again depending on vapor pressure differences.

Desiccant dehumidifiers make use of changing vapor pressures to dry air continuously in a repeating cycle described by the simplified equilibrium diagram at left. The desiccant begins the cycle at point one. Its surface vapor pressure is low because it is dry and cool. As the desiccant picks up moisture from the surrounding air, the desiccant surface changes to the condition described by point two. Its vapor pressure is now equal to that of the surrounding air because the desiccant is moist and warm. At point two, the desiccant cannot collect more moisture because there is no pressure difference between the surface and the vapor in the air.

Then the desiccant is taken out of the moist air, heated, and placed into a different airstream. The desiccant surface vapor pressure is now very high — higher than the surrounding air — so moisture moves off the surface to the air to equalize the pressure differential. At point three, the desiccant is dry, but since it is hot, its vapor pressure is still too high to collect moisture from the air. To restore its low vapor pressure, the desiccant is cooled — returning it to point one in the diagram and completing the cycle so it can collect moisture once again.

Thermal energy drives the cycle. The desiccant is heated to drive moisture off its surface (point two to point three). Then the desiccant is cooled to restore low vapor pressure (point three to point one). The efficiency of the process improves when the desiccant has a high moisture capacity and a low mass. The ideal desiccant dehumidifier 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 airstream. The heavier the desiccant assembly compared to its capacity, the more energy it will take to change its temperature — which accomplishes dehumidification.

Desiccants can be either solids or liquids — both can collect moisture. For example, the small packets inside camera cases and consumer electronics boxes often contain silica gel, a solid desiccant. Also, triethylene glycol — a liquid similar to auto antifreeze — is a powerful desiccant which can absorb moisture. Liquid and solid desiccants both behave the same way — their surface vapor pressure is a function of their temperature and moisture content.

One subtle distinction between desiccants is their reaction to moisture. Some simply collect it like a sponge collects water — the water is held on the surface of the material and in the narrow passages through the sponge. These desiccants are called ***ad***sorbents, and are mostly solid materials. Silica gel is an example of a solid adsorbent. Other desiccants undergo a chemical or physical change as they collect moisture. These are called ***ab***sorbents, and are usually liquids, or solids which become liquid as they absorb moisture. Lithium chloride is a hygroscopic salt which collects water vapor by absorption, sodium chloride — common table salt — is another.

Whether the desiccant functions by absorption or adsorption is not usually important to a system designer, but the distinction exists and engineers should be aware of the difference between the two terms.

So far we have discussed how the desiccant functions. Now we will examine what happens to the air being dehumidified. When moisture is removed from air, the reaction liberates heat. This is simply the reverse of evaporation, when heat is consumed by the reaction. In a cooling-based dehumidification system, the heating effect of dehumidification is less apparent because the heat is removed immediately by the cooling coil. In a desiccant dehumidification system, the heat is transferred

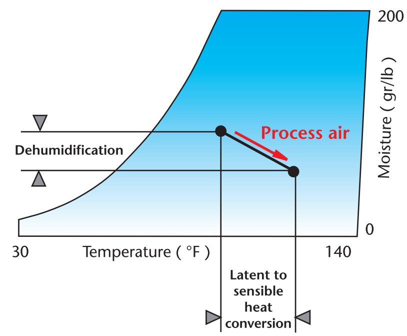


FIGURE 3.7

Dehumidified air path

As moisture is removed from the air, its enthalpy stays constant, so its sensible temperature rises. In fact, the enthalpy of the process air — air being dehumidified — actually increases slightly. This is because in many dehumidifiers, a small amount of residual heat from desiccant reactivation can be carried into the dry air stream.

to the air and to the desiccant, so the process air generally leaves the dehumidifier warmer than when it entered the desiccant unit.

The temperature rise is directly proportional to the amount of moisture removed from the air — the drier the air leaves the dehumidifier, the warmer it will be.

Looking at the process on a psychrometric chart, it is apparent how desiccant dehumidification differs from cooling-based dehumidification. Using our previous example of air entering the dehumidifier at 70°F and 50% relative humidity, the dry bulb temperature rises as the moisture falls, so that the total energy (enthalpy) of the air stays the same. In fact, the total energy actually increases slightly because of waste heat transferred to the air from the regeneration process. In many applications — notably product drying and unheated storage — this temperature rise of the dry air is desirable. In other cases additional sensible heat is not an advantage, so the dry air is cooled before being delivered to the point of use.

There are five typical equipment configurations for desiccant dehumidifiers:

- Liquid spray-tower
- Solid packed tower
- Rotating horizontal bed
- Multiple vertical bed
- Rotating Honeycombe®

Each configuration has advantages and disadvantages, but all types of desiccant dehumidifiers have been widely applied.

Liquid spray-tower

Spray-tower dehumidifiers function much like an air washer, except instead of water, the units spray liquid desiccant into the air being dried, which is called the *process air*. The desiccant absorbs moisture from the air and falls to a sump. The liquid is sprayed back into the air, and continues to absorb moisture until a level control indicates it should be dried out and re-concentrated. Then part of the solution is drained off and circulated through a heater. The warm desiccant is sprayed into a second airstream, called the *reactivation air*. Moisture leaves the desiccant and moves to the air.

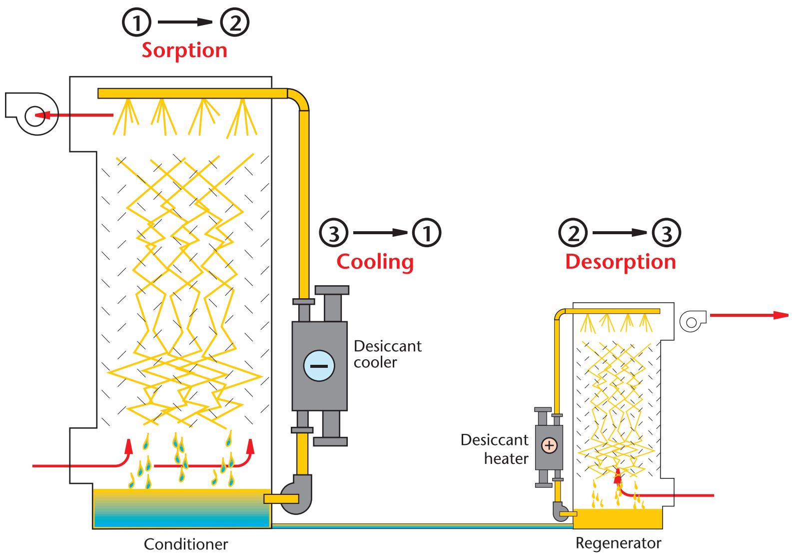
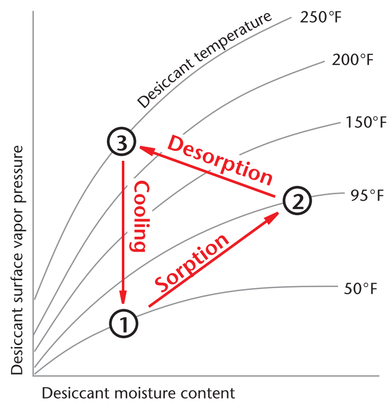


FIGURE 3.8

Liquid spray tower dehumidifier
These units are like air washers, except they spray liquid desiccant into the process air instead of simply water. The heat and moisture from the dehumidification process is transferred to the desiccant. Heat is rejected through an external cooling system and moisture is rejected in the desiccant regenerator, which re-concentrates the diluted desiccant solution.

The numbers on the diagram show how the hardware uses desiccant equilibrium characteristics to remove water first from the process air, and then from the desiccant. In the conditioner, the desiccant is absorbing water, becoming warmer and rising in vapor pressure—moving from point 1 to point 2 on the equilibrium chart. The desiccant in the sump is at point 2 in the chart — the desiccant solution has absorbed a great deal of water, and its surface vapor pressure is too high to attract more vapor. As the diluted desiccant passes through the heater, its vapor pressure rises, and when it is sprayed into the reactivation air the high pressure forces the water out of the desiccant and into the air. This corresponds to moving between point 2 and point 3 on the equilibrium chart.

FIGURE 3.9



As the desiccant returns from the regenerator to the sump, it is dry — concentrated — but still has a high vapor pressure because it is warm. To cool the desiccant, part of the liquid is pulled out of the sump and circulated through a heat exchanger connected to a chilling system or cooling tower. The desiccant then moves from point 3 to point 1 on the equilibrium chart. Its vapor pressure is low because it is both dry and cool, so it can be circulated back through the conditioner to absorb more moisture.

Liquid spray tower dehumidifiers have some uniquely favorable characteristics. Dehumidification is thermodynamically quite elegant, because the desiccant is only heated or cooled to the minimum necessary to accomplish the required dehumidification. Also, when the process requires a constant humidity and the inlet air is dry, water can be added to the desiccant solution so the conditioner will act as a humidifier instead of a dehumidifier. Extra liquid can also be regenerated and sent to a holding tank, providing energy storage in less than 20% of the floor space of an equivalent ice-storage system. Further, since the liquid desiccant contacts the air, particles are removed as well as water vapor.

Liquid spray dehumidifiers are frequently arranged in large, central systems rather than small, free-standing units for small spaces. This is partly because they tend to be somewhat more complex than solid desiccant units, but also because large systems can be designed with several conditioner units connected to a single regenerator. This configuration is similar to a mechanical refrigeration system with multiple evaporators connected to a single condenser. For large buildings with several dehumidification systems, this can have advantages of first cost, at the expense of complexity of controls.

Potential disadvantages of liquid systems include response time, maintenance, and first cost for smaller units. Because the desiccant solution may be distributed throughout a long piping system and large reserve sump, the system can take time to respond to fast changing internal moisture loads or different necessary outlet conditions — such as occur in environmental simulation chambers. But slow response on the outlet conditions also means slow response to inlet changes — which can be an advantage. A large mass of recirculating desiccant protects an internal process from rapid changes in weather moisture.

Maintenance varies, but some liquid desiccants are corrosive, and therefore need more attention. Also, at low humidity levels, some liquid desiccants can dry out rapidly, which means liquid levels must be carefully watched to avoid desiccant solidification.

Solid packed tower

In the packed tower dehumidifier, solid desiccants like silica gel or molecular sieve is loaded into a vertical tower. Process air flows through the tower, giving up its moisture to the dry desiccant. After the desiccant has become saturated with moisture, the process air is diverted to a second drying tower, and the first tower is heated and purged of its moisture with a small reactivation airstream.

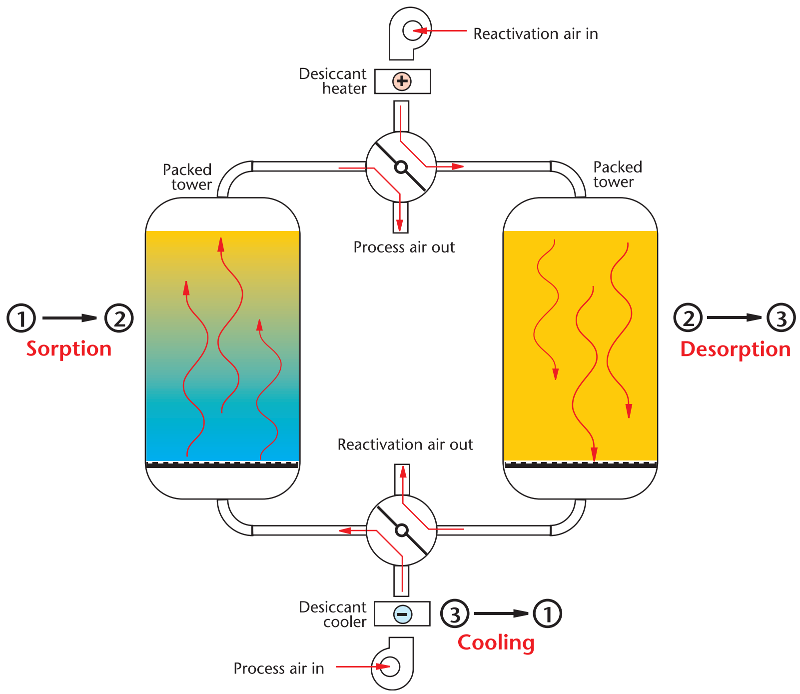


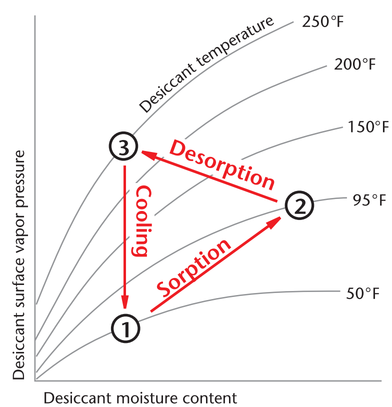
FIGURE 3.10

Packed tower dehumidifier

Air flows through large containers of granulated solid desiccant. The desiccant is dried by a different hot airstream that purges the container after the desiccant has been saturated. The system is used frequently for compressed air, pressurized process gases and sometimes even liquids that need dehumidification. It is less common in ambient-pressure applications.

The thermal energy that drives the desiccant cycle in a solid desiccant tower is added to the process by heating and cooling the reactivation and process airstreams. In other words, when the saturated desiccant must be heated to raise its surface vapor pressure (point 2 to point 3 on the equilibrium diagram), the heat is carried to the desiccant by hot reactivation air. Likewise, when the hot dry desiccant must be cooled to lower its vapor pressure (point 3 to point 1), the cool process air removes the heat from the bed.

FIGURE 3.11



Since drying and reactivation take place in separate, sealed compartments, the packed tower dehumidifier is frequently used to dry pressurized process gases. In fact, the same configuration is used to dry liquid chemicals as well as gases. When large amounts of desiccant are loaded into the towers, the process can achieve very low dew points — in many cases below -40°F. Desiccant dehumidifiers for compressed air are frequently the packed-tower type.

While the configuration allows very low dew points, the packed tower design can also result in a changing outlet condition. When the desiccant is first exposed to the process airstream, it can dry the air deeply. Later, as its moisture capacity fills up, the air is not dried quite so much. If a changing outlet condition would cause problems in a process, controls could be provided to ensure the towers are changed before the process air condition becomes too wet.

As process airflow requirements get larger, packed tower dehumidifiers get very large because air velocities are generally kept quite low. Low air velocities are necessary for two reasons. High velocities would lead to uneven air distribution through the bed because moist air would "tunnel" through the desiccant. Also, the air velocity must remain low to avoid lifting the desiccant, which would then impact against other particles and the walls of the dehumidifier vessel. The impact would fracture the desiccant, which would blow out of the unit in the form of a fine dust.

Engineers will notice that these types of units are frequently used in very small, low-dew point air streams and in process gas drying applications. The configuration offers compensating advantages in those situations that offset size and energy consumption disadvantages that appear in large-airflow, higher-dew point, atmospheric-pressure applications.

Rotating horizontal bed

In this device, dry, granular desiccant is held in a series of shallow, perforated trays that rotate continuously between the process and reactivation airstreams. As the trays rotate through the process air, the desiccant adsorbs moisture. Then the trays rotate into the reactivation airstream, which heats the desiccant, raising its vapor pressure and releasing the moisture into the air.

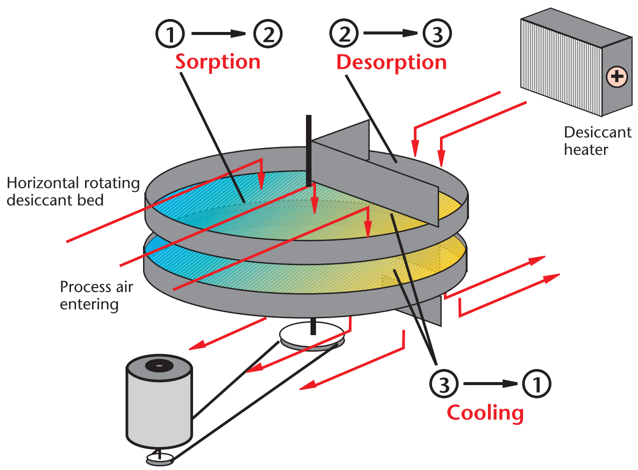
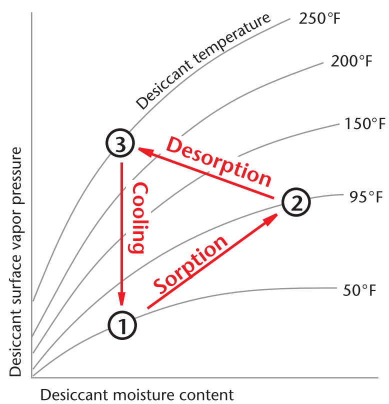


FIGURE 3.12

Rotating horizontal beds
Trays containing dry desiccant are slowly rotated between process and reactivation airstreams. Although care must be taken to avoid leakage between moist and dry airstreams, the design is inexpensive to produce.

Like in the packed tower, the process and reactivation air heats and cools the desiccant to drive the adsorption-desorption cycle. In the process side, the desiccant starts out dry — having just left the reactivation side. But the desiccant is still warm from the reactivation process. The desiccant is cooled by the process air during the first few degrees of rotation through the process side. This corresponds to the change from point 3 to point 1 on the equilibrium diagram. Then the desiccant dries the rest of the process air and picks up moisture — point 1 to point 2 on the diagram. As the trays rotate into the hot reactivation air, the desiccant is heated and releases moisture — point 2 to point 3 on the equilibrium diagram.

FIGURE 3.13



The design is modular. To increase capacity, the manufacturer can either increase the diameter of the rotating trays so they will hold more desiccant, or increase the number of beds stacked on top of one another. If the desiccant is evenly loaded through the trays, the rotating horizontal bed provides a fairly constant outlet moisture level, and a high airflow capacity can be achieved in less floor space than with a dual-tower unit. On the other hand, since the trays can never be filled absolutely to the top of the bed — the desiccant settles slightly in use — air leaks from the moist reactivation side to the dry process side within the tray just above the desiccant.

To avoid this leakage, rotating bed designs generally arrange the process and reactivation airflow in a parallel rather than a counterflow configuration. This keeps the pressures more equal between the process and reactivation sides of the unit, which reduces leakage and improves performance. The technique still has limitations, and such units are more sensitive to the moisture condition of the entering reactivation air than some other designs. Also, the parallel arrangement of process and reactivation airflows is not as energy efficient as a counterflow arrangement. As a result, reactivation energy consumption can be very high for these units compared to other designs.

Against these limitations, the rotating horizontal bed design offers a low first cost. The design is simple, expandable and easy to produce. Although the desiccant can fracture and carry over into the airstream, it can be replaced by disassembling the beds to refill them. In situations where energy cost is not high, or where it does not represent a great deal of money in absolute terms — such as in small dehumidifiers — the low first cost of these units may offset their high operating costs.

Multiple vertical bed

In recent years manufacturers have combined the better features of packed tower and rotating horizontal bed designs in an arrangement that is well-suited to atmospheric pressure dehumidification applications, yet can achieve low dew points. The single or double tower is replaced by a circular carousel with eight or more towers that rotate by means of a ratcheting drive system between the process and reactivation air streams.

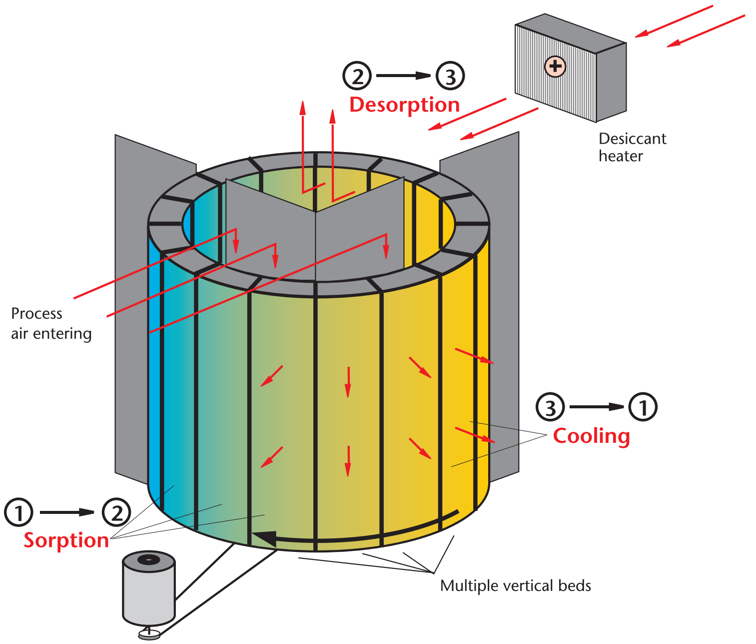


FIGURE 3.14

Multiple vertical beds

Arranging granular desiccant beds vertically rather than in flat trays combines the advantages of packed tower and rotating tray dehumidifiers. While the design includes more complex parts, the increased first cost is offset by a lower operating cost than either packed tower or rotating horizontal bed type units.

Like the packed tower, this design can achieve low dew points because leakage between process and reactivation air circuits is minimized. Also because the beds are separate and sealed from one another, the pressure difference between process and reactivation is not so critical, so airstreams can be arranged in the more efficient counterflow pattern for better heat and mass transfer. Like the rotating bed, the ratcheting, semi-continuous reactivation of the desiccant provides a relatively constant outlet air moisture condition on the process side, reducing the “sawtooth” effect that can occur in packed tower units.

These benefits are achieved at the expense of increased mechanical complexity. So compared to rotating horizontal bed units, the vertical bed dehumidifiers tend to be more expensive, and can require more maintenance. Generally, however, these are minor limitations compared to the large savings in energy and performance improvements at low dew points.

Rotating Honeycombe®

Another dehumidifier design uses a rotating Honeycombe® wheel to present the desiccant to the process and reactivation airstreams. This is sometimes called a DEW (**DE**siccant **W**heel) dehumidifier. The finely divided desiccant is impregnated into the semi-ceramic structure, which in appearance resembles corrugated cardboard that has been rolled up into the shape of a wheel. The wheel rotates slowly between the process and reactivation airstreams.

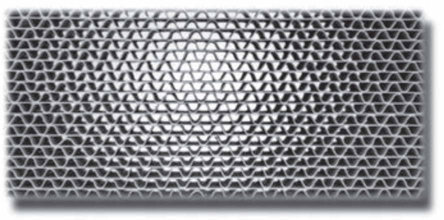


FIGURE 3.15

Rotating Honeycombe®
Desiccant is loaded into a lightweight, open structure. The design combines high surface area with low total desiccant mass, making these units especially efficient. The small number of parts reduces maintenance to a minimum.

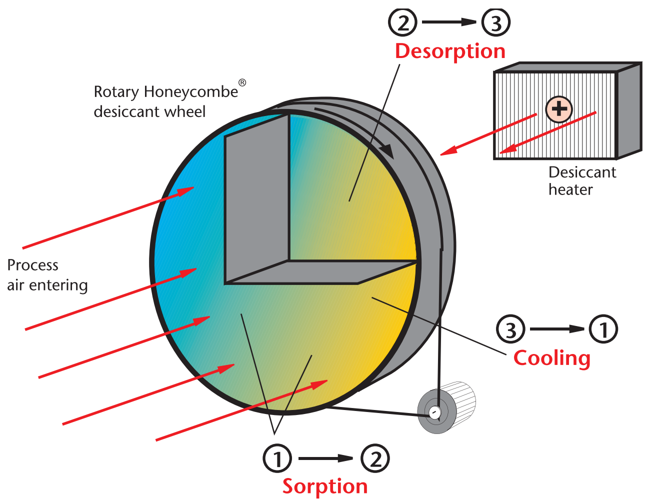
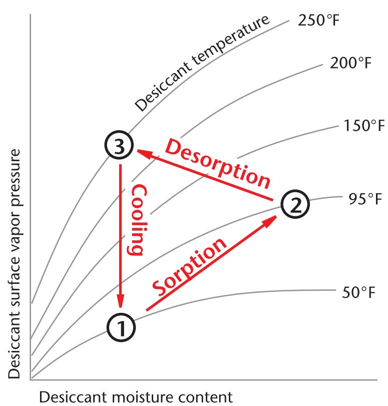


FIGURE 3.16



The process air flows through the flutes formed by the corrugations, and the desiccant in the structure absorbs the moisture from the air. As the desiccant picks up moisture it becomes saturated and its surface vapor pressure rises, which corresponds to the change between point 1 and point 2 on the equilibrium diagram. Then as the wheel rotates into the reactivation airstream, the desiccant is heated by the hot reactivation air, and the surface vapor pressure rises, allowing the desiccant to release its moisture into the reactivation air. This is the change from point 2 to point 3 on the equilibrium diagram.

Following reactivation, the hot desiccant rotates back into the process air, where a small portion of the process air cools the desiccant so it can collect more moisture from the balance of the process airstream. This is the cooling described by the change between points 3 and 1 on the equilibrium diagram.

The rotating Honeycombe® design has several advantages. The structure is very lightweight and porous. Different kinds of desiccants — both solid and liquid — can be loaded into the structure, which allows a wheel to be tailored for specific applications. Since the flutes of the structure are like individual, desiccant-lined air ducts, the desiccant surface area presented to the air is maximized even while the airflow remains smooth, reducing air pressure resistance compared to packed beds. Low dew points and high capacity — normally two mutually exclusive goals — can be achieved by combining different desiccants in the same wheel. And since the total rotating mass is low compared to its moisture removal capacity, the design is quite energy-efficient. The design is also quite simple, reliable and easy to maintain.

One design concern with Honeycombe® dehumidifiers is the cost of the rotating wheel. The structure is energy efficient, but costs more to produce compared to granules of dry desiccant. Care should be taken to insure the wheel is not damaged. The first cost is apparently balanced by operational advantages, since the design is the most widely installed of all desiccant dehumidifier configurations in ambient pressure applications.

Comparing desiccant dehumidifiers

All desiccant dehumidifiers can be made to work in virtually any application suited for desiccant dehumidifiers. The limitations of each configuration can be overcome and benefits of each design optimized by careful application engineering. No firm statement can be made about the limits of performance or amount of energy consumed or mechanical reliability of different dehumidifier types outside of a particular set of installation circumstances. *Application engineering rather than the dehumidifier type makes a dehumidification installation reliable, efficient and low in first cost.*

There are, however, some fundamental questions for the engineer to ask of potential system suppliers. These include:

• ***Installed cost***

The cost of the dehumidifier itself can be a small fraction of the cost of the installation. Since different configurations vary in their need for additional equipment, utilities and plant support — like chilled water, floor space and weather protection — the cost of the dehumidifier itself is less important than the cost of the complete installation.

• ***Operating cost***

The installed cost of mechanical systems is often small compared to their cost of operation and maintenance. The chief cost of running a dehumidification system is heat for reactivation and cooling for the desiccant and process air. When designs take advantage of low-cost energy sources for these utilities, they can often offset installed cost differences in a matter of months, yielding enormous financial benefits over the typical 15 to 30 year life of this equipment.

• ***Demonstrated operational reliability***

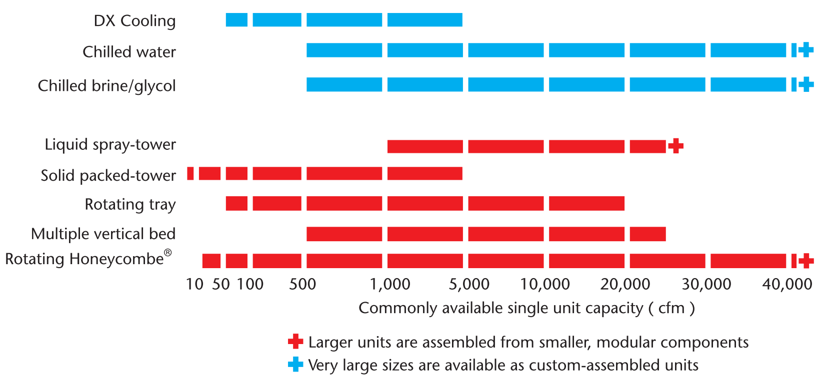
Each dehumidifier configuration behaves differently in different applications. If the installer or supplier can demonstrate an understanding of equipment behavior in the application in question, potential difficulties are reduced for the end user. While every installation is different, the engineer does well to limit the number of "unknowns" for his project.

• ***Design assumptions***

Different engineers and manufacturers necessarily make different assumptions concerning a given application. The selection of equipment configuration and size is completely dependent upon these assumptions. Often reasons for widely varying selections are the result of incomplete or erroneous data available to the system designer.

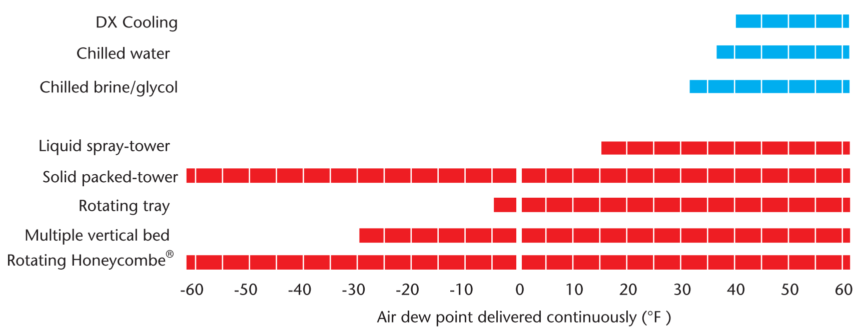
Putting the different types of dehumidifiers into perspective is very difficult, because specific circumstances make it impossible to state universally accurate comparisons. With that in mind, the graphics which follow are an attempt to show general relationships between dehumidifier types.

FIGURE 3.17



Among cooling-type dehumidifiers, the DX, pre-packaged units are generally more available in the smaller sizes, while chilled liquid types are more common when very large airflows must be dehumidified. Among desiccant types, the liquid spray type is most commonly used in larger sizes, and packed tower in smaller sizes. The more widely-applied Honeycombe® units are available in most all sizes. For larger airflows than shown here, most manufacturers build up the system using smaller units as modular components. Dehumidification systems processing over 250,000 cfm are quite common.

FIGURE 3.18



As the graphic above shows, cooling dehumidifiers are most often used for dehumidifying at higher moisture levels, and desiccant units used for lower level control. Figures 3.19 and 3.20 show some very general relationships concerning costs, as compared to the widely-applied Honeycombe® desiccant unit. However, in any specific circumstance, cost relationships may well be much different than the typical patterns shown here. More ink on the graphic means more money for the system compared to others.

FIGURE 3.19

Installed system cost as compared to a Honeycombe® dehumidifier

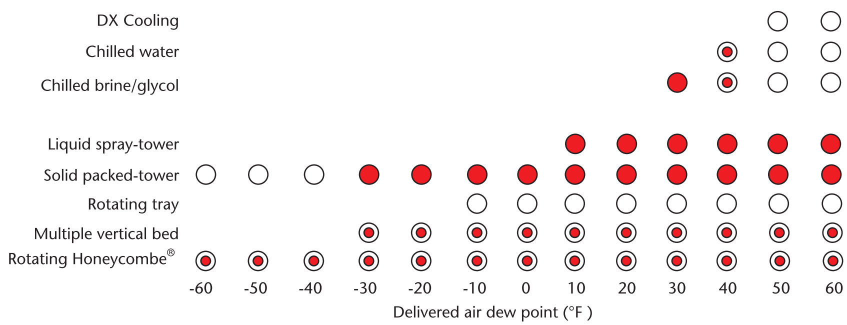
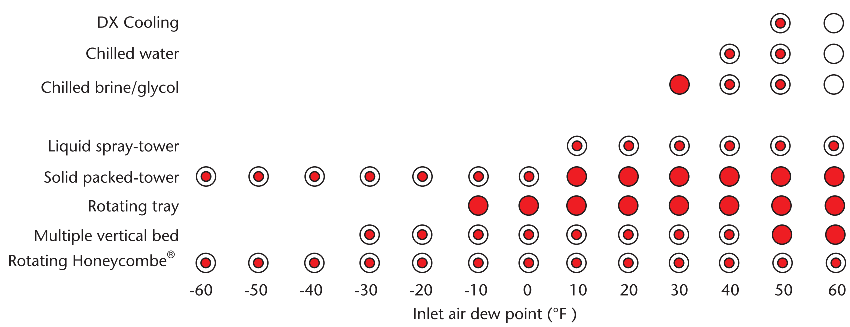


FIGURE 3.20

Operating cost to remove water vapor as compared to a Honeycombe® dehumidifier



Choosing between desiccant and cooling dehumidifiers

In many situations, both desiccants and cooling-based dehumidifiers can remove moisture from air, so the question arises — which type to use? Like choosing between different types of desiccant dehumidifiers, there are no simple answers to this question, but there are some general guidelines that have emerged in the industry:

• *Cooling and desiccant-based dehumidification systems are most economical when used together.* The technologies complement each other, each strength of desiccants covers a weakness of cooling systems and vice-versa.

• *The difference in the cost of electrical power and thermal energy will determine the optimum mix of desiccant to cooling-based dehumidification in a given situation.* If thermal energy is cheap and power costs high, the economics will favor using desiccants to remove the bulk of the moisture. If power is inexpensive and thermal energy for reactivation costly, the operating economics will favor using more cooling-based dehumidification in the project.

• *Cooling-based dehumidification systems are more economical than desiccants at high air temperatures and moisture levels.* They are very seldom used to dry air below a 40°F dew point because condensate freezes on the coil, reducing moisture removal capacity.

• *Desiccants may have useful advantages when treating ventilation air for building HVAC systems which use ice storage.* Since these systems deliver air at moderately low dew points (40 to 45°F), dehumidifying the fresh air with the desiccant system decreases the installed cost of the cooling system, and eliminates deep coils with high air and liquid-side pressure drops. This saves considerable fan and pump energy.

• *Desiccants are especially efficient when drying air to create low relative humidities, and cooling-based dehumidification is very efficient when drying air to saturated air conditions.* If the air should be drier than when it entered the machine, but still close to saturation at a lower temperature, cooling-based dehumidification would be a good choice. But if the desired end result is air at a condition far from saturation — low relative humidities — a desiccant unit would be ideal.

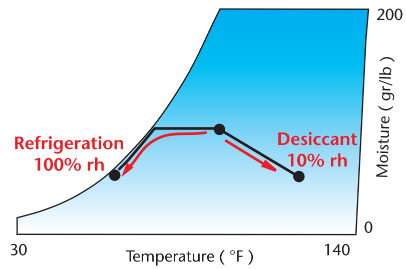


FIGURE 3.21

Cooling vs. desiccant

Air exits a cooling coil close to saturation — 100% rh. A desiccant unit produces air which is very dry in terms of relative humidity. Each has advantages depending on how close to saturation the air should be delivered from the system.

The two technologies work most efficiently when used together, so the advantages of each compensate for the limitations of the other.

4

APPLICATIONS

Corrosion Protection

- Military Storage
- Electronics Protection
- Power Plant Layup
- Lithium Battery Production

Condensation Protection

- Ice Rinks
- Water Treatment Plants
- Surface Preparation & Coating
- Injection Molding

Mold/Fungus Prevention

- Archival Storage
- Seed Storage
- Cargo Protection
- Breweries

Moisture Regain Prevention

- Candy Packaging
- Clean Rooms
- Safety Glass Laminating
- Composite Manufacturing

Product Drying

- Investment Castings
- Plastic Resin Drying
- Candy Coating
- Fish Drying

Dry Cooling

- Supermarkets
- Hotels & Motels
- Sick Buildings
- Advanced HVAC Systems

Applications for desiccant dehumidifiers are exceptionally diverse. Such equipment can have a very positive impact:

• Plastic soda bottles can be produced twice as fast when dry air is supplied to the mold, which means one plant can do the work of two, saving millions of dollars in new plant construction.

• Desiccant dehumidifiers have allowed Leonardo Da Vinci's famous fresco, *The Last Supper,* to survive urban pollution and floods, preserving the masterpiece for future generations.

• Suspension bridge cables corrode where they enter the concrete anchor blocks. Without dehumidifiers, the main cables would have to be replaced, costing millions of dollars and stopping traffic on some of the world's busiest river crossings.

• Without dehumidifiers, it would be impractical to manufacture the lithium batteries that power cardiac pacemakers, which keep hundreds of thousands of people alive and productive.

In this chapter we will discuss the six principal categories of dehumidification applications, with the purpose of stimulating the thoughts of creative engineers, and reassuring those who might believe they are "the first to try" a given application. Removing excess humidity from the air can have very interesting and profitable consequences.

Corrosion Prevention

All materials corrode, which is to say every substance eventually changes from one form to another through chemical reactions. Many of these reactions, especially those which depend on oxygen, are catalyzed and accelerated by moisture.

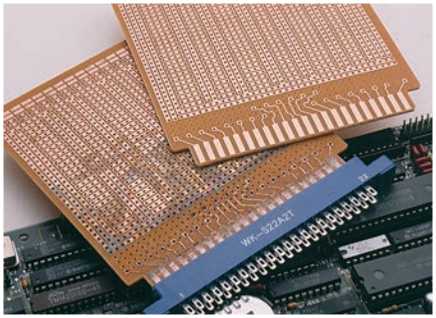
Ferrous metals like iron and steel are well known for their corrosion in the presence of moisture. Less well known is the fact that glass corrodes and cracks at a rate which varies according to the moisture on its surface. Pure crystals like sodium iodide and lithium fluoride also corrode, forming oxides and hydroxides in proportion to the moisture in the air. In the past, tens of thousands of desiccant dehumidifiers have been used to surround machinery and equipment with dry air, preserving ferrous metal parts from heavy rust.

In the present, dehumidifiers are working to protect materials from more subtle and expensive forms of corrosion. Modern society depends more and more on light equipment like computers, telecommunications gear, lightweight composite materials and high-energy batteries. While these are less subject to gross rusting, they are very sensitive to microscopic-level corrosion. These circuits simply do not have much material to begin with, so small amounts of corrosion create disproportionately large problems. Desiccant systems save owners literally hundreds of millions of dollars each year by preventing both gross and microscopic corrosion.

Typical Applications
- Marine drilling rig layup
- Industrial plant layup
- Galvanized steel storage
- Protecting box beams in bridges
- Ball bearing storage
- SCR motor control centers
- Generator rotor storage
- Sodium iodide crystal finishing
- Oil tanker layup
- Automotive stampings protection
- Military aircraft avionics protection
- Civil aircraft corrosion protection
- Calcium metal fabrication
- Metal hydride machining
- Computer storage
- Chemical plant pipe galleries
- Yacht protection
- Canal lock mechanism protection
- Pumping station
- Boiler protection
- Military tanks
- Helicopter storage
- Nuclear waste storage
- Precision tooling storage
- Ammunition storage
- Tire creel rooms & wire storage
- Razor blade manufacturing
- Tritium containment

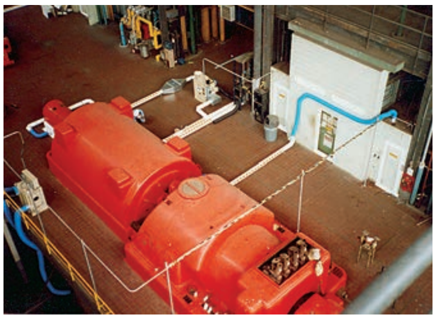
Military storage

In the 1950's after the Korean War, the U.S. military used dry air storage for long-term protection of inactive ships, machinery and weapons. The technique saved millions of dollars in preservation costs. In the 1970's, European military groups, notably those in Sweden and Denmark, pioneered the use of desiccant equipment for protection of active duty military material. Now tens of thousands of dehumidifiers protect expensive military equipment in all parts of the world, cutting maintenance costs drastically, and increasing the combat readiness of aircraft, tanks, ships and supplies.



Electronics protection

Computers and other electronic equipment use small voltages and low currents to perform their functions. When minute layers of corrosion build up on circuit surfaces, they increase electrical resistance and decrease capacitance, which can seriously effect calibration and performance. Also, when electrical equipment is rapidly cooled and heated — such as when cold aircraft descend into warm, humid airports — there is a potential for condensation and corrosion. Dehumidifiers prevent these problems, saving calibration time and improving the mean time between failure of electronic systems.



Power plant layup

When power plants are idled for maintenance or because of excess generator capacity, dehumidifiers are installed to blow dry air through both the steam side and generator windings. This costs less and is safer than blanketing with nitrogen, and is far more effective than either wet layup with corrosion inhibitors or preservation in grease. Dry air layup is very simple, and the plant can go back on-line in a matter of hours rather than weeks.

Lithium battery production

Lithium, plutonium and other high-energy metals are hazardous because they ignite when atmospheric water vapor makes them corrode. Dehumidifiers make it possible to work with such metals quite safely in open air. Desiccant units dry large production rooms with dozens of people to moisture levels below 1% relative humidity. This has made the large-scale lithium battery industry economically viable. Without the desiccant dehumidifier, the lithium battery would still be an expensive, special-purpose curiosity.

Condensation Prevention

When cold surfaces are surrounded by moist air, water vapor will condense on the surface like “sweat” on a cold beer glass in summertime. 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. Alternately, condensation can form on hidden aircraft structural members as a plane descends from the cold upper atmosphere into moist environments, which can accelerate stress corrosion, shortening the life of the airframe. In both of these cases, dehumidifiers are installed to surround the cold surface with dry air.

Often, condensation control creates economic opportunities rather than simply preventing a problem. For instance, chilled rollers are used in many production processes to cool thin films or coatings. By blanketing the roll surface with dry air, the coolant temperature can be much lower without causing condensation. This means the product can be cooled faster, perhaps eliminating the need for a second machine.

Typical Applications

- Optical polishing
- Refrigerated display cases
- Environmental simulation
- Large gear cases
- High vacuum sputtering
- Gumball forming rolls
- Chilled rollers for film casting
- Refrigerated warehouse floors
- Altitude simulation cooling
- Cold product receiving rooms
- Typewriter ribbon film coating
- Fruit & vegetable storage
- Rock cave storage

Ice rinks

The ice temperature is cooler than the dew point of the surrounding air, so moisture condenses on the rink surface. The ice will soften and roughen, and the refrigeration system must work hard to keep the ice suitable for skating. Desiccant dehumidifiers are used to remove moisture before it can condense on the ice. Operating costs for the refrigeration system are reduced by thousands of dollars per year. The dry air also prevents hazardous ice from forming on surfaces around the rink, and eliminates corrosion of structural members. Dehumidifiers allow increased profits in warm weather; and they improve building safety in cold climates.

Water treatment plants

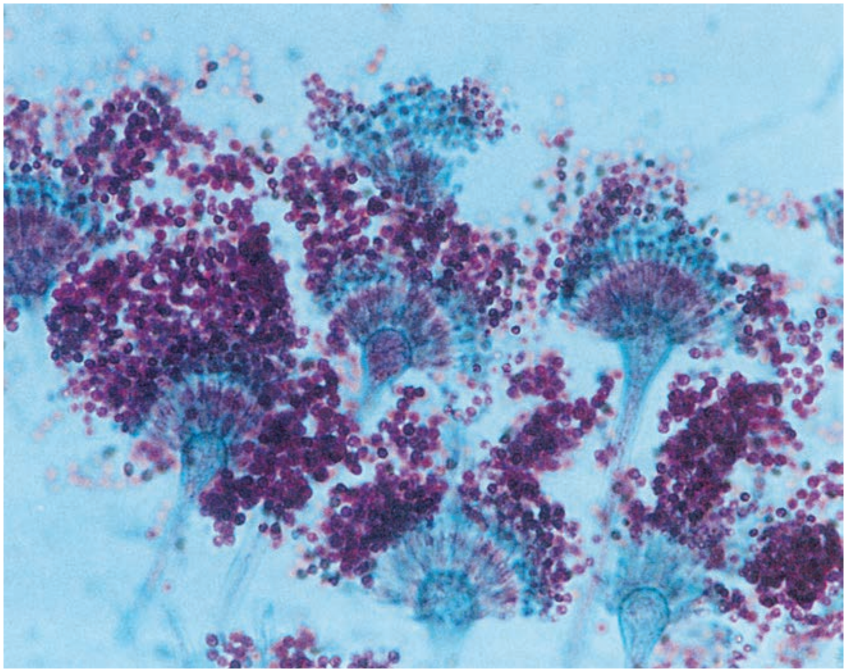
Ground water and lake temperatures are often much lower than air dew points, which can result in condensation on the outside of the pipes, valves and controls in the plant. Providing dry air to the plant prevents the condensation that leads to mold and fungus growth, as well as rusting the valves and controls. The cost of the dehumidifier is very modest compared to the cost of painting pipes and periodical replacement of controls and valves. Additionally, the sanitary benefits of eliminating fungus and bacteria have increased in importance as consumers become more sensitive to water quality issues.

Surface preparation & coating

Large, cool metal surfaces like ship interiors and chemical storage tanks must be periodically re-coated. Coating manufacturers will not guarantee the life of the coating unless the contractor can prove the surface was clean and dry when the coating was applied. Contractors now use desiccant dehumidifiers so they can blast and coat regardless of the weather. Dry air lets them keep a cleaned surface free from condensation indefinitely, which means they can schedule coating operations more economically. Dehumidifiers provide better coatings for end users and lower costs for contractors.

Injection molding

Chilled water is often provided to injection molds so plastic parts can be cooled rapidly. But if the chilled mold condenses moisture, the plastic surface is damaged, and part strength can suffer through thermal shock. Dehumidifiers blanket the mold surface with dry air, eliminating condensation. The coolant temperature can be reduced, which means the cycle time for each shot is reduced. Since machine time is often the key cost in highly competitive plastic molding markets, this processing speed gives manufacturers with dry molds a very important competitive advantage.



Mold/Fungus Prevention

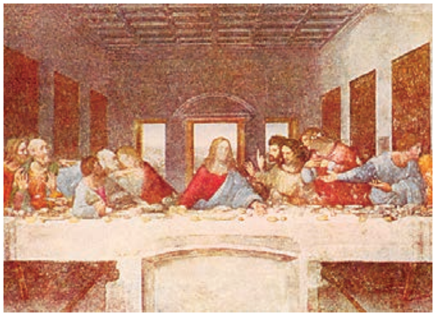
Mold and fungus are present in almost all materials. They can survive without moisture, remaining dormant for decades, even centuries. But when moisture and a food source become available they will multiply rapidly. This moisture does not have to be in liquid form. Microscopic organisms can use moisture present in solid materials because they need so little.

Two classic examples of this phenomenon are the prehistoric cave paintings in Lascaux, France, and the Egyptian artifacts preserved in the pyramids. The cave paintings survived virtually intact for 16,000 years. Then tourists began bringing moisture into the caves in their clothes and through their respiration. The paintings have deteriorated through microbiological attack in less than 40 years. Egyptian artifacts have had the same difficulty, with tragic consequences for history.

Even metals are attacked indirectly by certain bacteria. The U.S. Navy has traced corrosion in ships bilges to acidic by-products of microbiological metabolism, costing hundreds of thousands of dollars in repair expenses.

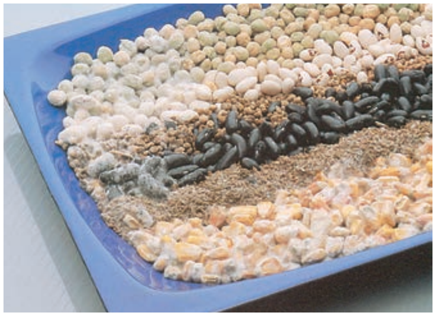
Typical Applications

- Grain storage
- Tulip bulb storage
- Food plant sanitation
- Rope fiber storage
- Historic building protection
- Photographic film storage
- Wooden sculpture preservation
- Fur storage
- Cocoa bean storage
- Dried fish storage
- Mummy preservation
- Wine cellars
- Underground food storage
- Milk powder storage
- Malt storage
- Museum storage
- Summer cottage winter protection
- Winter cottage summer protection
- Fabric & textile storage



Archival storage

Historic documents, photographs and art works are hygroscopic and often made of organic materials. When they absorb moisture, micro-organisms multiply and cause damage. Dehumidifiers are used to provide low relative humidity environments which prevents microbial attack. The dry air also allows these objects to be stored outside of temperature-controlled areas which saves money in building and heating costs. Many palaces in Great Britain and churches in Denmark are neither heated nor cooled — they are simply dehumidified — providing great savings in equipment and operational costs.



Seed storage

Organic materials are all subject to microbial attack, but few materials have the sensitivity of seeds. They are mechanically quite sturdy, but when seed moisture content is high, microbes can multiply and destroy the important nutrients they contain. Dehumidifiers allow preservation of rare seeds in scientific applications and they preserve seed value in commercial applications. In developing countries, grain losses in storage can exceed 30% of the crop. By preventing this cause of food shortages, dehumidifiers have the same effect as a 30% increase in crop yield.

Cargo protection

Lubrication oils, leather goods, canned foods, grain and fertilizers suffer from moisture in ocean transit. Bulk cargo like grain, flour, cocoa beans and tobacco is especially vulnerable because it may have a high moisture content when it embarks. Moist ocean air provides even more liquid to encourage microbiological growth. These problems can be prevented by drying the ventilation air supplied to holds and shipping containers. Drying the cargo prevents losses that can otherwise reach millions of dollars in a single shipment.

Breweries

Brewing requires large amounts of heat, water and malted grain. These are necessary for growing the yeast which turns natural sugars into alcohol. Yeast is a beneficial fungus, but other microbes can also grow in its place and disturb the brewing process. Dehumidifiers create dry environments for breweries, eliminating product contamination caused by destructive microorganisms.

Moisture Regain Prevention

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 just like thermal expansion, or make products that would otherwise flow freely stick together. The typical home salt shaker illustrates this point — moisture regain on humid days clogs the shaker holes. On a dinner table, this may be a minor problem. But in packaging machinery, sticky products have major economic consequences.

Hygroscopic products are sensitive to high relative humidities rather than absolute humidities, and relative humidity can be high at any time of year. In fact it is often higher in winter than summer. When products are stored at cool temperatures, problems can be especially acute. Desiccant dehumidifiers are effective in controlling humidity at low temperatures, and have been widely applied to prevent moisture regain.

Typical Applications

- Biomedical dry rooms
- Fertilizer packaging & storage
- Pharmaceutical tableting
- Foundry core storage
- Powdered drink mix packaging
- Plastic resin storage
- Instant coffee processing
- Sealed lead-acid battery filling
- Pre-preg material storage
- Paper storage
- Flour, plastic and powder airveying
- Sugar storage & packaging
- Epoxy powder manufacturing
- Pharmaceutical packaging
- Vitamin tableting & packaging
- Circuit board storage & machining
- Photolithography operations
- Precision color printing
- Cork storage
- Candy tableting & packaging
- Dried vegetable storage
- Propellant mixing & casting
- Missile motor storage
- Contact lens machining
- Filament winding
- Lens coating
- Veneer storage
- Power cable jointing
- Insulation for cold tanks
- Fruit powder handling
- Chewing gum ripening rooms

Candy packaging

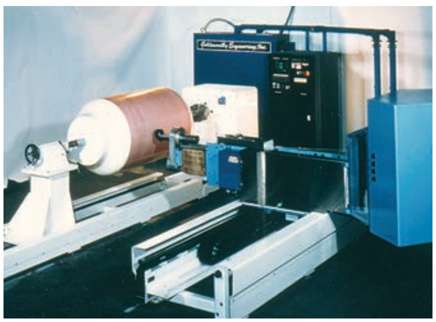
Hard candies often contain corn sugars and sorbitol, both of which are highly hygroscopic. When humidity is high, the product can absorb moisture and become sticky. Then it sticks to packaging machinery and wrapping material, slowing the process and creating sanitary problems. Desiccant dehumidifiers are used to keep packaging areas dry, letting equipment run efficiently and reducing the cost and time required for equipment cleaning.

Semiconductor and pharmaceutical clean rooms

When microcircuits are manufactured, hygroscopic polymers called photoresists are used to mask circuit lines for etching. These polymers are hygroscopic. If they absorb moisture, microscopic circuit lines are cut or bridged, resulting in circuit failure. In pharmaceutical manufacturing many powders are highly hygroscopic. When moist, these are difficult to process and have limited shelf-life. For these reasons and others, clean rooms are equipped with desiccant dehumidifiers, which prevent moisture absorption, allowing fast manufacturing of high-quality products.

Safety glass laminating

The thin, transparent plastic film which serves as the adhesive between layers of safety glass is quite hygroscopic. If allowed to absorb moisture, the film will boil it off in processing, creating steam bubbles which get trapped in laminated glass. Desiccant dehumidifiers create low-humidity environments for the manufacturing, storage and use of this laminating adhesive.



Composite manufacturing

Many uncured epoxy resins have a high affinity for moisture. When they absorb water vapor, the molecular sites which would otherwise cross-link to form strong bonds are blocked by water molecules. This slows down curing until the water is evaporated, or results in low strength parts. Printed circuit boards, filament-wound aerospace parts and pre-impregnated synthetic materials are all manufactured and stored in low-humidity environments to speed processing and improve product quality by preventing moisture regain.

Product Drying

Most products are dried using hot air to vaporize moisture and carry it away. Often, however, hot air is either too slow or results in damage to the product. Enzymes, for instance, are destroyed by heat, and if yeast is dried with very hot air, it cannot work properly.

When there is a benefit to drying at temperatures below 120°F, there is generally a benefit to using air which has been dehumidified rather than just heated. The cooler the temperature, the more the economics favor dehumidifiers. In one installation for example, enzymes used in commercial detergents are dried in a fluidized-bed drier. Because of the temperatures involved, the drying capacity of the fluid bed is doubled when the air dew point drops from 65°F to 20°. This means the size of the fluid bed can be cut in half.

The range of product drying applications for dehumidifiers has expanded significantly in recent years, as clients examine the positive effects of low-temperature drying on product quality. Dehumidifiers allow these quality improvements without sacrificing processing speed.

Typical Applications

- Uncured honey drying
- Photographic film manufacturing
- High voltage transformers
- Diagnostic reagent powders
- Whey powder
- Instant coffee creamer
- Pharmaceutical powders
- Yeast
- Sorbitol
- Sugar substitutes
- Chromatography materials
- Low-moisture welding rods
- Emergency oxygen “candles”
- PET plastic resin
- Nylon resin
- Expandable bead polystyrene
- Glass powder
- Paint pigment
- Tomato powder
- Cork
- Prosciutto ham
- Hard sausage
- Potato flakes
- Grated cheese
- Pasta
- Cigars
- Wax coated cheeses
- Seeds
- Tea
- Gelatin
- Katha cake
- Matches
- Onions & garlic
- Milk & whey powders
- Jelly beans
- Breakfast cereal
- Gelatin capsules

Investment castings

In this process, wax patterns are repeatedly dipped in a ceramic slurry. These layers build up to form a mold, which is filled with molten metal after the wax is melted out. Dry air rather than heat is used to dry the ceramic layers because hot air could deform the wax pattern. Using dehumidifiers lets investment casters operate foundries at consistent rates all year long, without having to allow for slower processing in humid months. This lets manufacturers improve compliance with clean air laws by substituting water for solvents. Productivity improves considerably — in many situations drying time is reduced by over 50%.

Plastic resin drying

Plastic resins are all hygroscopic to some degree. The moisture they absorb boils off when they are heated by the molding and extrusion processes when plastic pellets are formed into products. This boiling vapor creates structural and cosmetic flaws in the products, which reduces their value. Desiccant dehumidifiers are used to dry plastic pellets to very low moisture levels before they are melted, which improves end-product quality and raises profits for the fabricator.

Candy coating

Any coated candy like gumballs or chocolate-covered nuts can benefit from using dry air to speed drying. In the case of chocolate, high temperatures would melt or dent the product. With gumballs, the gloss of the coating is enhanced by using dry air, and the product is less likely to stick together in the coating pan. Coated candy or chocolate with a high-quality, glossy surface finish is almost always produced with the aid of dry air from desiccant dehumidifiers.

Fish drying

Delicate foods like fish are very sensitive to heat. Ideally, they should be dried at cold temperatures so bacteria will not grow and proteins will not break down. Such problems affect texture, flavor and sanitation. Desiccant dehumidifiers are used with cooling systems to dry fish efficiently at low temperatures. Without dehumidifiers, drying times at low temperatures would be too long to be economically practical.

Dry Cooling

Air conditioning systems generally use cooling coils to control both air temperature and humidity. In most comfort-conditioning applications, this is an extremely efficient method of controlling humidity.

However, in some circumstances, there is a benefit to adding a desiccant dehumidifier to the cooling system to control humidity separately from air temperature. The benefits are greatest where the moisture loads are high compared to the sensible heat loads, or where they peak at different times. For example, a building may be able to use 65°F outside air for cooling, but the air will still be carrying enough moisture to require dehumidification.

Also, when energy to regenerate a desiccant dehumidifier is very inexpensive and electric power is very costly, a desiccant unit can be a useful addition to an air conditioning system. It shifts the moisture load from electricity to a lower-cost utility. For example, cogeneration systems supply excess heat which can be used by a desiccant dehumidifier to remove moisture from air.

Retrofitted HVAC systems also take advantage of desiccant moisture removal capacity. This is of interest to many engineers since environmental concerns about chlorinated fluorocarbon refrigerants can lead to major changes in the latent removal capacities of existing systems.

Typical Applications

- Public buildings & auditoriums
- Nursing homes
- Health clubs
- Retail stores
- Hotels & motels
- Office building retrofits
- Supermarkets
- Medical office buildings
- Hospitals

Supermarkets

When refrigerated display cases operate they chill the air, automatically condensing large amounts of water. A desiccant dehumidifier removes water vapor much less expensively than the cooling system in the cases. If the air is already dry, cases do not condense moisture so they run more efficiently. The energy savings can represent the profit equivalent of hundreds of thousands of dollars in merchandise sales. In addition, since the store is dry, ice does not build up on frozen food and ice cream, which makes these high-margin products more appealing to customers.

Hotels and motels

Transient accommodations and conference facilities have large peaks and valleys in sensible heat loads. As people come and go, loads change radically. Moisture loads, on the other hand are quite constant because fresh air is brought into the building constantly, bringing along water vapor. Conventional cooling systems drop the room temperature so quickly that they shut off, leaving the moisture to build up, causing moldy odors and damaging the building structure and furnishings. Desiccant dehumidifiers are used to dry make-up air very deeply, so it can act as a "sponge" to remove excess humidity. This improves customer satisfaction and reduces maintenance costs.

Sick buildings

When air is cooled by a vapor-compression cooling system, it leaves the cooling coil essentially saturated with moisture. In some cases, the ducts downstream and the drain pan of the coil provide breeding places for mold and fungus. This growth contributes to indoor air quality problems and creates moldy odors. Drying the air with a desiccant dehumidifier upstream of cooling coils eliminates the problem — a useful feature in systems for medical facilities or other buildings where indoor air quality is an issue.

Advanced HVAC systems

Heating and air conditioning (HVAC) systems that use cogeneration for electrical power produce excess heat which comes from cooling the generator. This energy can be used to regenerate a desiccant dehumidifier, which removes the moisture load from the cooling system, improving overall system efficiency and saving operating costs. Also, systems which use ice storage to take advantage of low-cost electrical power can use desiccant dehumidifiers to produce very dry air. The combination of very dry and very cool air allows the system to reduce the total volume of air in the system, which in turn reduces the cost of ductwork installation and fan operation.

5
MOISTURE LOAD CALCULATIONS

When designing dehumidification installations, there are few tasks as important as quantifying the moisture loads that must be removed by the system. Without a clear understanding of the dimension of the loads, their frequency and source, it is impossible to design a system to remove them.

While the task is essential, moisture load calculation has historically been a somewhat controversial and personal procedure. That may seem odd, given the detailed calculations and apparently well-defined tables and graphs developed to assist the engineer. But the uncomfortable fact remains that two equally qualified individuals may arrive at different total moisture loads for the same space.

Differences arise from differing assumptions. Calculations for moisture leakage through a wall can be the same, but can the designer be sure the contractor will really seal the electrical junction boxes, or will the vapor barrier really be taped as specified? Will the room supervisor really control access to the space, or will doors be opening constantly? The engineer must make many assumptions — best estimates — before the calculation and system design can proceed.

In the final analysis, the moisture load calculation sheet represents a common agreement between equipment supplier, system designer, owner and installer. It defines the assumptions that form the foundation of the system design.

This chapter will discuss the basic elements of moisture loads, how they can be quantified, and most importantly, the relative importance of each element in different situations.

Selecting Design Conditions

The greater the difference in moisture between the controlled space and the surrounding environment, the greater the load will be from each load element. For example, assume a building is controlled at 70°F and 50% relative humidity (56 gr/lb), and the surrounding weather in the summertime is 90° and 95 gr/lb. Then each pound of air infiltrating into the room brings 39 grains of moisture that must be removed by the dehumidifier. But if the building is controlled at 70° and 2% rh (1.8 gr/lb), each pound of infiltrating air brings 93.2 grains into the room. With this in mind, it becomes clear that the first and most important step in calculating moisture loads is to determine the temperature and moisture conditions inside and outside the controlled space.

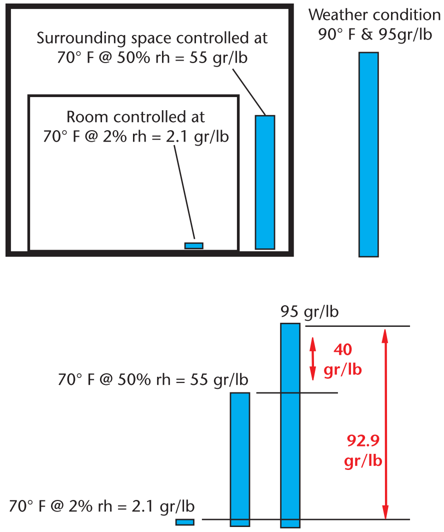


FIGURE 5.1

Loads depend on moisture differentials

All other things being equal, the moisture load is proportional to the difference between the specific humidity difference between the controlled space and the surrounding environment. The larger the difference, the greater the load.

The internal condition is determined by the product or process, and there are as many possible specifications as there are different applications. The important point about the control level is that it must be specified in absolute terms—grains per pound—before any calculations can proceed.

For instance; "30% relative humidity" is not a useful specification, because the moisture content of air at 30% rh varies with temperature. When the engineer further defines the temperature to be 70°F, the relative humidity value can be converted to an absolute value by consulting a psychrometric chart. In this case, 70°F and 30% rh is a moisture content of 32 grains of water vapor per pound of dry air.

If the application calls for a constant 30% humidity at fluctuating temperature, the designer must pick the temperature that — at 30% rh — will represent the greatest difference in absolute moisture between the inside and outside conditions. This is often the lowest temperature in the expected range, although the moisture difference will depend on what is happening outside at the time the low temperature and moisture condition occurs inside.

There are actually two sets of "outside" conditions which must be selected for the moisture load calculation. The first includes the conditions immediately surrounding the controlled space. These will be used to calculate loads from wall surface permeation, door openings and air leakage. The second set of conditions is the temperature and moisture of the fresh air brought to the space for ventilation or to replace air exhausted from the space.

In standard air conditioning design practice, an engineer often uses weather data compiled by ASHRAE (American Society of Heating, Refrigerating and Air Conditioning Engineers) which is contained in table 1B of chapter 27 of the 2001 *ASHRAE Handbook—Fundamentals*. However, that table contains three sets of design extremes, all of which have values for moisture. When designing dehumidification systems, *use the values for the peak dew point*, not for the peak dry bulb or peak wet bulb temperatures. Select one of the values from columns 4b, 4c or 4h of table 1B. Do not use the values from columns 2 or 3, because those are for cooling or evaporative cooling calculations. Peak cooling loads occur under drier conditions than do the peak dew point conditions. At most locations, the absolute humidity is 20 to 40% higher during peak dew point conditions than it is during peak dry bulb conditions.

The owner must apply some judgment to the selection of the outdoor design conditions. In columns 4b,4c and 4h, the owner has the choice of selecting extremes of moisture that are not likely to be exceeded for either 0.4%, 1% or 2% of the hours in an typical year. In other words, the outdoor moisture level will only exceed the stated values for either 35, 88 or 175 hours if the year is typical. If the engineer uses a 2% design condition, the owner should understand that the moisture loads could exceed the designers estimates for at least 175 hours during a typical year. The risk of being out of control for seven days — or parts of many more days — may be acceptable in some situations and not in others. The client must decide how conservatively the system should be designed.

The weather data for extreme moisture contained in the Appendix to this book was reprinted from the 2001 *ASHRAE Handbook—Fundamentals* (with permission from ASHRAE). If you are working on a project outside of the locations reprinted in this book, you may find it helpful to consult the ASHRAE Handbook itself, or the *ASHRAE Humidity Control Design Guide* (ISBN 1-883413-98-2), which also contains the complete set of peak dew point design data for 1453 worldwide locations.

Once the engineer has selected the inside and outside design conditions, the moisture load calculations for each element of the load can proceed.

Moisture Load Sources

There are seven principal sources of moisture:

- Permeation through floors, walls and ceiling
- Evaporation from peoples clothing, breath and perspiration
- Desorption from moist products, including packaging materials
- Evaporation from wet surfaces or open tanks
- Generation from combustion — open flame in the space
- Air infiltration through leaks, holes and door openings
- Fresh air ventilation from outside the space.

The importance of each of these sources depends on its size compared to the other sources, which varies by individual circumstances. Later in the chapter, we will review some examples which illustrate this point with specific numbers. But for the moment, it is useful to remember that the larger the individual load element, the more it demands careful attention. Conversely, if the load element is small, the details of its calculation may be less important to the success of the project.

Permeation moisture

Water vapor moves through solid materials at a rate proportional to the difference in vapor pressures on either side of the material. The moisture moves faster if the condition on one side of a wall is very dry than if air on both sides are similar in absolute moisture content.

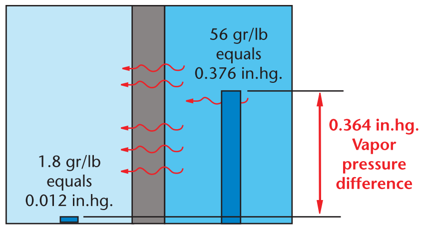


FIGURE 5.2

Permeation

Moisture will diffuse through solid material at a rate proportional to the vapor pressure differential across the material and inversely proportional to the materials porosity.

Also, each material has a different permeance rating, according to how much water vapor it will pass per square foot in a given period of time at a given vapor pressure differential. Since moisture travels through air more quickly than through solids, the permeance factor strongly depends on porosity of the material.

For example, a film of polyethylene plastic only .006 inches thick passes moisture at a rate of 0.06 grains per hour per square foot per inch of mercury column vapor pressure difference. On the other hand, an 8 inch thick concrete aggregate block passes 2.4 grains per hour, which is forty times faster than the film even though the thickness of the block is 1300 times greater than the plastic.

The vapor pressure difference is directly proportional to the difference between the specific humidity levels on either side of the material. In the normal range of engineering interest, each gr/lb of moisture corresponds to 0.0067in.Hg. vapor pressure. The designer can use this value to calculate vapor pressure difference when the difference across

TABLE 5.3

Building material permeance factors
The thickness of the material is not as influential as porosity. A plastic film only .006 inches thick is 40 times more effective in retarding vapor flow than a concrete block measuring 8 inches thick.

BUILDING MATERIAL PERMEANCE FACTORS[1]
grains/hr/sq ft/in hgΔV.P.

DESCRIPTION OF MATERIAL OR CONSTRUCTION	*PERMEANCE* No Vapor Seal	*PERMEANCE* With Vapor Retarder Paint (perm = .45)
Materials Used in Construction		
Brick, 4 inch Masonry	0.8	0.29
Brick, 8.5 inch Masonry	0.38	0.21
Concrete, 1:2:4 mix, 8 inch	0.40	0.21
Concrete, 1:2:4 mix, 1 inch[2]	3.20	—
Concrete Block, 8 inch	2.40	0.38
Plaster on Metal Lath, 3/4 inch	15	0.44
Plaster on Gypsum Lath (with studs)	20	0.44
Gypsum Wall Board, Plain 3/8 inch	50	0.45
Insulating Board, Sheathing, 1 inch[2]	50	—
Hardboard (standard), 1/2 inch	11	0.43
Plywood, Exterior Blue, 1/2 inch	0.35	0.20
Plywood, Interior Blue, 1/2 inch	0.94	0.30
Wood, Sugar Pine, 1 inch[2]	5.3	—
Insulation Materials		
Air (still) 1 inch[2]	120	
Corkboard, 1 inch[2]	9.1	
Fibrous Insulation (unprotected), 1 inch[2]	116	
Expanded Polyurethane Board, 1 inch[2]	1.6	
Expanded Polyurethane (extruded), 1 inch[2]	1.2	
Vapor Barrier Materials		
Aluminum foil, .002 inches	.025[3]	
Polyethylene, .002 inches	.16	
Polyethylene, .006 inches	.06	
Metal Deck, Built-up Roofs	0.0	
Paper, Felts		
Saturated and Coated Roll Roofing, 65 lb/100 sq ft	0.24	
Insulation Back-up Paper, Asphalt Coated	4.2	
Asphalt Coated Vapor Retarder Paper	0.6	
15 lb Asphalt Felt	5.6	
18 lb Tar Felt	18.2	
Single-Kraft, Double Layer	42	
Paints and Coatings		
Latex Vapor Retarder Paint, .003 inch	.45	
Commercial Latex Sealer, .0012 inch	6.28	
Various Primers plus 1 Coat Flat Oil Paint On Plaster	3.0	
2 Coats Aluminum Paint, Estimated	0.8	
2 Coats Asphalt Paint, Estimated	0.4	
2 Coats Flat Paint of Interior Insulation Board	4	

Notes:
1. Values shown above are estimates **only** based on a variety of test methods. When a range or more than one value is available, the higher value is shown. Contact manufacturer of materials being considered for exact values.
2. Permeance at a different thickness t, in inches, may be determined from the permeance value for 1 inch thickness by multiplying by the factor 1/t.
3. Permeance value shown is based on damage (pinholes) which may occur in handling.

Munters Table, primarily adapted from ASHRAE Handbook of Fundamentals, 1981, Chapter 21.

a material is known in grains. At high altitude, a psychrometric chart should be consulted since the vapor pressure-humidity ratio relationship is slightly different.

To calculate the permeation moisture load, read the material's permeance factor from the table and solve:

EQUATION 5.1

$$Wp = P \times A \times (\Delta VP)$$

Permeance (gr/hr)

Difference in vapor pressure across the material (in.Hg.)

Surface area of the material (sq.ft.)

Material permeance factor (gr/hr/sq.ft./in.Hg)

When a wall or package consists of more than one layer of material, the net permeance factor can be calculated the same way composite heat transfer coefficients are determined:

EQUATION 5.2

$$\frac{1}{P_T} = \frac{1}{P_1} + \frac{1}{P_2} \cdots\cdots\cdots \frac{1}{P_n}$$

Reciprocal of the composite material permeance factor (gr/hr/sq.ft./in Hg.)

Sum of the reciprocals of permeance factors for each material in the composite (gr/hr/sq.ft./in Hg.)

Moisture from products and packaging

If a solid material and surrounding air are in equilibrium, the moisture content of the material is proportional to the relative humidity of the surrounding air. Most solid materials absorb moisture from the air when it is moist, and give up the moisture when the surrounding air is dry.

When a moist material is brought into a dry environment, it will give up moisture until the water vapor pressure at its surface is the same as the vapor pressure in the surrounding air. The surface vapor pressure of a solid material depends primarily on its chemical and physical structure. So each material has different equilibrium characteristics. Surface vapor pressure is also proportional to both material temperature and moisture content, but since most controlled spaces are held at relatively constant temperature, moisture sorption and desorption has the deceptive appearance of being a function of relative humidity. Actually, there is a different moisture content — relative humidity equilibrium relationship for each different air and material temperature.

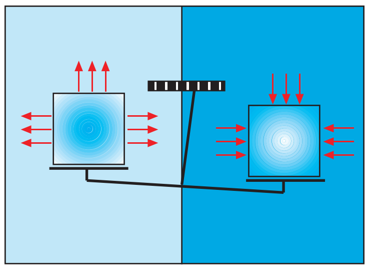


FIGURE 5.4

Product moisture

Materials collect and hold moisture when they are exposed to a humid environment. When they move to a less humid atmosphere, they give off the moisture they collected. This moisture load can be significant in applications like warehouses, where large amounts of moist cardboard packing material enter and leave the building regularly.

FIGURE 5.5

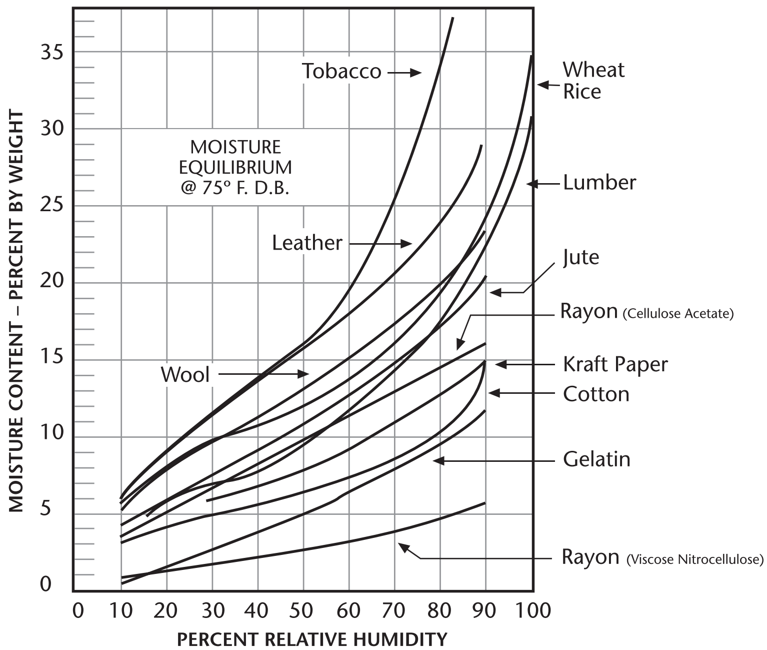


Figure 5.4 describes the equilibrium moisture contents of several different materials when their temperature is a constant 75°F. To calculate how much moisture they will give off when brought into a dry room, solve the following equation:

EQUATION 5.3

$$Wpp = lbs / hr \times (pw_1 - pw_2)$$

Water vapor from products & packaging (lbs/hr)

Total mass of material entering the room every hour (lbs/hr)

Equilibrium moisture content of material at the control condition in the space (lbs/lb)

Equilibrium moisture content of material before entering the space (lbs/lb)

Note that this equation makes the very conservative assumption that all the water vapor that can be desorbed from the material will be released in one hour. In fact, the process may take considerably longer. The exact rate depends on a host of variables — primarily the air velocity across the moist surface, the energy available for evaporation and the thickness of the material. For example, thin sheets of paper located under a warm, dry air discharge duct will dry within a few minutes. But cardboard boxes filled with paper and stacked on top of one another on a pallet make take days or even weeks to desorb moisture and reach a new equilibrium.

When the dehumidification system is designed primarily to dry a wet product like investment castings, seeds or powders, the equilibrium chart presented here will not be especially useful. Each product drying application is quite unique because the product is presented to the air a different way. Also, the product will have very specific and unique limitations of temperature, drying rate and drying air dew point. Experiments will be necessary to determine these characteristics. However, the engineer often has some methods of rough estimating at his disposal.

For instance, in an investment casting operation, the consumption rate of casting solutions must be known, because it represents a major cost item in the process. Since the water content of the solution is also known, the engineer can estimate water vapor load in pounds per hour by multiplying the number of gallons consumed per hour by 8.34 (water weighs 8.34 lbs per gallon). Another common example is the moisture load in a supermarket evaporating from moist vegetables. The manufacturer of the vegetable spraying system supplies the spraying rate for the equipment in gallons per hour, so the moisture load in lbs per hour is derived by multiplying the spraying rate in GPH by 8.34.

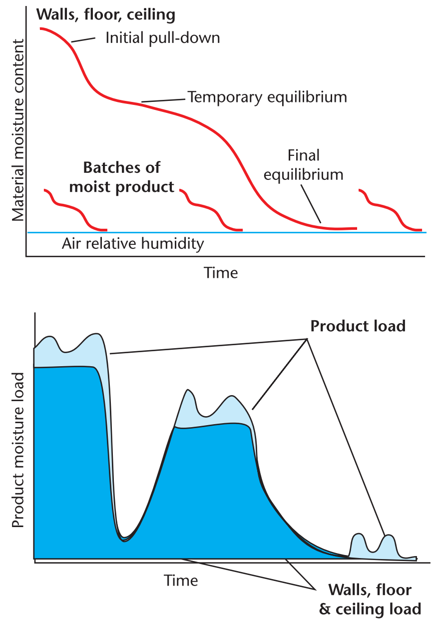


FIGURE 5.6

Product moisture loads

When a newly-constructed room is equipped with a dehumidifier, it may take some time before all the product moisture is pulled out of the material and into the air. The materials reach a temporary equilibrium after the surface moisture is removed. As the moisture migrates from the center to the edge of the material, more water vapor is liberated, and eventually the product reaches a true equilibrium.

If batches of moist product are brought into the room periodically, they will each behave like the building materials, loosing moisture in two stages. The net effect is that after initial construction, the moisture load varies as the building dries out. Even if a dehumidifier is running, the room humidity level can be expected to fluctuate somewhat until the building materials stop adding moisture to the air.

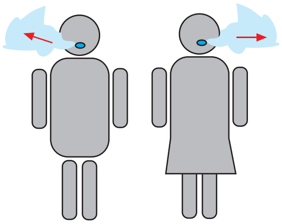


FIGURE 5.7

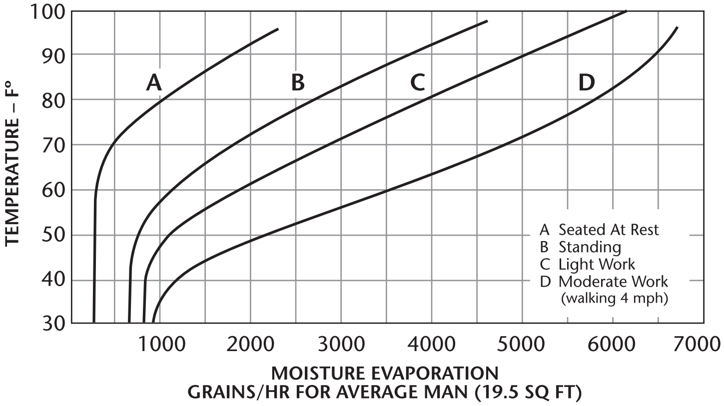
People

As they work in a dehumidified room, people breathe out moisture and evaporate perspiration from their skin. The rate depends on how hard they are working, and to some degree, it depends on the humidity level. For instance, manufacturers using super-dry rooms (below 2% rh) for production of lithium batteries and pharmaceutical products have found that the load from people is much lower than rates at moderate humidity levels. Apparently, people adapt to the low humidity, giving off less than half the moisture indicated by the graph at left.

Moisture from personnel

People breathe out moisture and release water vapor by perspiring. They also carry moisture into a low humidity space in their clothes. Each time someone breathes out, they release a lungful of air at a condition of 98°F and 283 grains per pound — essentially saturated. The number of breaths of air depends largely on the person's physical activity. More active people will breath more frequently and deeply. The chart provides typical moisture release rates for people engaged in different degrees of physical activity.

FIGURE 5.8



EQUATION 5.4

$$Wn = (Pa \times Fa) + (Pb \times Fb) + (Pc \times Fc) + (Pd \times Fd)$$

Moisture load from respiration and perspiration (gr/hr)

Load for people at moderate work (gr/hr)

Load for people at light work (gr/hr)

Load for people standing (gr/hr)

Evaporation per person (gr/hr)

Number of people seated

Load for people sitting (gr/hr)

In addition to this moisture, if a large number of people enter and leave the conditioned space frequently rather than simply staying in the room, the designer may need to consider water vapor from moist clothing. This is calculated like moisture desorbed from products and packaging materials. In most circumstances, the net amount of water vapor from clothes is almost negligible. However, when moisture control conditions are very low — below 5% relative humidity — or the personnel traffic very high, moisture desorbed from clothing can be significant.

For example, commercial buildings like supermarkets have a high moisture load from clothing since customers can enter the space directly from rainy, humid weather. Theaters and conference facilities may also have high intermittent loads from groups of people who enter from wet weather.

Moisture from combustion

In spaces where open gas burners are used for heating, the moisture that results from combustion can be a significant load. The exact amount of water vapor produced will vary with the composition of the gas, but where the value is unknown, the engineer can estimate that each cubic foot of gas burned produces 650 grains of water vapor.

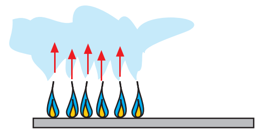


FIGURE 5.9

Combustion

When fossil fuels are burned, water vapor is one of the principal products of combustion. If the flame is open and un-vented, this moisture load must be included in calculations.

EQUATION 5.5

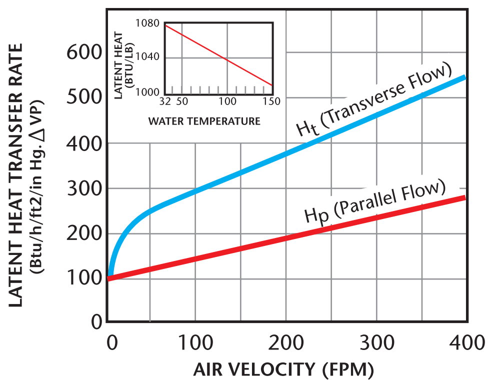
$$Wg = G \times 650$$

- Wg — Moisture load from gas combustion (gr/hr)
- 650 — Moisture produced per cubic foot of gas burned (gr/hr)
- G — Gas firing rate (cu.ft./hr)

Moisture evaporated from wet surfaces

In many food processing applications, the equipment in the space must be periodically washed and the floors cleaned. Also, in water treatment plants, there may be large, open tanks from which water will evaporate into the air. The rate of evaporation is directly proportional to the difference in vapor pressure between the surface and air, and also proportional to the rate that heat is transferred to the surface water film. The heat provides the energy necessary for evaporation, and the difference in vapor pressure provides the driving force to lift the water from the liquid surface into the air.

FIGURE 5.10



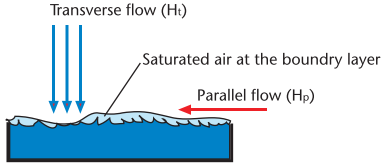


FIGURE 5.11

Wet surfaces

Moisture evaporates from wet surfaces quite slowly, even when the surrounding air is very dry. The rate increases dramatically when high velocity air is blown at the surface perpendicularly, and when the wet surface water is warm.

The water surface vapor pressure is high if the water temperature is high, since the air boundary layer is essentially saturated at the water temperature. So the highest evaporation rates occur when the water surface is warm, the air dry and when air travels rapidly and turbulently across the water surface. Fast, turbulent airflow provides high heat transfer rates from the air to the water film, and high mass transfer rates from the water to the air.

These relationships are expressed in the graph on the previous page, and the following equation, which originated from Dr. Willis Carrier's research in the 1920's:

EQUATION 5.6

$$We = \frac{H \times A \times (VPs - VPa) \times 7000}{H_L}$$

- H — Latent heat transfer rate (Btu/hr/sq.ft./in.Hg.)
- A — Total surface area wetted (sq.ft.)
- VPs — Vapor pressure of air saturated at the water temperature (in.Hg.)
- VPa — Water vapor pressure of the ambient air at its dew point temperature (in.Hg.)
- 7000 — Grains of water vapor in a pound of water
- H_L — Latent heat of vaporization at the water temperature (Btu/lb)
- We — Evaporation load from a wet surface (gr/hr)

Moisture from air leaks through cracks and holes

No building or air handling system is hermetically sealed. All rooms and ductwork will leak a certain amount of air, which carries a certain amount of moisture into the conditioned space. Typical air leak locations include:

- Cracks at duct joints
- Gaps in the vapor barrier film at wall-ceiling, wall-floor and wall-wall joints
- Gaps between wallboard and electrical fixtures that penetrate the wall
- Electrical conduits through which wires pass into a room
- Oversized holes for pipe and duct wall penetrations
- Gaps where strips of vapor barrier film are overlapped but not taped
- Cracks at access doors of air handling equipment casings
- Wall or ceiling penetrations to pass products in and out of the room
- Old ventilation louvers in walls and doors
- Cracks between the doors and their frames and sills.
- Open construction above hung ceilings
- Open doorways, loading docks or doorways with plastic strip curtains

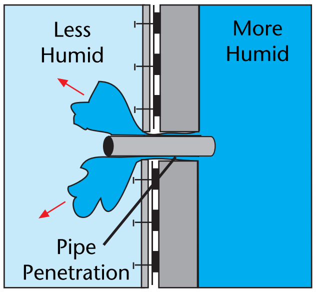


FIGURE 5.12

Air infiltration

The moisture load carried into the room by air flowing through cracks and wall penetrations is far more important than the load which results from diffusion through materials. For example, a crack 1/16th inch thick and 1 foot long will leak moisture at a rate of 1480 grains per hour in a light breeze. In contrast, an entire painted concrete block wall measuring 10 feet tall by 50 ft long will only leak 87 grains per hour under the same circumstances.

***Please Note:** This equation assumes the surface of the water is at or near ambient temperature. When hot water is used in wash down situations, the user should limit VPS to the vapor pressure of the air corresponding to the ambient wet bulb temperature.*

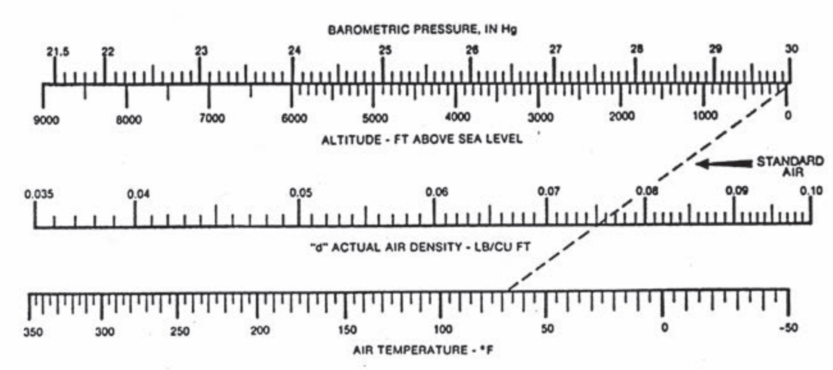


FIGURE 5.13

Air density nomograph

Air infiltration is measured in cubic feet per unit time, but the weight of the air — therefore the weight ratio of water vapor to air (gr.lb.) — depends on air density per cubic foot. Density changes with temperature and total pressure. This nomograph allows the designer to find the correct density per cubic foot for different air temperatures and pressures.

EQUATION 5.7

$$Wi = A \times d \times 60 \times Va \times (M_o - M_i)$$

- Moisture load from air infiltrating through an opening (gr/hr)
- Area of the opening (sq.ft.)
- Moisture inside the space (gr/lb)
- Moisture outside the space (gr/lb)
- Air velocity through the opening (ft/min)
- Minutes per hour
- Density of the infiltrating air (lb/cu.ft.)

The simplicity of this equation is somewhat deceptive. The difficulty comes in establishing the open area of numerous small cracks and holes, and in determining the air velocity through the openings. Differences in assumptions concerning these values are the root of differences between calculations performed by two equally well qualified system designers. Some of the issues and different approaches to the problem are outlined below:

• *Exterior wall leakage*

Light commercial and industrial buildings sided with vertical, rolled steel panels present a classic problem in estimating air infiltration. In theory, the metal walls should be an excellent vapor barrier, but in fact, the siding can leak air through seams between panels, and where walls meet other walls, the ceiling and floor. There are thousands of feet of potential cracks.

When working with an existing building, the designer can measure infiltration quite accurately by engaging an air test and balance contractor to perform a blower-door test of the building's ability to maintain both positive and negative internal air pressures. The building is closed, a special panel equipped with a fan covers the door, and then the pressure differential across the building wall is compared with the fan airflow. Then the airflow is reversed and measured once again.

Measurements for both positive and negative pressures are important since some openings, cracks and gaskets leak more readily in one direction than the other. For example, window seals are designed to resist infiltration from outside the building, but will blow away from the frame if the building is placed under positive pressure, leading to a false conclusion about actual infiltration potential. Properly performed, such blower-door tests measure air leakage more accurately than what can be determined with theoretical calculations.

For larger spaces, the U.S. Brookhaven National Laboratory has developed a tracer gas air infiltration measurement technique called the AIMS test. A tracer gas emitter and receiver are placed in the space for a two to four week period. Then the receiver is sent to a laboratory for analysis. The amount of tracer gas it collected is inversely proportional to the air leakage. In other words, as air leaks in and out of the building, there is less gas absorbed by the receiver because it has been diluted by the fresh air. Details of this procedure are available from the U.S. National Institute of Standards and Technology, Buildings Research Division.

Chapter 23 of the 1989 *ASHRAE Handbook—Fundamentals* deals extensively with the subject of air infiltration through the building envelope. The information in the tables presented here has been adapted from that data, and provides a means of estimating exterior wall leakage on buildings that have not yet been constructed.

In using the tables, note that if all four sides of a conditioned space are exposed to the weather, not all will be receiving full wind pressure simultaneously. Once the air leak rate has been established, the designer can use the following equation to establish the moisture load through an exterior wall.

TABLE 5.14
Air Leakage

Air Leakage Through Wall Components ($ft^3/hr/ft^2$)

Pressure Difference		1	2.5	5	10	15	20
	mph	1	2.5	5	10	15	20
	fpm	88	220	440	880	1350	1750
	in.wc.	0.0005	0.003	0.012	0.048	0.11	0.193
Older windows							
Wood	loose fit	3	8	20	49	81	122
Wood	Avg. fit, no weather strip	1	3	7	17	28	43
Wood	Avg. fit, w. weather strip	0.5	1.5	4	9	15	23
Wood	Casement, w.s.	0.2	0.6	1.5	4	6	9
Metal	Casement, pivoted	6	18	43	108	176	262
Metal	Double-hung, no w.s.	2.5	8	19	47	77	114
Metal	Double-hung, w.s.	1	3	8	19	32	47
Newer windows							
	Wood & metal	0.5	1.5	3.6	9	15	22
Doors							
Sliding	Aluminum	1.5	4	10	26	43	64
Sliding	Wood	0.7	2	5	13	21	31
Hinged	Well-fitted	0.5	1.3	3	8	13	19
Hinged	Well-fitted + w.s.	0.3	1.3	3	8	7	11
Hinged	Poorly-fitted (1/16″ crack)	5	16	40	100	158	240
Door & window frames							
	Masonry wall, uncaulked	0.5	2	4	10	18	26
	Masonry wall, caulked	0.1	0.4	0.8	1.5	3	5
	Wood frame wall	0.5	1.5	3.5	9	14	21
Brick							
	8.5″ plain	0.35	1	2.6	5	9	16
	8.5″ w. 2 coats of plaster	n/a	0.010	0.025	0.050	0.80	0.140
	13″ plain	0.3	0.95	2.3	5	8	14
	13″ w. 2 coats of plaster	n/a	n/a	n/a	0.010	0.040	0.050
	13″ w. furring, lath & plaster	0.01	0.03	0.08	0.3	0.38	0.46
Frame walls							
	3 coats of plaster	0.01	0.02	0.04	0.09	0.16	0.22
Metal walls							
	Tight joints	n/a	n/a	0.013	0.03	0.05	0.08
	Average joints	n/a	0.015	0.04	0.09	0.17	0.23
	Loose joints	0.01	0.03	0.08	0.19	0.33	0.46
Cracks [1]							
	1/16″	7	23	53	130	217	315
	1/8″	13	45	105	255	435	630
	1/4″	25	90	210	515	865	1260

[1] These values represent cubic ft of air leakage per hour **per linear ft.** of crack. Equation 5.8 can still be used for the calculation.

EQUATION 5.8

$$Wi = Q \times d \times (M_o - M_i) \times A$$

Moisture carried through cracks in an exterior wall (gr/hr)
Air density (lb/cu.ft.)
Area of the opening (sq.ft.)
Moisture inside the wall (gr/lb)
Moisture outside the wall (gr/lb)
Air leakage rate (cu.ft./hr/sq.ft.)

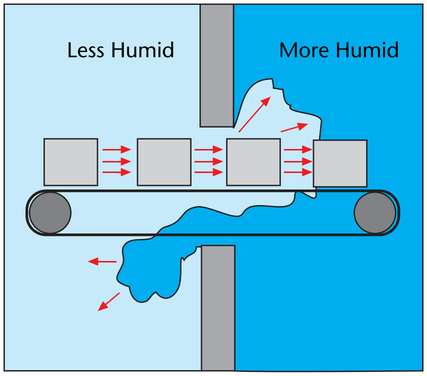


FIGURE 5.15

Conveyors

Product riding a conveyor will push dry air out of the room, creating a local low pressure area under the conveyor. Moist air from outside the room will flow into the room to equalize this pressure difference.

• *Air leakage through conveyor openings*

Products riding on a conveyor will push air out of the room at roughly the speed of the conveyor belt. The air will be replaced by a counter-flowing moist airstream which enters through the same opening below and beside the conveyor. A conservative calculation assumes the full open area of the wall opening has moist air flowing into the room at the speed of the conveyor. Designers can apply judgment to each situation, perhaps reducing the open area assumption if plastic strip curtains are placed over the opening, or very flat product rides the conveyor.

EQUATION 5.9

$$Wi = V \times A \times d \times 60 \times (M_o - M_i)$$

Moisture carried by air through a conveyor opening (gr/hr)
Moisture inside the room (gr/lb)
Moisture outside the room (gr/lb)
Minutes per hour
Air density (lb/cu.ft.)
Conveyor opening area (sq.ft.)
Conveyor velocity (ft/min)

• *Moisture infiltration against out-flowing air*

To prevent moisture from being carried into a room by air currents, system designers often bring enough fresh air through the dehumidification system to provide a slight positive pressure on the room relative to the surrounding area. That way, air is always flowing out of rather than into the room through cracks, wall openings and doors. This is a very effective technique for small openings like cracks, but hard experience has shown that positive room air pressure does not stop all moisture from entering the room through larger openings like open doors or conveyor openings.

In past years, system designers believed that counterflow infiltration was caused by the large vapor pressure difference between moist air outside the room and dry air inside. Since the vapor pressure difference can be measured in inches of mercury, and air pressure simply in inches of water column, engineers reasoned that the vapor pressure difference could overcome air pressure differences and result in moisture flowing against an airstream. In fact, recent research has shown that while true, the absolute amount of moisture transferred by this diffusion mechanism is very small — even negligible — compared to that transferred by two other mechanisms: thermal differences across a large opening and velocity-induced, low-pressure areas at the edges of small openings.

Purely empirical experience suggests that maintaining 150 feet per minute out-flowing air velocity will reduce the counter-flowing moisture to negligible amounts provided that precautions are taken by the designer. For large, tall openings like warehouse doors, plastic strip curtains are used to block airflow resulting from pressure differences created by thermal differences across the opening. The airflow necessary to maintain 150 feet per minute is calculated using the remaining open area after the strip curtain is installed, remembering that there are still cracks between the strips which open and close with air movement and door activity.

In smaller openings, counter-flowing air infiltration can be limited by installing short (two foot long) tunnels around the opening. This moves the low-pressure area at the edge of the opening outside the room, effectively eliminating counterflow air at 150 foot per minute outward air velocities.

This experience is presented for the designer's information, but should be used with the clear understanding that the theoretical basis of the empirical evidence is weak and all mechanisms of moisture transport in this situation are not well understood. Out-flowing air velocities of 150 fpm, tunnels and plastic strips appear to work well in limiting counterflow infiltration at higher humidity control levels, but when moisture differentials are very large, such as when a room is being maintained below 15% relative humidity at 70°F, they do not appear to be effective. In those situations, the designer must assume that moisture does enter through the opening in spite of these precautions. Conservative calculations assume a 50 foot per minute airflow into the room through the full area of the opening.

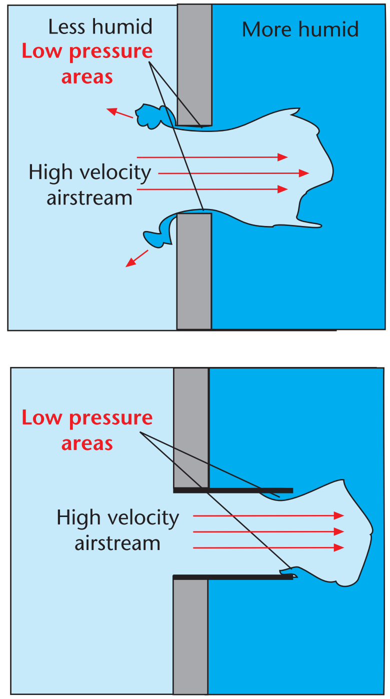


FIGURE 5.16

Ducted openings

Moisture infiltration through smaller openings can be greatly reduced by providing enough air to the room to maintain an outward air velocity of 150 feet per minute. But the beneficial effect is limited unless the opening is ducted, so that low pressure areas caused by the high velocity are moved well outside the room wall.

• ***Air infiltration as doors are opened***

When a door opens, it creates local, short-term pressure differences and air turbulence that can pull in air even if the room is under positive pressure. If a door opens to the weather, assume the local average wind velocity governs the airflow rate though the door for the amount of time the door is open. If a door opens to another space, assume there is an air current of 50 feet per minute into the room for the time the door remains open, unless there is better specific data available.

• ***Air infiltration through air lock vestibules***

A double-door air-locked vestibule is a common feature of many low-humidity rooms. Dehumidification system economics make airlocks almost essential for cold storage facilities and dry rooms maintaining air dew points below 10°F. A conservative calculation assumes that each time the air lock is opened, the air in the lock comes to a moisture condition half way between the condition outside the room and the moisture level inside:

EQUATION 5.10

$$Wi = (h \times l \times w) \times d \times \left(\frac{M_o - M_i}{2}\right)$$

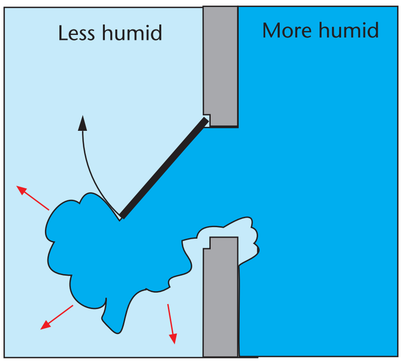


FIGURE 5.17

Door activity

Each time a door is opened, humid air is pulled into the controlled space by eddy currents in the air produced by the motion of the door. To maintain very dry environments, one way to reduce the load from this source is to place an air lock vestibule on each doorway.

• ***Number of door openings per hour***

The most fragile estimate a designer will make is his or her judgment of the number of times a door will be opened in a typical hour. If the room has been built and is being used, the designer is urged to perform a visual survey of door activity, or install a counting mechanism. The number of door openings will generally be surprisingly high. The activity is seldom less than two door openings per person per hour, and often much higher.

The moisture load this represents can be discouraging. It prompts the designer to have heart-to-heart conversations with supervisors about minimizing door activity, and then use a more optimistic and smaller number of openings for calculations. However, the designer should remember that supervisors and personnel change, and if the system is short of capacity, nobody is likely to remember promises made years earlier by different people. When in doubt, the designer is encouraged to make conservative rather than optimistic estimates of door activity.

Periodic vs. continuous loads

In the calculations above, we have treated moisture loads as a continuous, even flow of moisture into the room from a variety of sources—essentially a constant rate every hour. In fact, loads seldom occur so uniformly. For example, people entering a dry room for work will do so at 7:00 am and leave for a lunch break at 12:00 noon. So the number of door openings per hour is very high at the beginning of the shift and at the end, and between those times, the door activity is very low. The designer has two choices: average the intermittent loads or use the peak hourly load for the calculation. Each choice has advantages and drawbacks.

If the designer uses the peak intermittent load, the system will always have enough capacity to maintain control conditions, but it will be oversized for average circumstances. This means the system will be difficult to control, will use a large amount of energy and will cost more to install. If the designer averages the peak loads with periods of low loads, the system will be easier to control than a larger system, and will cost less to install and operate. On the other hand, it may not have enough capacity to maintain conditions if all peak loads occur simultaneously.

The issue is only as important as the size of the intermittent load. If the intermittent load is small compared to the total load, a minor excursion from the control condition may be inconsequential. Likewise, it may not be important to maintain control when the intermittent load occurs. For instance, it is seldom important that the humidity level rises when a food plant is sprayed down for cleaning — the more important issue is how long it will take to dry. However, if the maximum humidity level is critical, the designer should make plans to design a large system to handle the peak load, or design a second system to supplement the basic system when large, intermittent loads occur.

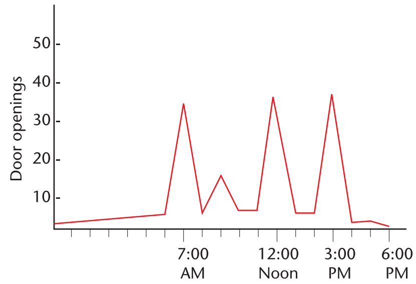


FIGURE 5.18

Intermittent loads

Many loads are not continuous—they come in large bursts at different times of the day. The designer must decide if it is important to maintain control at these peak times, or if it may be better to run the risk of being out of control for short periods in order to reduce the cost of the system and improve its operating efficiency. The decision generally depends on the relative size of the periodic loads and therefore how long a smaller system would take to recover from load peaks—minutes, hours or days.

Fresh air moisture load

All of the moisture loads discussed above are internal. That is to say they occur inside the room, which is *down stream* of the system. However, generally the largest moisture load originates in the fresh air brought to the system *upstream* of the dehumidifier. Any moisture carried by fresh air will be removed before it enters the room. This is important, because if fresh air moisture went directly to the room, the system would have to be much larger.

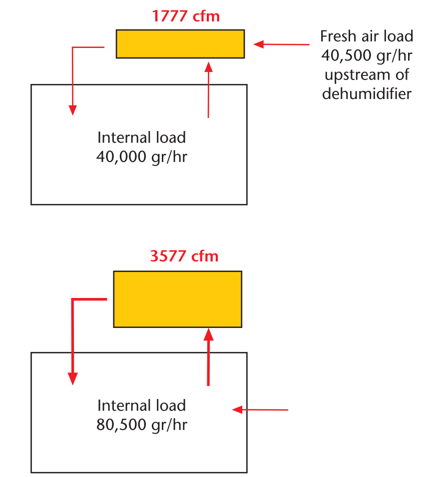


FIGURE 5.19

Minimizing internal loads

Removing the moisture from the ventilation air before it enters the room will greatly reduce the size and cost of the dehumidification system. Assuming the designer will choose to do this, it is important to keep the ventilation load calculation separate from the internal loads to avoid distorting the system dry air requirement.

For example, assume a system produces air at a moisture level of 25 gr/lb in order to maintain a control level of 30 gr/lb in the room. If the internal moisture load is 40,000 grains per hour, the system must supply 1,777 cfm of 25 grain air to maintain the 30 grain condition. Dry air requirements are calculated as follows:

EQUATION 5.11

$$Q = \frac{W_t}{d \times 60 \times (M_m - M_c)}$$

Air flow rate required to remove the moisture load while maintaining the specified control level in the space (cfm)
Total internal moisture load (gr/hr)
Moisture level of dry air supplied to the room from the dehumidification system (gr/lb)
Moisture control level inside the room (gr/lb)
Minutes per hour
Density of air (lbs/cu.ft.)

$$Q = \frac{40{,}000}{.075 \times 60 \times (30 - 25)}$$

$Q = 1777$ *cfm*

If 100 cfm of air at 120 gr/lb is brought directly to the room from the weather for ventilation, it adds 40,500 grains per hour to the internal load. Solving the same equation for a total internal load of 80,500 gr/hr shows the system must now supply 3,577 cfm at 25 gr/lb to maintain the 30 grain control condition, more than doubling the size of the system.

This is why fresh air moisture is removed by the system before it enters the room whenever possible — the practice results in smaller, more cost-effective dehumidification systems. *Therefore, the designer should be careful not to simply add the ventilation load to the internal loads during calculations* — it would lead to grossly oversized systems.

There are three reasons to bring fresh air into a system: to provide ventilation for people and make-up air for exhaust hoods or fans, or to maintain a positive air pressure in the room compared to the surrounding environment. The amount of air for each of these functions will vary considerably between different applications.

• *Ventilation for people*

ASHRAE standard 62-1999 (Ventilation for acceptable indoor air quality) establishes a minimum fresh air requirement of 15 to 25 cfm per person. The exact amount varies according to the type of building and it's use. However, local building codes can vary. The designer is obliged to follow local law where it requires more ventilation than the minimum amounts suggested by ASHRAE in their Standard 62.

• *Make-up air for exhaust fans*

Fume hoods pull air from the room which must be replaced by fresh air from outside. The amount will vary depending on the size of the hood and the individual fan characteristics. Most fume hoods pull enough air to maintain an air velocity of 150 feet per minute across the open hood area. But designs vary considerably, so the designer must check the air quantity with the hood manufacturer. The same conversation must cover the issue of fan control. If the hood fan is not controlled it will pull air from the room at different rates depending on pressures inside and outside the hood system. In any humidity-control application with an exhaust fan, the designer must insure its airflow is known and controlled.

• *Make-up air for room pressurization*

The amount of air necessary to develop and maintain a positive room pressure depends on the open area through which air will leak out of the room, and the shape of the edges of the openings. *Fan Engineering,* a handbook published by the Buffalo Forge Company of Buffalo, NY, provides the following formula for estimating airflow necessary to develop a specified positive pressure on one side of a rectangular, square-edged opening:

EQUATION 5.12

$$Q = 4005 \times 0.60 \times A \times \sqrt{SP}$$

Air flow rate required to maintain a specified positive air pressure in a room with a square-edged wall opening (cfm)

Specified static pressure differential (in.wc.)

Area of the opening (sq.ft.)

Coefficient of entry for air entering a square-edged opening (dimensionless)

Calculation factor (dimensionless)

For smaller, crack-like openings, the relationship changes somewhat because of the greater airflow resistance of turbulence at the edges compared to smooth flow through the middle of the opening. The last term in the equation changes from $SP^{0.5}$ (the square root of static pressure) to $SP^{0.65}$.

When the total fresh air requirement is known, the designer can calculate the moisture load it represents for the system. Note that if an exhaust system or room pressurization air operates continuously, it may satisfy part or all of the requirement for personnel ventilation.

EQUATION 5.13

$$Wm = Q \times d \times 60 \times (M_o - M_i)$$

Moisture load from fresh air (gr/hr)
Moisture level inside the room (gr/lb)
Moisture level of the fresh air (gr/lb)
Minutes per hour
Air density (lb/cu.ft.)
Sum of airflows necessary for ventilation, pressurization and exhaust air make-up (cfm)

Sample moisture load calculations

As described at the beginning of this chapter, load calculations require a series of assumptions beyond simple, quantifiable equations. Two equally qualified and experienced individuals working separately may well come to different conclusions on the same project.

For best results, the equipment supplier, system designer, installing contractor and end user must all be aware of each assumption, and they should agree on the final value for each load element.

The examples which follow show the typical thought process and subsequent mathematics for two different kinds of projects: a relatively passive storage application where the moisture control level is quite high, and an active manufacturing environment where the loads are high and the moisture control level is low.

Beyond these two examples, Chapter 7 (System Design) provides five more projects which include load calculations, although in somewhat less numerical detail than what appears here.

Warehouse dehumidification

An electro-mechanical parts manufacturing operation located in Chicago, Illinois stores a finished inventory of machined parts and raw material in an older warehouse near the plant. There has been a problem with corrosion on highly polished surfaces. The purpose of this project is to limit the relative humidity to a maximum of 40% at all times of the year, in order to eliminate the re-work that results from corrosion.

The project engineer examines the weather data for the local area and decides to use summer extreme conditions of 95°F and 146 gr/lb, and winter design extremes of -5°F and 4 gr/lb. These are the 1% extremes. The engineer reasons that while using them will result in a large system, the annual cost of rework has exceeded $250,000, so the incremental cost of a larger dehumidification system is likely to be paid back in a matter of months. Chicago is at an elevation of 658 ft above sea level, so the engineer will use an air density of 0.070 lbs/cu.ft. for calculations. The average annual ground water temperature in the area is 52°F, which the engineer will need to calculate moisture permeation through the concrete floor.

To establish the inside temperature and moisture control conditions for the calculations, the engineer must determine what the lowest moisture content will be inside when the moisture is highest outside — in other words, the conditions which create the largest vapor pressure differential. Local weather data suggests that the extreme outside moisture of 146 gr/lb occurs when the outside dry bulb temperature is 85°F. At that outside temperature, the building is generally at 75°F. The internal moisture level will be 52 gr/lb when inside conditions are 75° and 40%rh.

The building measures 250 ft long, 75 ft wide and 14 ft high. It is attached to the plant by a passageway for transport of material. There are two loosely-fitted doors measuring 10 ft x 10 ft. One door opens into the plant passageway, and the other is a loading dock. The loading dock door opens about twice an hour for thirty seconds each as trucks pull into and out of the loading platform. The plant passageway door opens eight times an hour, also for an average of thirty seconds each time so material can pass to and from the warehouse.

The building walls are brick, 8.5 inches thick, and the interior surface has been painted with a vapor-retarding industrial latex paint. The floor is an 8 inch thick concrete slab, poured on packed crushed stone without a vapor barrier. The roofing is metal-deck, built-up construction.

Two packing and inventory clerks work in the building during the day shift, and the parts are packed in heavy cardboard on wooden pallets. The cardboard totals approximately 5,000 lbs dry weight. The material turns over about six times a year.

Inside the building, the space is open, without partitions or internal rooms. There is no air cooling system or fresh air ventilation apart from what leaks in through the building walls.

The calculation sheets on the following pages show how the load is calculated. Some of the engineer's thoughts include:

• ***Weather design conditions***
The biggest problems with corrosion seem to occur in the spring and fall, when rising weather air moisture actually causes condensation on the cool metal surfaces of the raw material and finished parts. The engineer could have chosen a lower spring or fall moisture extreme, but decides to take the summer condition for maximum safety. However, since the dehumidifier must operate year round, the engineer will select the reactivation heaters based on the winter air temperature to insure adequate heater capacity for desiccant regeneration.

• ***Building penetrations***
The weather gets very muggy and uncomfortable during summer months, and the roof of the building has ventilators installed to avoid heat build-up in the warehouse. These must be covered with plywood and sealed with metal foil tape. The engineer assumes this will be done, and also allows a modest budget for small, portable "man-cooler" units for the workers assigned to the building. Also, the two inactive doors are sealed with metal foil tape to avoid air leaks, and the broken glass panel in one of the active doors is replaced. Finally, the oversized wall openings for the heater vent pipes and gas supply pipes are sealed with sheet metal collars and glass fiber insulation.

• ***Observations concerning the loads***
Calculations show the importance of limiting air infiltration through cracks and door openings. An investment in sealant and better gaskets would allow a smaller, less costly system with lower operating costs. The even larger load comes from the open shipping door. The load could be reduced somewhat by plastic strips in addition to the solid door, and it may help to put an alarm light or bell on the door to remind workers to close the door as rapidly as possible after a truck pulls through. Also note that another coat of vapor-barrier paint or expensive vapor barrier film to limit permeation would not change the load by any significant amount. The owner will be better served by concentrating on reducing cracks and shortening door opening times.

Moisture Load Calculation Sheet

Project Data

Parts Warehouse
Project Name

Chicago, IL
Location

Corrosion protection
Application

M. McDonald — 2/02
Calculations made by (name) — Date

S. Brickley — 3/02
Approved by (name) — Date

Purpose of the project:

Reduce the cost of re-work by preventing storage corrosion of finished goods inventory

Design Conditions

	Weather Extremes Summer	Weather Extremes Winter	Ground water	Internal Conditions Room	Internal Conditions Building
Dry Bulb Temperature	95	-5	52	75	75
Dew point	76	-4	52	49	58
Humidity Ratio	14	65	58	52	72
Vapor pressure	0.980	0.001	0.395	0.350	0.480

Elevation 658 ft. above sea level—standard air density = 0.070 lb/cu.ft.

Permeation

	Surface Area (sq.ft.)		Permeance Factor (gr/hr/sq.ft.)		Larger Vapor Pressure (in.hg)		Smaller Vapor Pressure (in.hg)		Permeation Load (gr/hr)	
Wall 1	1050	x	0.21	x (	0.98	-	0.35	) =	140	gr/hr
Wall 2	1050	x	0.21	x (	0.98	-	0.35	) =	140	gr/hr
Wall 3	3500	x	0.21	x (	0.98	-	0.35	) =	463	gr/hr
Wall 4	3500	x	0.21	x (	0.98	-	0.35	) =	463	gr/hr
Floor	18,700	x	0.21	x (	0.395	-	0.35	) =	177	gr/hr
Ceiling	–	x	–	x (	–	-	–	) =	–	gr/hr
								Total	1,383	gr/hr

Products, Packaging & Clothing

	Item Entry Rate (lb/hr)		Original Moisture Content (lb/lb)		Final Moisture Content (lb/lb)		Grains Per Pound		Moisture Load (gr/hr)	
Item 1	3.42	x (	0.11	-	0.07	) x	7000	=	958	gr/hr
Item 2		x (		-		) x	7000	=		gr/hr
Item 3		x (		-		) x	7000	=		gr/hr
Item 4		x (		-		) x	7000	=		gr/hr
								Total	958	gr/hr

$\frac{5{,}000 \text{ lbs} \times 6}{8{,}760 \text{ hrs/yr}} = 3.42$ lb/hr

Personnel

	Number Of People		Moisture Load (gr/hr/person)		Moisture Load (gr/hr)	
Seated		x		=		gr/hr
Standing		x		=		gr/hr
Light work		x		=		gr/hr
Moderate work	2	x	5500	=	11,000	gr/hr
Room visitors		x		=		gr/hr
				Total	11,000	gr/hr

Open Gas Flame

Gas Burning Rate (cu.ft./hr)		Water Vapor Generation (gr/cu.ft.)		Moisture Load (gr/hr)	
	x	(Typical Value 650 gr/cu.ft.)	=	NONE	gr/hr

Wet Surfaces

[Wetted Surface Area (sq.ft.) ____ x Latent Heat Transfer Rate (Btu/sq.ft./in.hg.) ____ x (Water Surface Vapor Pressure (in.hg.) ____ - Air Vapor Pressure (in.hg.) ____) / Latent Heat Of Vaporization At The Water Temperature (Btu/lb) ____] x 7000 (Grains Per Pound) = NONE Moisture Load (gr/hr) gr/hr

Exterior Walls

	Surface Area (sq.ft.)		Air Infiltration Rate (cu.ft./hr/sq.ft.)		Moisture Outside (gr/lb)		Moisture Inside (gr/lb)		Air Density (lb/cu.ft.)		Moisture Load (gr/hr)	
Wall 1	1050	x	0.06	x (	146	-	52	) x	0.07	=	415	gr/hr
Wall 2	3500	x	0.06	x (	146	-	52	) x	0.07	=	1382	gr/hr
										Total	1797	gr/hr

Cracks

	Crack Length or Component Area (ft or sq.ft.)		Air Infiltration Rate (cu.ft./hr/ft.)		Moisture Outside (gr/lb)		Moisture Inside (gr/lb)		Air Density (lb/cu.ft.)		Moisture Load (gr/hr)	
Interior – Door Frames	40	x	25	x (	72	-	52	) x	0.07	=	1,400	gr/hr
Loading Dock Door Cracks – ~~Windows~~	80	x	865	x (	146	-	52	) x	0.07	=	455,336	gr/hr
Ductwork		x		x (		-		) x		=		gr/hr
										Total	456,736	gr/hr

Door Openings

	Airflow Velocity (fpm)		Open Area (sq.ft.)		Air Density (lb/cu.ft.)		Time Open (min/hr)		Air Moisture Outside (gr/lb)		Air Moisture Inside (gr/lb)		Moisture Load (gr/hr)	
Interior – Door 1	50	x	100	x	0.07	x	4	x (	72	-	52	) =	28,000	gr/hr
Loading Dock – Door 2	968	x	100	x	0.07	x	1	x (	146	-	52	) =	636,944	gr/hr

Airlock Vestibule (Height ___ x Length ___ x Width ___) [Airlock Dimensions (ft)] x Air Density (lb/cu.ft.) ___ x Opening Frequency (openings/hr) ___ x [(Air Moisture Outside (gr/lb) ___ - Air Moisture Inside (gr/lb) ___) / 2] = ------- gr/hr

Total 664,944 gr/hr

Wall Openings

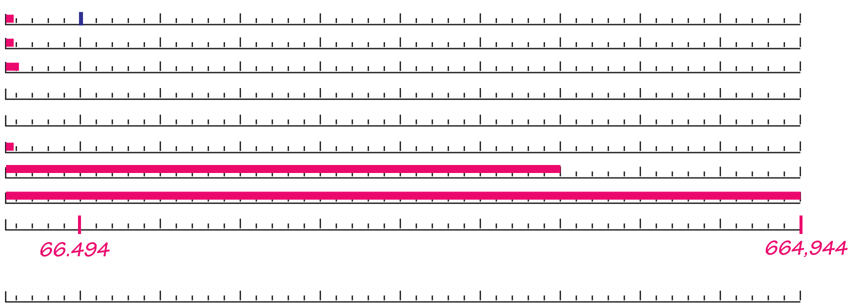
	Open Area (sq.ft.)		Air Entry Velocity (fpm)		Moisture Outside (gr/lb)		Moisture Inside (gr/lb)		Air Density (lb/cu.ft.)	Minutes Per Hour		Moisture Load (gr/hr)	
Conveyor Openings		x		x (		-		) x		x 60	=		gr/hr
Open Doorways and Holes		x		x (		-		) x		x 60	=		gr/hr
											Total	NONE	gr/hr

Fresh Air

	Fresh Air Flow Rate (cfm)		Moisture Outside (gr/lb)		Moisture Control Level (gr/lb)		Air Density (lb/cu.ft.)	Minutes Per Hour		Moisture Load (gr/hr)	
Net Fresh Air for Personnel, Exhaust Air Makeup And Room Pressurization		x (		-		) x		x 60	=	NONE	gr/hr

Summary

Permeation	1,398
Products	958
Personnel	11,000
Gas Flame	---
Wet Surfaces	---
Exterior Walls	1,797
Cracks	456,736
Door Openings	664,944
Wall Openings	---
Total Internal Moisture Load	1,136,819
Fresh Air	NONE



Glass lamination room

A manufacturer of automobile windows in Dallas, Texas is expanding a plant where glass sheets are laminated. The process uses thin films of polyvinyl butyral plastic as the adhesive between the glass sheets, and the plastic is hygroscopic. If the plastic has absorbed moisture, the water vaporizes during the autoclaving process, which traps bubbles within the laminated sheets. The windows would then be unusable.

The purpose of the project is to prevent the plastic from regaining moisture by maintaining the room at 20% relative humidity. The temperature is not critical, but should remain comfortable for workers, and temperature swings should be avoided so the large plastic sheets do not expand and contract. The designer decides on 70°F as the control temperature, which means that 20%rh represents an air moisture content of 22 gr/lb.

The room will be built inside an existing plant which is semi-cooled in the summer to temperatures that can reach 80°F and 50% relative humidity. The raw material enters the room from a warehouse that is not climate-controlled. Conditions will approach the ambient design extremes of 102°F and 146 gr/lb. To minimize moisture infiltration from the warehouse when product enters the room, the designer has arranged an air lock vestibule for the product door. Raw material deliveries vary somewhat, but seem to peak at a rate of 4 times per hour. The cardboard material packaging brings some moisture into the room. The packaging weight is 15 lbs for material consumed in an hour.

There are five workers in the room, and occasional traffic from supervisors and machine technicians. The designer allows for five permanent employees and three transient people in the room simultaneously. A door counter on an existing room suggests the personnel door will be opened 16 times each hour during production. The door measures 30 inches by 6 ft, 6 inches. Unfortunately, the space around the personnel door will not accommodate an air lock.

A conveyor carries the finished product out of the room through an opening in the wall that measures 2 ft. high by 4 ft. wide. The designer decides to equip the conveyor opening with a short tunnel as well as a plastic strip curtain at each end of the tunnel. There will also be some fresh air brought into the room for the personnel. This air will exit through the conveyor opening, further minimizing moist air infiltration.

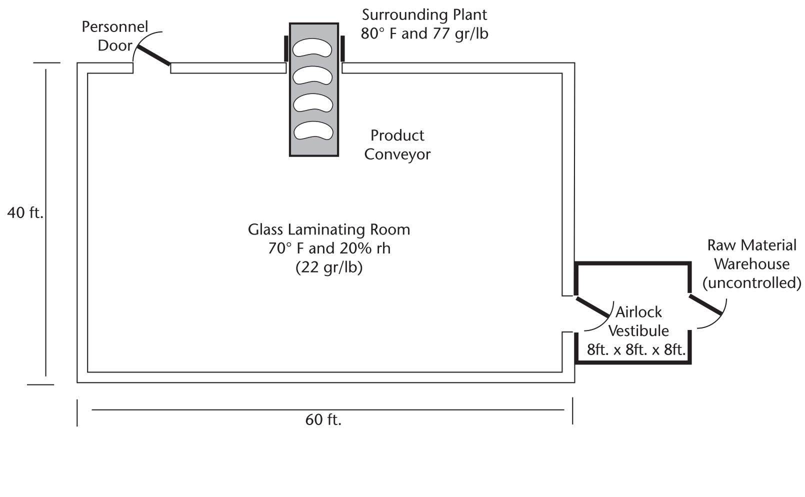
The room measures 60 ft long by 40 ft wide and 10 ft high. One of the four walls forms part of the exterior wall of the building, the other separates the process from the rest of the plant. The floor is an 8 inch thick concrete slab on grade and the walls are all concrete block with an interior finish that includes an aluminized plastic film vapor-retarder covered by plywood and vinyl wall covering. The ceiling of the room is the formed concrete “waffle-grid” slab that acts as the second floor of the building. It has been painted with two coats of an industrial latex vapor-retarder paint.

The engineer must make judgments concerning:

• *Relative humidity in the warehouse*
Packaging material for raw material from the warehouse will give off moisture, but what is the original moisture content? The design extremes of temperature and moisture translate to 47% relative humidity, but the engineer knows that as the dry bulb drops in the evening, the relative humidity rises. A simple humidity indicator placed in the warehouse suggests the typical relative humidity is 80% in the summer. The designer uses that figure for calculations, since even if it were wrong by 100%, it would not drastically affect the total load.

• *Volume of fresh air*
Local codes require 20 cfm per person fresh air ventilation, which would total 160 cfm for the eight people expected in the room. But the engineer wants to minimize moist air infiltration through doors, cracks and especially the conveyor opening. The opening measures 2 ft. high by 4 ft. wide, and the strip curtains cover the open area, except when product exits on the conveyor. The engineer wants to maintain 150 fpm outward air velocity. She assumes the product will force open the curtains so that one half of the opening will be exposed most of the time. So the airflow requirement will be 600 cfm, which will be more than adequate for personnel as well.

FIGURE 5.20



Moisture Load Calculation Sheet

Project Data

Project Name: *Laminating Room*

Location: *Dallas, TX*

Application: *Prevent moisture re-gain*

Calculations made by (name): *O.D. Colvin* Date: *1/02*

Approved by (name): *C. Munters* Date: *2/02*

Purpose of the project:

Prevent moisture re-gain in polyvinyl butyral plastic film by maintaining the laminating room at 70° F and 20% RH (22 gr/lb)

Design Conditions

	Weather Extremes Summer	Weather Extremes Winter	Ground water	Internal Conditions Room	Internal Conditions Building
Dry Bulb Temperature	102	18	67	70	80
Dew point	78	18	67	28	60
Humidity Ratio	146	14	100	22	77
Vapor pressure	0.48	0.10	0.68	0.15	0.52

Elevation *495* ft. above sea level—standard air density = *0.074* lb/cu.ft.

Permeation

	Surface Area (sq.ft.)		Permeance Factor (gr/hr/sq.ft.)		Larger Vapor Pressure (in.hg)		Smaller Vapor Pressure (in.hg)		Permeation Load (gr/hr)	
Wall 1	400	x	0.025	x (	0.520	-	0.15	) =	4	gr/hr
Wall 2	400	x	0.025	x (	0.520	-	0.15	) =	4	gr/hr
Wall 3	600	x	0.025	x (	0.980	-	0.15	) =	13	gr/hr
Wall 4	600	x	0.025	x (	0.520	-	0.15	) =	6	gr/hr
Floor	2400	x	0.45	x (	0.680	-	0.15	) =	572	gr/hr
Ceiling	2400	x	0.45	x (	0.520	-	0.15	) =	400	gr/hr
								Total	999	gr/hr

Products, Packaging & Clothing

	Item Entry Rate (lb/hr)		Original Moisture Content (lb/lb)		Final Moisture Content (lb/lb)		Grains Per Pound		Moisture Load (gr/hr)	
Item 1	15	x (	0.13	-	0.06	) x	7000	=	7350	gr/hr
Item 2		x (		-		) x	7000	=		gr/hr
Item 3		x (		-		) x	7000	=		gr/hr
Item 4		x (		-		) x	7000	=		gr/hr
								Total	7350	gr/hr

Personnel

	Number Of People		Moisture Load (gr/hr/person)		Moisture Load (gr/hr)	
Seated		x		=		gr/hr
Standing		x		=		gr/hr
Light work		x		=		gr/hr
Moderate work	5	x	5500	=	27,500	gr/hr
Room visitors	3	x	5500	=	16,500	gr/hr
				Total	44,000	gr/hr

Open Gas Flame

Gas Burning Rate (cu.ft./hr)		Water Vapor Generation (gr/cu.ft.)		Moisture Load (gr/hr)	
	x	(Typical Value 650 gr/cu.ft.)	=	NONE	gr/hr

Wet Surfaces

$$\left[\frac{\text{Wetted Surface Area (sq.ft.)} \times \text{Latent Heat Transfer Rate (Btu/sq.ft./in.hg.)} \times (\text{Water Surface Vapor Pressure (in.hg.)} - \text{Air Vapor Pressure (in.hg.)})}{\text{Latent Heat Of Vaporization At The Water Temperature (Btu/lb)}} \right] \times 7000 \text{ (Grains Per Pound)} = \text{Moisture Load (gr/hr)}$$

Moisture Load: *NONE* gr/hr

Exterior Walls

NONE — exterior wall is equipped with metal foil vapor retarder

	Surface Area (sq.ft.)		Air Infiltration Rate (cu.ft./hr/sq.ft.)		Moisture Outside (gr/lb)		Moisture Inside (gr/lb)		Air Density (lb/cu.ft.)		Moisture Load (gr/hr)	
Wall 1		x		x (		-		) x		=		gr/hr
Wall 2		x		x (		-		) x		=		gr/hr
									Total		NONE	gr/hr

Cracks

NONE — room will be supplied with excess air to minimize infiltration through cracks

	Crack Length or Component Area (ft or sq.ft.)		Air Infiltration Rate (cu.ft./hr/ft.)		Moisture Outside (gr/lb)		Moisture Inside (gr/lb)		Air Density (lb/cu.ft.)		Moisture Load (gr/hr)	
Door Frames		x		x (		-		) x		=		gr/hr
Windows		x		x (		-		) x		=		gr/hr
Ductwork		x		x (		-		) x		=		gr/hr
									Total		NONE	gr/hr

Door Openings

	Airflow Velocity (fpm)		Open Area (sq.ft.)		Air Density (lb/cu.ft.)		Time Open (min/hr)		Air Moisture Outside (gr/lb)		Air Moisture Inside (gr/lb)		Moisture Load (gr/hr)	
Interior – Door 1	50	x	13.75	x	0.074	x	8	x (	77	-	22	) =	22,385	gr/hr
Loading Dock – Door 2		x		x		x		x (		-		) =		gr/hr

Airlock Vestibule: (Height 8 x Length 8 x Width 8) (Airlock Dimensions (ft)) x Air Density (lb/cu.ft.) 0.074 x Opening Frequency (openings/hr) 4 x [(Air Moisture Outside (gr/lb) 146 - Air Moisture Inside (gr/lb) 22) / 2] = 9,396 gr/hr

Total 31,781 gr/hr

Wall Openings

	Open Area (sq.ft.)		Air Entry Velocity (fpm)		Moisture Outside (gr/lb)		Moisture Inside (gr/lb)		Air Density (lb/cu.ft.)	Minutes Per Hour		Moisture Load (gr/hr)	
Conveyor Openings		x		x (		-		) x		x 60	=		gr/hr
Open Doorways and Holes		x		x (		-		) x		x 60	=		gr/hr

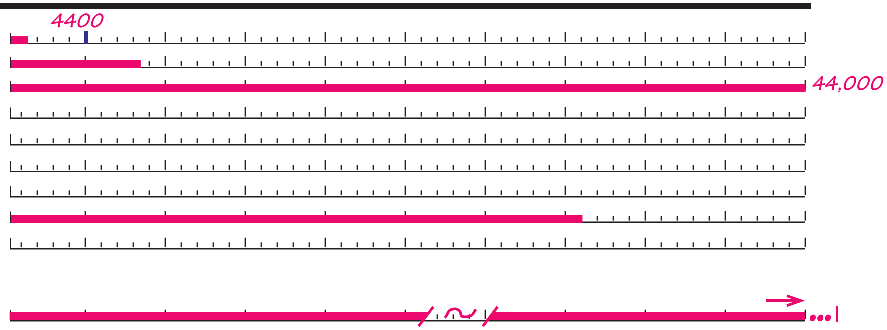
Assume that tunnel plus curtains plus 150 FPM air exit velocity will eliminate infiltration. Total NONE gr/hr

Fresh Air

	Fresh Air Flow Rate (cfm)		Moisture Outside (gr/lb)		Moisture Control Level (gr/lb)		Air Density (lb/cu.ft.)	Minutes Per Hour		Moisture Load (gr/hr)	
Net Fresh Air for Personnel, Exhaust Air Makeup And Room Pressurization	600	x (	146	-	22	) x	0.074	x 60	=	330,336	gr/hr

Summary

Permeation	999
Products	7,350
Personnel	44,000
Gas Flame	---
Wet Surfaces	---
Exterior Walls	---
Cracks	---
Door Openings	31,781
Wall Openings	---
Total Internal Moisture Load	84,130
Fresh Air	330,336



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Chapter 44 — Industrial drying systems

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Chapter 22 — Candies, nuts, dried fruits & vegetables (Table 1 - Optimum design air conditions, and Table 2 - Expected storage life for candy)
Chapter 26 — Commodity storage requirements (Table 2 - Storage requirements of perishable products)

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6

DESICCANT DEHUMIDIFIER PERFORMANCE

Operating Variables

Process Inlet Moisture
Process Inlet Temperature
Process Air Velocity
Reactivation Air Temperature
Reactivation Air Moisture
Reactivation Air Velocity
Amount of Desiccant
Desiccant Sorption Characteristics
Desiccant Performance Over Time

Desiccant dehumidifiers all function the same way — they remove water from air using the vapor pressure differences between the air and the desiccant surface to attract and release moisture. Chapter 3 explained how these vapor pressure differences are achieved in different types of desiccant dehumidifiers. Now we will discuss desiccant equipment performance in more depth.

A system designer generally selects equipment based on peak load requirements, but to fully satisfy all project requirements, it is useful to understand how performance changes when operating conditions are different than extreme peak design. This chapter discusses how and why dehumidifiers perform the way they do, and it explains the relationships between the key operating variables of the equipment. It also describes some implications of these relationships — which the designer may want to consider when engineering a desiccant system.

Operating Variables

There are eight key parameters which affect the performance of desiccant dehumidifiers. These include:

1. Process air moisture
2. Process air temperature
3. Process air velocity through the desiccant
4. Reactivation air temperature
5. Reactivation air moisture
6. Reactivation air velocity through the desiccant
7. Amount of desiccant presented to the reactivation and process airstreams
8. Desiccant sorption-desorption characteristics

The exact effect of each parameter depends on the type of dehumidifier in question. To simplify the discussion, we will examine a basic case of a rotary Honeycombe® dehumidifier, and then comment on differences with other types of equipment. In our basic example, we will assume the process air enters at "comfort" conditions — 70°F and 56 gr/lb, which is near 50% relative humidity.

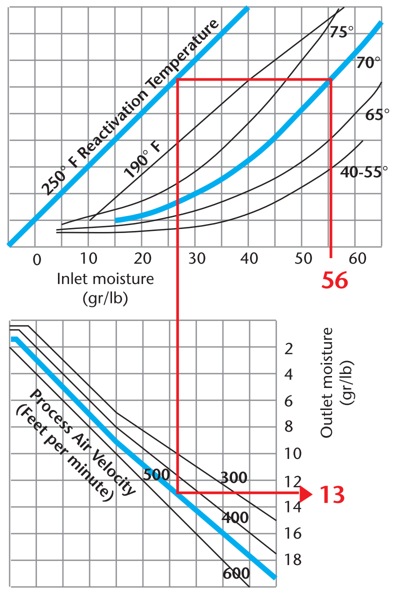


FIGURE 6.1

Honeycombe® Dehumidifier performance

Desiccant dehumidifiers remove moisture from one airstream, called the "process" air, and move it to another airstream, called the "reactivation" air. The amount of moisture moved depends on the variables shown in the curves above. High initial moisture in the process air, high reactivation air temperature and low process air velocity combine to remove the largest amount of moisture from the process air.

Figure 6.2 shows what happens to the air on each side of the dehumidifier. In our example, the process air leaves the dehumidifier warmer and drier than when it enters — 109°F and 13 gr/lb. On the reactivation side, a smaller air volume enters the dehumidifier from the weather. It passes through a heater and proceeds to the desiccant wheel. It heats the desiccant, which gives up moisture. The air is cooled as it absorbs the moisture from the desiccant, leaving the dehumidifier very moist, but much cooler than when it entered the desiccant wheel.

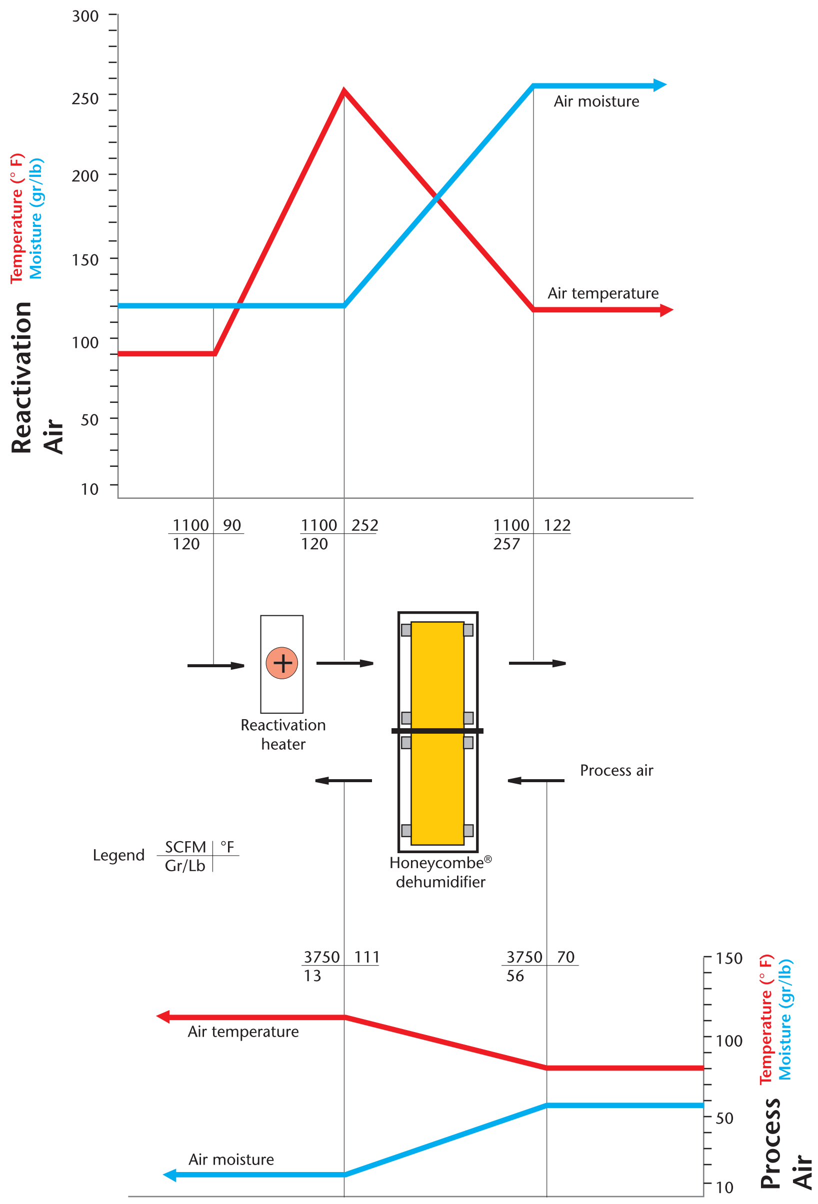
The operation of a desiccant dehumidifier has been compared to that of a "humidity pump". Just like a heat pump moves sensible heat from one airstream to another, so a desiccant dehumidifier moves latent heat — moisture — from one airstream to another. One can also think of a desiccant dehumidifier as a "moisture concentrator", since it removes moisture from the process airstream, and moves it to a much smaller reactivation airstream, which in effect concentrates the process moisture into the reactivation air.

In discussing dehumidifier performance, we must make one basic assumption at the start — the dehumidifier is operating at equilibrium. In other words, the total energy on the process side is balanced by the energy in reactivation. If the system is not in equilibrium — which can occur if airflows and temperatures are not controlled — a dehumidifier will not perform in an easily predictable manner. Assuming equilibrium, we will consider how unit performance changes as each key variable changes from the basic case.

FIGURE 6.2

Process and reactivation airflow temperature and humidity changes

As the process air is dehumidified, its temperature rises. Conversely, the reactivation air is being humidified, so its temperature drops as it picks up moisture.



1. Process inlet moisture

First consider the effect of changing the original moisture content of the air entering on the process side. The effect on outlet moisture is quite predictable. If the moisture level is lower to start, it will be lower than 13 gr/lb leaving. For example, air entering at 70°F and 35 gr/lb will leave at 6 gr/lb.

The temperature of the leaving air will be 101°F instead of 109°F. This is because the temperature rise of the process air is proportional to the amount of moisture removed from the air. In the basic example, the unit removed 42 gr/lb. In the second case, the unit removed only 29 gr/lb, so the air is heated less than in the basic case.

Conversely, if the entering moisture is higher than the base case, the air will leave the dehumidifier slightly more humid, but also warmer, since more moisture will have been removed. For example, if air enters at 70°F and 65 gr/lb, it will leave at 113°F and 17 gr/lb. This means 48 gr/lb have been removed from the air, which explains why the air is warmer than when only 42 gr/lb were removed.

A system designer can consider some implications of inlet moisture changes:

- If moisture entering is greater than expected, the process air will be warmer than expected, so if a constant temperature leaving the system is important, additional cooling will be necessary.
- If moisture is less than expected, the air will leave drier than expected, so if a constant moisture leaving the system is important, less air should be processed through the dehumidifier.

2. Process inlet temperature

As described in Chapter 2, desiccant surface vapor pressure depends on the temperature of the material as well as on its water content. So it is not surprising that desiccant performance is affected by the temperature of the incoming air. In our basic case, the air temperature is 70°F. If we lower that to 65°F, the moisture leaving process will be 9 gr/lb instead of 13 gr/lb. The moisture removal performance is improved because the desiccant is cooler, and therefore has a lower surface vapor pressure so it can attract more moisture than in the basic example. Conversely, if the inlet temperature is increased to 75°F, the air leaving process is not as dry — 17 gr/lb rather than 13 gr/lb.

The relationship of inlet temperature to performance is clear — when all other variables are constant, lower inlet temperatures enhance performance and higher temperatures reduce performance.

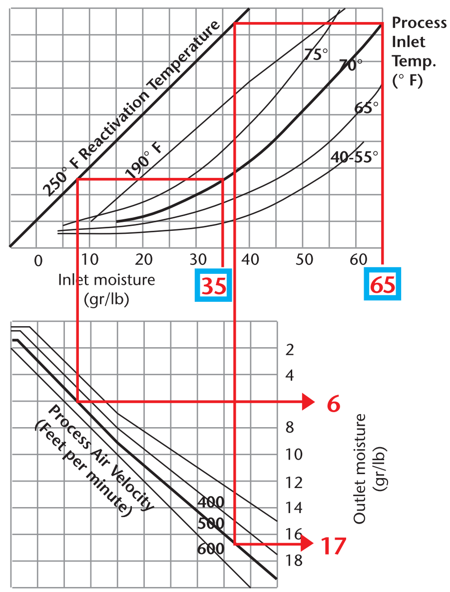


FIGURE 6.3

Changing process air moisture

The lower the moisture in — the lower the moisture level of the leaving process air.

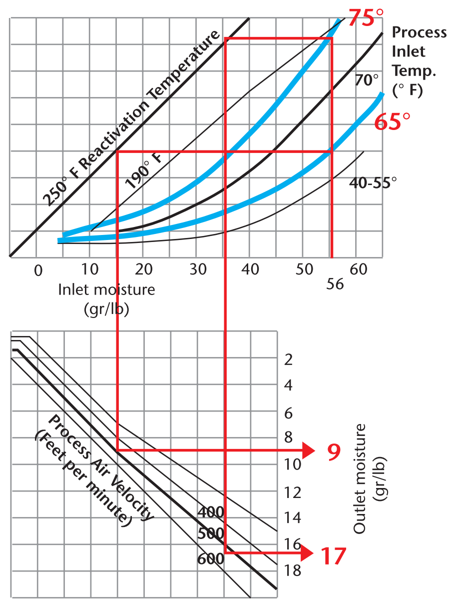


FIGURE 6.4

Changing process air temperature

Lower temperatures improve moisture removal.

For a system designer, this relationship has several implications, including:

• If high temperatures are expected, the engineer needs to confirm capacity at the highest expected process air temperature, perhaps selecting a desiccant which may be less temperature-sensitive. Or the designer can pre-cool the air to assure the desired outlet moisture.

• Cooler than expected process air conditions do not degrade performance. This is one reason why desiccant units rather than cooling-based dehumidifiers are often used in cold-storage areas and spaces which are not heated in the wintertime.

• The beneficial effect of lower process inlet temperatures is smaller at very low inlet moisture contents. For instance below 15 gr/lb, the effect is quite small, so pre-cooling the air from 65° to 55°F may only change the outlet moisture by 0.1 gr/lb — probably not enough to justify the investment in the cooling equipment.

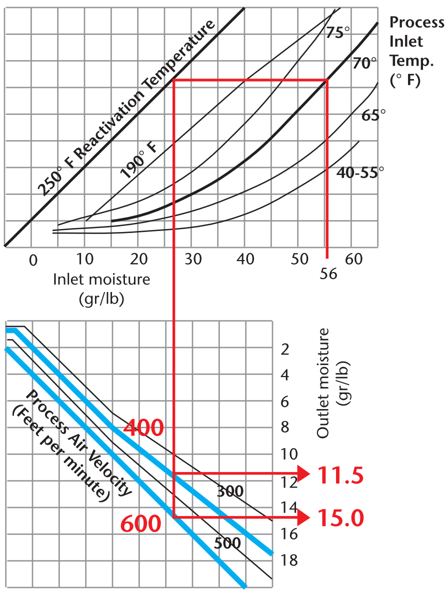


FIGURE 6.5

Changing process air velocity

Lowering process velocity allows more time for the air to contact the desiccant, so more moisture is removed. However, lower velocities mean larger equipment for a given airflow, so dehumidifiers are generally selected at the highest process air velocity that the application will allow.

3. Air velocity through the process side

The slower the air moves through the desiccant bed, the drier the outlet moisture will be. In our base example, the air travels through the bed at a velocity of 500 feet per minute, and leaves at a moisture condition of 13 gr/lb. If the velocity is reduced to 400 fpm, the leaving moisture will be lower — 11.5 gr/lb rather than 13. If the air velocity is increased to 600 fpm, the leaving air will be more moist — 15 gr/lb rather than 13. In this example, the difference seems minor, and indeed, dehumidifiers are generally selected at the highest velocity that will accomplish the moisture removal because high velocities mean smaller, less costly equipment.

However, when air must be delivered at very low conditions, it will be advantageous to use the lower velocity. For example, if the moisture control level in the space is 16 gr/lb, the difference between 11.5 and 13 gr/lb supplied to the space represents a 44% difference in the moisture removed from the room. 1000 scfm x 4.5 x [16 – 13] = 13,500 gr/hr. Compare this to the system's capacity at 13 gr/lb: 1000 scfm x 4.5 x [16 – 11.5] = 20,250 gr/hr.

For a designer some implications of changes in process air velocity include:

• If the outlet moisture must be very low, the process air velocity is quite critical, and the designer may want to install an airflow monitoring device and control system to avoid unplanned velocity changes.

• At high inlet moisture levels, the performance difference between low velocities and high velocities is rather small, so larger equipment may provide only small improvements in performance.

• If moisture removal rate—pounds removed per hour from the air stream—is more important than delivered air dew point, then high velocities will result in smaller, more economical equipment. For example, consider an industrial drying process in which moisture evaporates rapidly from a product, but the air supplied to the product should not be too dry. A small dehumidifier running high process air velocities will remove the load quickly, but with less risk of over-drying than a large unit processing the same amount of air.

4. Air temperature entering reactivation

In a rotary dehumidifier, the desiccant is heated by air entering reactivation. The hotter the desiccant, the more easily it gives up moisture, so the reactivation air temperature has a strong effect on performance. Essentially, the drier the desiccant can be made in reactivation, the more moisture it can absorb when it rotates into the process airstream.

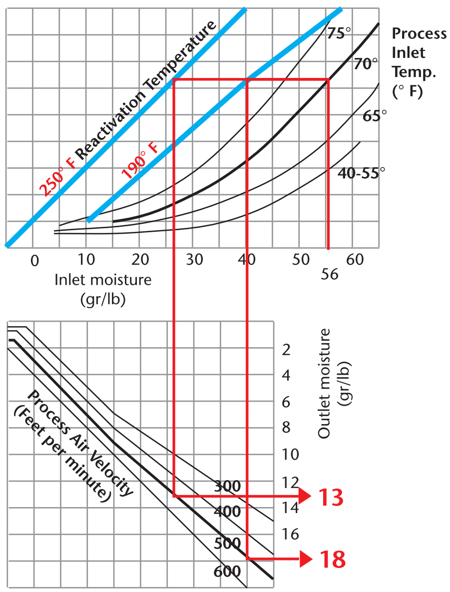


FIGURE 6.6

Changing reactivation air temperature

Reducing the temperature of the reactivation air generally reduces operating costs, since low-temperature energy is often less expensive than high-temperature energy. Sometimes, however, lowering reactivation temperature results in a larger dehumidifier, since the process air outlet moisture is higher than with high-temperature reactivation.

Our basic example assumes the air entering reactivation is at 250°F, which produces an outlet moisture on the process side of 13 gr/lb. If the reactivation temperature is reduced to 190°F, the process air outlet becomes more moist — it leaves the unit at 18 gr/lb instead of 13.

For the system designer, the effect of reactivation temperature suggests:

• If very dry outlet conditions are necessary, plan to use high reactivation temperatures.

• Desiccant dehumidifiers can make use of even very low temperature reactivation heat sources. When available, the designer should consider using economical energy sources like hot water from cogeneration, or steam condensate. However, the dehumidifier will need to be larger than one which uses a high-temperature reactivation energy source to produce the same outlet condition in the process air. In that situation the designer uses a large unit (lower process air velocities) to achieve the same effect as high reactivation temperatures. Also, larger reactivation airflows are necessary because the net energy added for reactivation (scfm x 1.08 x ΔT) must remain the same to keep the system in equilibrium. The moisture load from process has not changed. Since the temperature difference — ΔT — is lower, the airflow — scfm — must increase.

• Dehumidifiers are often selected at peak summer design conditions, when air entering the reactivation heaters is quite warm even before it is heated. During the winter, spring and fall, the air is much cooler. Since the reactivation air temperature affects dehumidifier performance, the designer should check unit capacity at the lower entering air temperature. It may be necessary to increase reactivation heater capacity for winter, spring and fall operation if the moisture load on the process side stays constant between summer and winter. This would be true of a dehumidifier that dried make-up air downstream of a condensing cooling coil. If the load decreases in winter — typical in a storage application with recirculating rather than fresh air — reactivation energy requirements are also lower, and heaters sized for summer peaks may be adequate even if the reactivation air starts out much colder than in summer.

5. Moisture of air entering reactivation

In this example, the desiccant is lithium chloride, which is not especially sensitive to moisture levels of the entering reactivation air. However, some mechanical concerns enter the discussion at this point, and other desiccants like molecular sieves have some sensitivity to reactivation inlet moisture.

The mechanical concern is air leakage between the moist air entering reactivation and the dry air leaving the process side of the unit. Any air leakage from reactivation to process will raise the moisture level in the process air considerably. For example, assume 500 cfm of process air normally leaves the unit at 1 gr/lb, but then 20 cfm of moist reactivation air at 120 gr/lb. leaks into the process airstream. With this additional moisture, the process air leaves the unit at 5.5 gr/lb — a considerable performance reduction.

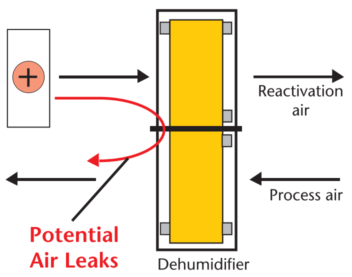


FIGURE 6.7

Reactivation-to-process air leakage
Rotary dehumidifiers must have good air seals between reactivation entering air and the dry process air leaving the unit. Any leakage at this point can raise the moisture level of the dry process air.

For system designers, this suggests that when using rotary bed, multiple vertical bed or rotary Honeycombe® desiccant units:

• The manufacturer of the dehumidifier should be consulted concerning the effect of any air leakage between reactivation and process in a given set of circumstances.

• When extremely dry outlet conditions are necessary, it may be advisable to mount the process air fan before rather than after the dehumidifier so that any leaks would move dry process air to reactivation rather than the reverse.

The performance of some other desiccants, notably molecular sieves and activated alumina, is considerably more sensitive to moisture in reactivation air. These desiccants are very useful for adsorbing moisture from warm airstreams. This means they do not lose performance on the process side when that air is warm, but conversely, such desiccants can still adsorb moisture even from warm reactivation air. If the desiccant partially fills with water in reactivation rather than empties, it will lose capacity for adsorption from the process air.

For the system designer, this suggests that when using solid adsorbents:

- If extremely dry process outlet conditions are necessary, it may be necessary to insure reactivation air is not extremely moist.
- If the reactivation air comes from the weather, relatively high reactivation temperatures will be necessary for best performance in summertime.

6. Velocity of air through reactivation

In a rotary dehumidifier, the reactivation air carries heat to the desiccant as well as carrying away moisture once it is released by the desiccant. More air (higher velocity) is necessary for heating than for carrying away moisture.

It is important to keep the reactivation airflow and temperature in proportion to the moisture load being absorbed by the desiccant on the process air side. If the moisture loading of the desiccant increases, more energy must be applied to the desiccant to insure complete reactivation and to keep the system in equilibrium.

The effect of *less* reactivation air is similar to having a lower reactivation temperature, because the net heat available to the desiccant is a function of airflow multiplied by the temperature difference between the air and the desiccant. In other words, high temperatures and high reactivation airflows deliver more heat to the desiccant, which means it can absorb more moisture in process because the material has been more completely dried in reactivation.

However, if the temperature entering reactivation stays constant and airflow is *increased* beyond the minimum necessary to carry the heat to the desiccant, the energy will simply be wasted. Unless there is an increase in the moisture to be removed from the desiccant, the reactivation air leaves the dehumidifier warmer than necessary, carrying heat off to the weather.

Implications for the system designer include:

• Rough filtration of reactivation air is always necessary. The designer should insure that changing or cleaning these filters is an easy and obvious task for maintenance personnel to accomplish. Otherwise, the filters will eventually clog, reducing airflow, which reduces performance.

• Reactivation airflow should be properly controlled to avoid higher than expected air velocities through the desiccant — fixed or modulating airflow dampers usually suffice.

• Mechanical considerations limit the practical number of choices for fans and heaters. However, the operational range necessary in reactivation may be very broad because of wide seasonal variations in process air moisture load. The designer must size for the maximum load, but recognize that the heater and fan selections will be less than optimum at part-load conditions unless the owner is willing to invest in modulating controls.

There are other implications for the designer that depend on the type of desiccant used in the unit, for instance:

• In the basic example, the dehumidifier uses lithium chloride. When it is a liquid, the desiccant has capacity to absorb hundreds of times its own weight in water vapor, which has many advantages. However, when lithium chloride is used in a rotating dehumidifier, it must be well-reactivated — dried out — otherwise it may absorb too much water and migrate through the support structure. So complete reactivation — which means maintaining heat and airflow to the reactivation sector — is especially important any time the machine is absorbing moisture on the process side.

• Granular desiccants like molecular sieves and silica gel are often used in packed tower, rotating tray or multiple vertical bed dehumidifiers. Higher than expected air velocities in either process or reactivation in these units can "fluidize" the desiccant — the air lifts it, and the desiccant bounces, which leads to air leakage through the bed and desiccant powdering. Both problems affect performance, so airflows should be controlled to avoid them.

• In liquid desiccant dehumidifiers, excessive air velocities can pull droplets of desiccant into the air. In reactivation, this can lead to corrosion of ductwork, and on the process side, the desiccant may not be useful to have in the conditioned space. As in all other types of dehumidifiers, airflow through liquid units should be controlled to avoid potential difficulties.

7. Amount of desiccant presented to the airstream

Along with other factors, the amount of moisture removed from the air depends on how much desiccant the air contacts as it moves through the dehumidifier — more desiccant means more moisture removed. In a rotary Honeycombe® dehumidifier, there are two ways to present more desiccant to the air — make the wheel deeper, or turn it faster. Either strategy will remove more moisture from the air, but both have an additional energy cost.

Increasing the depth of the wheel increases the resistance to air flowing through the unit. More surface area in contact with the air means more air friction. In the case of rotary tray, packed towers or multiple vertical beds, the desiccant is granular, so the flow is turbulent. This means the resistance to airflow essentially increases as the square of the air velocity. Liquid systems are similar — the airflow is turbulent through the conditioner and regenerator. Resistance — therefore fan energy — becomes very high very quickly in deep beds.

In the case of Honeycombe® dehumidifiers, the effect is somewhat less. Airflow through the straight passages is laminar rather than turbulent, but resistance still increases in proportion to wheel depth. So in all cases, increasing bed depth also increases fan energy costs.

Desiccant exposure to the airstream can also be increased by turning the wheel or bed faster between process and reactivation for solid desiccant units, or by pumping a desiccant solution more quickly between the conditioner and regenerator in liquid desiccant dehumidifiers. In many cases, this can increase unit capacity, but again — like deeper beds — the extra capacity increases energy consumption out of proportion to the extra water removed.

When the desiccant returns to the process air after reactivation, it brings heat from the reactivation process. The amount of heat is directly proportional to the mass of the desiccant and the temperature difference between the warm desiccant and the cooler process air. So when more desiccant cycles between process and reactivation, more heat is carried over into process, where energy will often have to be invested to cool the desiccant and the process air. Note that increasing bed depth also has the effect of moving more mass between process and reactivation, which will call for more cooling in process as well.

In general, manufacturers are keenly aware of these trade-offs, and units are designed to optimize the relationship between energy and capacity. The system designer does not usually have to determine bed depths or circulation rates — they will be established by the manufacturer.

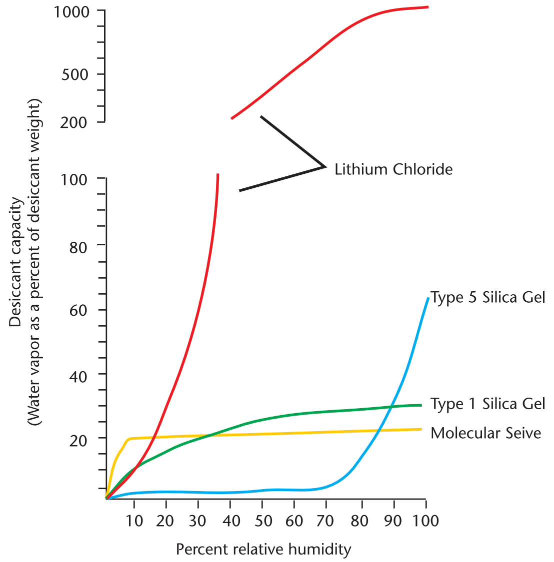
8. Desiccant sorption and desorption characteristics

Each desiccant has unique sorption characteristics which affect the performance of the dehumidifier. These characteristics can be shown graphically as a capacity *isotherm,* which indicates how moisture capacity changes as a function of relative humidity when both desiccant and air are at the same temperature.

Figure 6.8 shows the capacity of four desiccants at 77°F. Capacity is expressed as the weight of water retained as a percent of the dry weight of the material. For instance, at 20% relative humidity, Type 5 silica gel can hold 2.5% of its dry weight in water, Type 1 gel holds 15%, molecular sieve holds 20% and lithium chloride holds 35%. So if the dehumidification process was purely isothermal, and if the dehumidifier had to remove 50 lbs of water vapor from an airstream at 20% relative humidity, then the unit would have to contain at least the amounts of desiccant shown in figure 6.8.

Amounts of desiccant in dehumidifiers are not determined this way because the problem is considerably more complex. For instance, the quantities in the table would absorb the moisture if enough time were allowed for all the desiccant to come into equilibrium with the air — but that would take too long. Much more desiccant is necessary to remove a pound of water from a fast-moving airstream. Also, in an actual dehumidifier, the sorption process is not in any sense isothermal — the desiccant and air temperature and moisture content change constantly as air moves across the desiccant surface. But a desiccant isotherm does serve to illustrate why different desiccants are sometimes used in some applications and not in others.

For instance, molecular sieves are often used in drying air to very low dew points. Looking at the isotherm, it is clear that molecular sieves have a larger capacity than others when air is below 10%rh, so less desiccant will be necessary to remove a given weight of water.



	Capacity @ 20% rh (% dry weight)	Desiccant weight for 50 lbs. of water (lbs.)
Lithium Chloride	35%	**143**
Molecular Sieve	20%	**250**
Type 1 Silica Gel	15%	**333**
Type 5 Silica Gel	2.5%	**2000**

FIGURE 6.8
Desiccant capacity
At constant temperature, each desiccant has a fixed capacity to absorb moisture. Its capacity is a function of relative humidity. The table shows one consequence of that fact. To collect a fixed weight of water vapor, the weight of desiccant necessary varies according to the sorption characteristics of each material.

Also, some manufacturers use more than one desiccant in their equipment, and the isotherms can suggest the reason. For instance, two types of silica gel can be combined in the same unit — Type 1 providing capacity in the lower ranges, and Type 5 adsorbing larger amounts of water above 90%rh.

As with desiccant bed depth and cycling rate, desiccant selection and amount used in a given situation are generally accomplished by the manufacturer rather than by the system designer. Desiccant behavior is much more complex than a single isotherm, particularly when one considers that the desorption characteristics in reactivation are equally as important to unit performance as sorption from the process air. The behavior of combinations of desiccants are even more complex, and many manufacturers use such combinations to gain competitive advantages in particular applications.

Of more immediate concern to a system designer is the performance of a given desiccant or desiccant combination over time — what is the effect on performance of thermal cycling and airstream contaminants like particles and gases.

In general, desiccants are selected precisely because they are thermally durable and resistant to the effects of contamination, but each desiccant has some characteristics that have implications for the system designer:

• Solid adsorbents like silica gel and molecular sieves collect water on their surface and in narrow crevices of each particle. If the crevices are filled with dust, there will be less room for water, and performance will slowly diminish each year as particle loading increases.

• Organic vapors are often adsorbed by solid desiccants as efficiently as water, and indeed, some desiccant units are used specifically to remove such contamination from indoor air. However, some materials can polymerize when subjected to the high heat of reactivation, which may eventually clog the pores or modify desiccant surface characteristics.

• Liquid absorbents like lithium chloride and triethylene glycol collect water through a chemical reaction. The presence of other chemicals in the airstream can interfere with the reaction, or change the desiccant chemically. For instance, heavy airstream contamination by sulfur trioxide can — over a period of years — convert lithium chloride to lithium sulfate, which is not an effective desiccant.

• All desiccants, liquid or solid, can emit small particles into the air when air velocities are high, or as thermal cycling eventually fragments a desiccant. Again, the process takes years of continuous operation, and the amounts are very small — on the order of one part in 10,000 of the mass of the desiccant in a year.

The implications for the system designer are simple — always filter the air entering a desiccant dehumidifier. Consider filtering the air leaving the unit if minute amounts of particulate would be harmful downstream, and consult with the manufacturer if the designer expects large amounts of vapors other than water in the system.

7

SYSTEM DESIGN

System Types

Design Procedure

Passive Storage Examples

Archival Storage
Military Storage

Active Storage Example

Refrigerated Warehouse

Commercial HVAC Example

Supermarket

Industrial HVAC Example

Pharmaceutical Tableting Room

Product Drying Example

Candy Coating

Desiccant dehumidification systems vary in size, cost and complexity, but they generally fall into five basic types according to their application:

• Passive storage

Passive storage includes any storage environment where a controlled space is not frequently opened or closed, such as the storage of museum material, or the protection of military and industrial equipment.

• Active storage

These applications have more door activity, with higher intermittent temperature and humidity loads than static applications. Examples include steel storage warehouses, fertilizer warehouses and cold storage facilities.

• Commercial and institutional HVAC systems

Such systems have a need to combine comfort-level temperature control for people with low-level humidity control for products and processes. Supermarkets, hotels, medical facilities and laboratories are typical of this category. Such systems can have a comparatively low sensible heat load ratio (SHR), so dehumidification is a concern.

• Industrial HVAC systems

Many industrial processes benefit greatly from close control of temperature and humidity. Such applications typically have very tight tolerances and must function 24 hours, 7 days a week. These demanding requirements call for heavy-duty equipment with plenty of extra capacity. Applications include hundreds of high-value-added processes where product quality and high-speed production are the primary concerns.

• Product drying systems

In these applications, the dehumidification system is designed to remove moisture from materials rather than maintain a constant humidity level in a room or building. Pharmaceutical products, coated candy, photographic film, fish and plastic resins are all dried with desiccant systems. Water-damaged buildings and materials are also dried by service companies which bring free-standing dehumidifiers to an object or room which must be dried quickly. These applications are all similar in that maintaining high temperature and air velocity are not enough to accomplish the drying fast enough or without heat damage to the product, so dehumidification of the drying air becomes economically advantageous.

In all five of these system types, the design process is the same:

1	Define the purpose of the project
2	Establish control levels and tolerances
3	Calculate heat and moisture loads
4	Select, size and position components
5	Select and locate controls

For maximum efficiency, the designer must clearly understand the project purpose. This quickly sorts all the various design issues in order of their relative importance.

For example, if the purpose of a project is to prevent the growth of mold on corn, there is no need to maintain a strict tolerance of ±1% relative humidity throughout the storage bin. The only real concern is that the humidity does not exceed 60% and that condensation does not occur. The control system can be quite simple, and the equipment is small and economical.

On the other hand, if the purpose of the system is to prevent the corrosion of lithium, there is no point in trying to save money by using a control which has a tolerance of ±5% relative humidity. Above 2% relative humidity, lithium corrodes giving off hydrogen, which eventually explodes. A control with a tolerance larger than the control level itself cannot hope to start the dehumidification system in time to prevent that explosion. Understanding the project purpose helps the designer avoid unnecessary expense and false economy in both design time and equipment cost.

Additionally, the best dehumidification system designs are laid out with one thought firmly in mind — the system is dynamic, and no single component performs in isolation. Changes in any aspect of the system affect all other aspects. This is useful to remember because designers often concentrate on the mechanical aspects of the project, neglecting architectural and management issues that determine the sensible heat and moisture loads on the system.

With a clear understanding of the project purpose and a good understanding of how the system relates to its surroundings, the designer will understand the "big picture", and he can then proceed to lay out the system using the sequence illustrated by the examples which follow.

Any project needs a clearly stated purpose to avoid confusion and to guide design decisions. For example, several years ago, a military storage warehouse was built with a system specification requiring a maximum 40% relative humidity. Although the design accomplished that specification, the real purpose of the building was to prevent corrosion of munitions, and in that, the project failed.

The sheet steel roof cooled rapidly at night, causing water to condense on the ceiling even though the humidity was below 40% inside the building. The condensed water dripped on the munitions, which of course rusted badly. If the purpose of the project was more explicit, the designer might have looked beyond the 40% condition to other possible corrosion mechanisms such as condensation.

Passive storage - Museum example

The adjective "passive" is a relative term. In this case, we use it to distinguish between systems which have large transient loads and those which do not. For example, a photographic film archive is a relatively "passive" storage application if no one works in the room. In contrast, we consider an ice cream warehouse to be "active" because fork lift trucks move product in and out all day long, creating large transient heat and moisture loads.

Passive storage systems tend to be the simplest dehumidification systems. The loads are small and the tolerances are often wide. There is generally no benefit to complex central air handling systems unless the facility is quite large. System design often consists of placing the correct size dehumidifier in the room, ducting the reactivation air in and out and placing the humidistat in a location with good air circulation. In our first example, consider a small room in the basement of a museum that will be used to store motion pictures and still photographs.

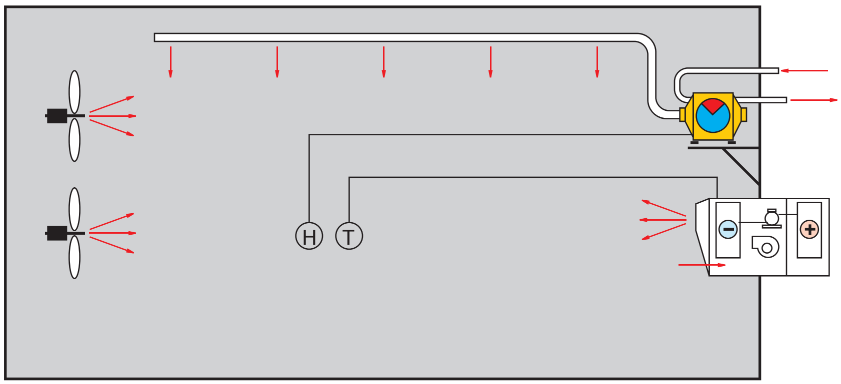


FIGURE 7.1
Dehumidified storage installations can be very simple and economical. The chief requirement is an enclosure which does not leak air.

In this example, the room is standard drywall construction, but it could just as easily be a frame of 2" x 4" studding covered with plastic film as long as all joints are sealed with aluminum foil vapor-retarder tape. The important point is that the vapor-retarder film must be continuous, so that moist air cannot leak into the enclosure. Floors must be sealed with epoxy floor paint, and ceilings must also have a continuous film of paint or vapor-retarder film. Ceiling tiles in a grid frame are not sufficient.

Step One — Define the purpose of the project

The purpose of the project is to prevent the fading of organic dyes used in the color film, and to prevent damage from mold formation on negatives and prints. The museum has recently acquired this significant collection with the stipulation that suitable archival conditions be provided. However, there is no room in the upstairs air-conditioned offices, and previous experience with storage in the cool basement was unfortunate. Nearly all the negatives had developed mold within a matter of months because of the high humidity. With a small budget, the curator is attempting to provide the best possible facility for the lowest possible cost.

Step Two — Establishing control levels and tolerances

The best information available from photo film manufacturers suggests the optimum condition for the storage of color film is 0°F and 35% relative humidity. At those conditions, color images should last indefinitely — perhaps longer than 500 years. (See references) Unfortunately, the budget will not accommodate such a facility. Also, when material is kept at such low temperatures, it must be brought into a special “dry warming room” for several hours before it can be exposed to normal room conditions. If it was brought into the working areas of the museum from a temperature of 0°F, it would condense water from the air like a cold beer can in summer, causing major damage to the images.

So, for a ten-fold reduction in the project cost, and to simplify access to the material, the curator determines that the humidity should be kept between 30 and 40% RH, and the temperature between 55 and 70°F. Current research suggests this will preserve the images for at least 50 years, and certainly will prevent any extreme damage.

Step Three — Calculate heat and moisture loads

Rather than trying to condition the whole basement with its ancient mortarless-fieldstone foundation and rammed-earth floor, the designer suggests the construction of a free-standing room. This improves both the security and cleanliness of the collection and simplifies environmental control.

The room measures 20' x 30' x 10' high. It has a single door, but no windows. The walls and ceiling are simple 3/8" drywall-on-stud construction, with taped and sanded joints and two coats of latex-based vapor retarder paint. The floor is poured concrete, painted with epoxy floor paint. In short — simple, residential-grade room construction. This is adequate, as long as close attention is paid to taping and sealing joints where walls meet the ceiling and floor. Since the room is in the basement, it is not exposed to the weather, so there is no need for more durable construction.

If it is important to maintain a higher minimum temperature in the room, a layer of 4" fiberglass insulation is a simple and cost-effective addition to the specifications. But in this case, cooler is better, and since the basement never gets below 55°F even in the winter, there is no need for either supplemental heat or insulation.

During the summer, the basement temperature can go to 80°F, so there will be a need for supplemental cooling. Heat load calculations determine that even without wall insulation, the load through the ceiling and walls will be only 400 BTUs per hour. The moisture load is also very small — it consists of the diffusion through the walls, ceiling and floor, some air leakage around the door frame, and the air exchange that results from people occasionally opening the door. In this case, the curator does not expect the door will be opened more than once or twice a day since the collection is not active.

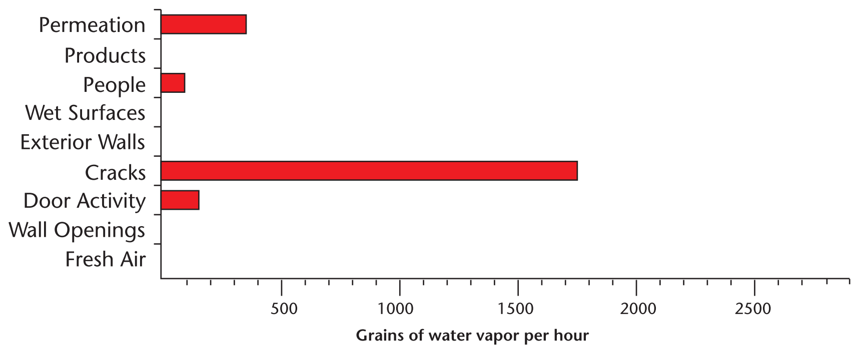


FIGURE 7.2
A detailed moisture load calculation performed as outlined in Chapter five shows that the hourly moisture load is quite small, totalling only 2092 grains per hour. This is less than 0.5 lb of water. The crack around the door is the largest load. A gasket would reduce this to almost zero.

The load from people and door openings is calculated by taking the single person and two openings per day and dividing the load these represent by eight hours to arrive at an average hourly load. A more conservative calculation would include the entire load in a single hour — which is the way the load actually occurs. However, since the stored product is not really sensitive to occasional minor excursions from the control point, the designer chooses the less conservative calculation method. This saves money, but the curator must recognize that the room may need some time to "pull down" following high door activity, since the dehumidifier will not be sized to remove the entire load immediately.

Step Four — Size the components to remove the loads

The cooling system must be able to remove 400 BTUs per hour from a room held at 70°F. This is well within the capacity of even the smallest window air conditioners sold for residential applications. Such a unit can be easily mounted through the room wall, with the condenser section rejecting heat from the room to the surrounding basement. However, the designer needs to be aware that these units often have an outside air intake for ventilation. Sometimes this can be sealed by a switch so air recirculates in the room, but other times the intake must be sealed with aluminum foil tape to prevent its becoming a leakage path for moisture.

The dehumidifier must be capable of removing at least 2100 grains of water vapor per hour from any condition between 55 and 70°F at 35% relative humidity. This is still easily within the capacity of a very small desiccant dehumidifier. Since both the dehumidifier and cooler have more capacity than required, they will operate intermittently, turning on when the control sensors call for cooler and drier conditions.

Step Five — Select the control system

In this case, one of the goals is to maintain a constant condition. But with such small, simple components, the control system is on-off — no smooth capacity modulation is easily achieved. As the units switch on, they make fast changes in the air because their capacity is large compared to the load.

This can be further aggravated by the congestion present in a typical store room. Stacks of documents and shelving restrict airflow, so unless steps are taken to circulate the air through the whole room, the units can over-cool and over-dry one part of the room while other parts are above the desired control point.

The solution to both of these problems need not be costly. The designer can call for simple, propeller-type circulation fans to hang on the walls or ceiling and to operate continuously. That way the cooler and dehumidifier can cool and dry all parts of the room rather than just the areas around the thermostat and humidistat. These controls should be located well out of the airstreams coming directly from the dehumidifier and air conditioner — otherwise they will turn off the machines before the entire space is dry and cool.

If the humidity and temperature are indicated through dials or recorded on charts, it is best to locate the instruments near the humidistat and thermostat. No two sensors read precisely the same, but co-locating the recorder and controller sensors minimizes confusion.

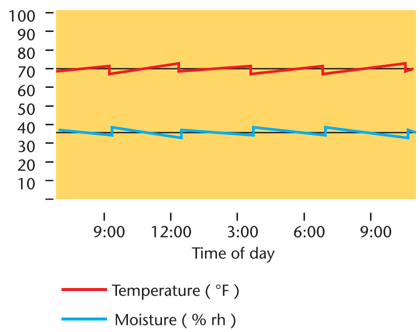


FIGURE 7.3
Simple on-off humidistats and thermostats are appropriate when the product can tolerate a range of conditions, and where the equipment is well matched to the loads.

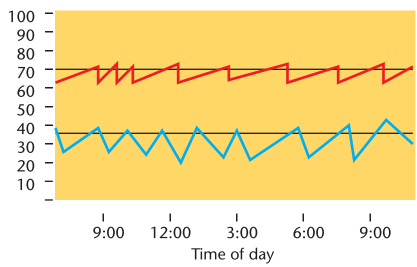


FIGURE 7.4
However, the natural tendency to oversize equipment must be kept in check, because with on-off control large equipment can cause big fluctuations in the room conditions.

Other considerations

Sometimes, air distribution and uniformity is a larger issue, for instance in long, narrow rooms or in rooms which are L-shaped or Z-shaped in plan. One approach, used very successfully in large steel warehouses, is to use several small dehumidifiers rather than one larger one, and place them throughout the room, each responding to a local humidistat.

Another approach is to replace the propeller fans with a simple ducted air distribution system which discharges air on one side of the room and collects it along the opposite wall, operating continuously like the propeller fans. The dehumidifier and cooler can remain free-standing as long as they discharge their treated air near the air collection points in the ductwork. In larger rooms with central ductwork, it is best to use several humidistats rather than just one. These are placed in different locations throughout the room and wired to the dehumidifier in parallel, so that any of them can start the unit.

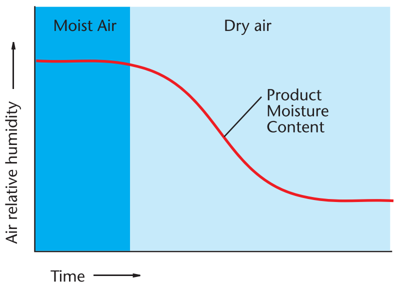


FIGURE 7.5
When a storage system starts up, it may have to dry out moist materials stored in the room. This extra moisture load may cause some delay in bringing the space to the specified humidity condition.

New buildings can be especially difficult to dry out, since rain and high humidity can saturate building materials during construction. The designer should be aware that these initial loads must be removed before the system will operate as planned.

When storage systems are started up, they sometimes run more often than expected, and with apparently little effect on the room humidity at first. Then slowly the room condition comes into specification. This will happen whenever there is a large amount of absorbed moisture in either the materials of construction or in the material stored in the room.

For example, the equilibrium moisture content of typical corrugated board is 14% at 80% relative humidity and 6% at 35% rh. So if papers and cartons are stored in a high humidity basement and then transferred to the archive in this example, they would eventually give up 8% of their weight to the air in the form of water vapor. A 30' x 20' room could easily hold 3,000 lbs of paper and film, which means the dehumidifier must remove 240 lbs of water from the material until it reaches equilibrium with the dry air in the archive. This may take several days, and in the meantime, the room humidity may be above specification.

Good air circulation through the room and around the shelves will speed the drying process, but in most cases, archival materials do not profit from rapid changes in moisture content. With sensitive materials that have a high initial moisture content, it may be best to gradually reduce the room humidity over a period of several days or even weeks, avoiding any product damage due to sudden drying.

References: Passive storage (Museum example)

McGuinness, John J. Jr. 1976. *Environmental control for historic properties.* Technology & conservation magazine, Winter 1976

Preservation and restoration of moving images and sound. 1986. Féderation Internationale des Archives du Film. Coudenberg 70 B-1000 Brussels, Belgium

Thompson, Garry. 1978 *The Museum Environment.* Butterworth Ltd. London & Boston. 19 Cummings Park Woburn, MA 01801

Harriman, Lewis G. III, Brundrett, G., Kittler, R., 2001. *Humidity Control Design Guide*. ASHRAE, 1791 Tullie Circle, NE, Atlanta, GA 30329 USA ISBN: 1-883413-98-2.

Passive Storage - Military Example

Another form of passive storage system is widely used to protect military and industrial hardware. In these cases, the equipment contains instruments, electronics and precision-machined surfaces which can be damaged by high humidity or condensation.

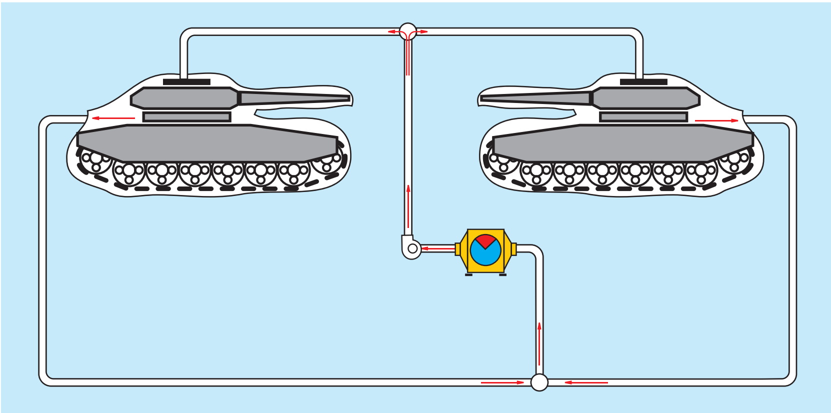


FIGURE 7.6
Industrial and military equipment is often more sensitive to humidity damage than to high or low temperatures. Although such equipment is often designed to be weathertight, it is not usually airtight.

In this case, the tanks have so many air leak points that it is more practical to enclose them in flexible fabric covers than to cover all the holes with foil tape. Air is then recirculated through these "bags" by a desiccant dehumidifier.

Step One — Define the purpose of the project

The purpose of this project is to prevent corrosion damage without the cost, time delay and difficulty of preserving the material with greases and other chemicals because the equipment needs to be ready for instant use. The project has come about because a military unit has had difficulty maintaining their tanks in fully mission-capable condition. Electronic failures have created major combat-readiness issues. The cost of instrument recalibration is very high, and more importantly, very few skilled technicians are available to accomplish recalibration. Although such equipment is normally stored in humidity-controlled warehouses, there has been a shortfall in the military construction budget, and no warehouse space is available. We will suppose a project officer has been assigned to design and install a desiccant wheel (DEW) dry air protection system.

Step Two — Establishing control levels and tolerances

Technical references dealing with atmospheric corrosion explain that ferrous metals corrode at slow and constant rates below 50% relative humidity, and at accelerating rates above that point. (See references) However, in this case, the more important concern is preventing corrosion of electrical contacts. These are made of copper alloys and are more sensitive to corrosion than plates of ferrous metal. The contact joints often connect two dissimilar metals or alloys, which makes the joint electrochemically more active. Also, it takes very little corrosion to cause problems because there is very little material in the joint. An increase in electrical resistance at a corroded contact causes problems even before the corrosion is visible to the naked eye.

Assuming normal atmospheric concentrations of sulfur dioxide and trioxide, technical references suggest the corrosion rate of copper is cut in half when the relative humidity moves from 80% to 60% relative humidity, and reduced to 10% of the normal rate when humidity is controlled at 30%. With that in mind, the project officer decides to set the control level at 35% relative humidity, with a tolerance of ±5% rh. Since the equipment is not temperature-sensitive, there is no need to heat or cool the tanks while in storage.

Step Three — Calculate moisture loads

The tanks will be kept in protective bags made of heavy-duty vinyl fabric. The project officer designs what is commonly known as a "closed", or "recirculated" dehumidification system. Air is drawn from the bags through the dehumidifier and supplied back to the bags around the tanks.

There is a small amount of air leakage at the seams, and around the joints where air distribution piping meets the fabric. There is also a minute amount of leakage through the vinyl fabric itself — it is not a perfect vapor barrier. Apart from this minor infiltration and transmission, there are no moisture loads. No doors are open, no moist materials flow in and out of the tanks and no people breathe out moisture inside the enclosures. Also, there is no need for fresh air.

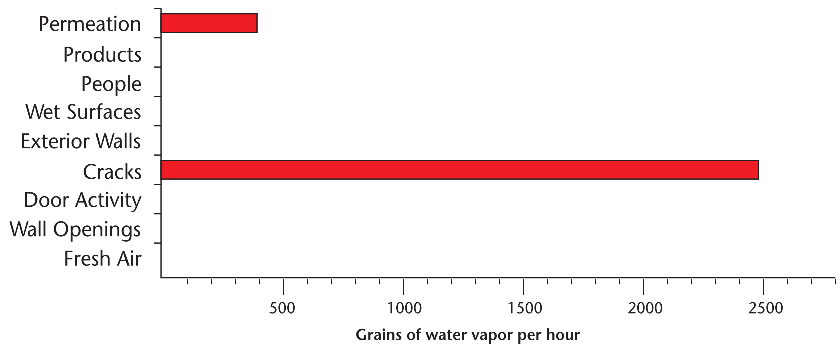


FIGURE 7.7
In a "closed" system, there is very little moisture load. The air recirculates through a closed circuit of vapor-tight covers and ductwork. There is a minute amount of vapor transmission through the plastic fabric, and a larger (although still very small) load which leaks through fabric seams, zippers and ductwork joints.

The tanks will be fueled and combat-ready in storage, so there is a question concerning the possibility of flammable fumes inside the enclosure. However, in this case, the tanks are fueled with diesel oil, so there is no hazard. In the case of more volatile fuels — for instance jet fuel for aircraft — the system could be arranged to bring fresh air from the weather to push the vapors out of the covers through relief valves. In that "open" type of system, the moisture load in the fresh air would be quite large in comparison to this small load in the closed system.

Step Four — Size the equipment to remove the loads

There is no cooling or heating requirement in this application, so that type of equipment is not necessary. The moisture load in the four tank enclosures totals less than one pound of water vapor per hour, so the dehumidifier could theoretically be quite small. However, there are two other issues in this case — having enough dry air capacity to reduce the humidity in response to rapid weather changes, and having enough fan capacity to force air through the system.

To visualize the first concern, assume that air in the system is at a condition of 80°F and 35% relative humidity. Therefore the specific humidity is 55 grains of water vapor per pound of air. Then the weather changes as a cold front moves through the area, and the air inside the enclosure cools to 40°. The original 55 grain moisture would now create a condition of 100% rh, so the dehumidifier must switch on and rapidly dry the enclosure.

Another form of this problem occurs when the weather temperature rises rapidly on a spring or fall morning. The air in the enclosure starts out at 40°F and 35% rh. The metal surfaces in the tank are also at 40°. Then the air around the tanks warms up to 80°. The humidistat is sensing the rh of the air, not the metal surface, so it does not turn on the dehumidifier until the relative humidity goes above 35%. Since the metal surfaces are still cold — particularly those inside the tank — they condense moisture because 35% rh at 80°F is above a 40° dew point.

To avoid both of these problems, the dehumidifier must have the capacity to dry all the air in the system very rapidly. The project officer decides the dehumidifier must be large enough to dry all the air within a half hour. The decision is based on the fact that the tanks are exposed to the weather. They are not inside a large warehouse — which would slow down the effect of weather changes. In fact, in sealed warehouses, the dehumidifier capacity specified by U.S. Army guidelines only needs to dry the air once every four hours.

So, since there are four tanks to dry, and each has an enclosed volume of roughly 510 cubic feet, the total air volume is 2040 cubic feet. The air in the ductwork totals 60 cubic feet. If that must be dried in half an

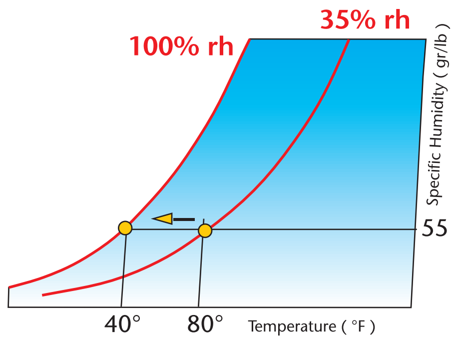


FIGURE 7.8

If air in the enclosure is 80°F and 55 gr/lb, it meets specification because at 80°, 55 grains represents 35% relative humidity. But if a weather front moves through — cooling the air to 40° — the dehumidifier must quickly dry the air, because 55 gr/lb is more than 100% relative humidity at 40°.

The dehumidifier must have the capacity to remove the basic moisture load, but if the temperature can change rapidly, the unit must have enough capacity to dry the enclosure as fast as the temperature change occurs. This can be expressed in air changes per unit time. In this case, the project engineer wants one air change within a half hour.

hour, the dehumidifier must process 70 cfm. (2100 cubic feet divided by 30 minutes equals 70 cubic feet per minute) The dehumidifier is still quite small.

Small dehumidifiers are not normally equipped with fans powerful enough to push through long runs of ductwork. The project officer consults Chapter 32 of the ASHRAE Handbook—Fundamentals, and concludes that if the air is flowing through 4" PVC pipe to all of the tanks, the pressure loss due to air friction will be less than 1.5 inches water column.

A small fan is added to the system downstream of the dehumidifier to blow the air into the tanks. The fan actually has a capacity of 2.0" WC at 70 cfm, so it is important to add some form of airflow control. If there is no control, the fan pulls 125 cfm, which is its capacity at 1.5" of air pressure.

In a system consisting of bags and ductwork, the moisture loads are very small. In old, large buildings, there is often a question as to how best to seal the building to make it vapor-tight. Sometimes a designer can be overly-concerned with hermeticity. In moderate humidity-control ranges — 40 to 60% rh — the issue is not as critical as at lower control conditions.

The largest moisture load comes from infiltration air, which leaks in through large holes in the wall. These can be fixed with plastic film stapled in place, with edges sealed with aluminum foil tape. It is not necessary to weld up all the cracks in the building. Air leaks are the problem, not vapor transmission.

One project in a steel warehouse was held up for two years because it would have cost $250,000 to caulk or replace leaking steel siding. The project finally went ahead with simple plywood covers over old ventilator louvres, and it only required a slightly larger dehumidifier to maintain control. The additional cost of the larger dehumidifier was less than 5% of the cost of replacing the steel siding.

In another case, the engineer simply built a frame of 2" x 4" studding covered with clear plastic sheeting inside an older, leaky building. The equipment was protected from the weather by the large building. Water vapor was controlled by a very small unit which dehumidified only the plastic "room".

Step Five — Select the control system

The system must control the amount of air pulled by the fan, and the humidity level inside the tanks.

Airflow control is very simple. Since there is no need for the airflow to vary, a sliding plate is placed on the inlet to the dehumidifier, and its position is fixed with a screw. Such sliding plates are standard air-conditioning duct fittings.

Humidity control is slightly more complex. A humidistat is placed inside each tank, and all four units are wired in parallel, so that a rise in humidity above 35% in any of the tanks will switch on the dehumidifier and booster fan. The important decision is the location of the humidistat within the tank.

Even though the tank is a relatively small space, the relative humidity can vary inside because of differences in air temperature, and proximity to small air leaks from the weather. The appropriate location for the humidistat is close to the electronics, since protecting them is the principal purpose of the project.

To be sure there will not be a problem with condensation on cold surfaces even when the relative humidity is 35%, the project officer invests in a second set of controls for each tank. A dew point controller is fastened to the chassis of the sensitive electronics. This device measures the relative humidity at the surface of the metal rather than the air. If there is any danger of condensation, the controller will turn on the system to dry out the tanks.

Other considerations

Air distribution

Since the goal of the project is to protect the instruments in the tank interior, it makes sense to carry the dry air to the tank interior rather than just into its cover. A flexible hose directs the dry air into the bottom of the tank, and moist air is forced out of the tank into the surrounding cover. From the cover, moist air is returned to the dehumidifier.

Tank security

Since the system is closed, the dehumidifier will not run very often. If it does turn on frequently, there is probably a leak in the enclosure, which could mean the tanks are insecure. Some installations are monitored so security personnel can be alerted if the dehumidifier runs for long periods.

References: Passive storage (Military example)

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Active Storage - Refrigerated Warehouse Example

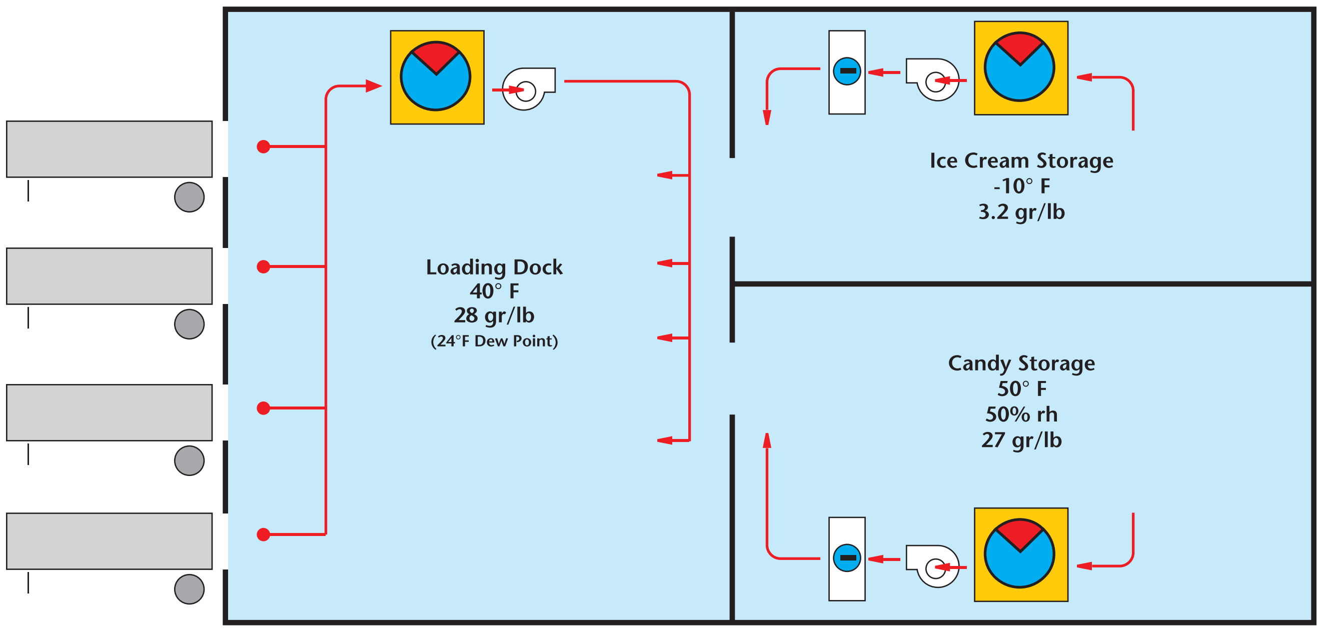
Active applications differ from passive storage in that moisture and heat loads are periodically very high. They are similar to passive storage because there is often no need for extreme precision, and the temperature requirement is either modest or dealt with by other building heating and cooling systems.

Since the moisture loads are quite different from those of a passive storage application, the system is designed in a different way as well. The designer's principal concern is to minimize loads without interfering with the main functions of the storage facility. For our example, we will examine a contract warehouse which stores ice cream and candy.

Step One — Define the purpose of the project

The purpose of the project is to eliminate wet floors and ice build-up on walls, floors and equipment. These are safety issues. Slips, falls and collisions are the leading causes of worker's compensation claims against refrigerated warehouses. The general manager intends to improve conditions by installing desiccant dehumidifiers to reduce the icing that causes such costly accidents.

FIGURE 7.9
Refrigerated warehouses are a typical dynamic storage application. The principal moisture loads come from doors opening as product moves in and out of the warehouse. The operations manager and designer can minimize system cost by reducing the time the truck doors stay open and by installing fast-opening doors across open passageways. These measures significantly reduce the size of the dehumidifier needed to remove the excess moisture load.



Step Two — Establish control levels and tolerances

The warehouse manager has no decision to make on temperature — the storage contracts are clear on this point. The manufacturers that rent warehouse space specify ice cream storage at -10°F and candy storage at 50°F. Humidity control levels require more thought. What conditions will meet the project goals — no condensation and no moisture absorption by cardboard?

Condensation and frost will not occur if the air dew point is below the inside surface temperature of the walls, floor and equipment. The manager obtains a low-cost infrared surface temperature thermometer, and measures each surface with these results:

FIGURE 7.10

Ice cream room

Conveyors..................................0°
Walls....................................... -8°F
Floors...................................... -8°F
Evaporator coil surface........... **-20°F**

Candy room

Walls.......................................52°F
Floors......................................52°F
Evaporator coil surface............**40°F**

Loading dock

Walls.......................................62°F
Floor near freezer ....35°F
Conveyors to ice cream............**0°F**

The lowest dew point in the building is the -20°F condition on the evaporator coils for the ice cream room. But the cooling system is equipped with an effective automatic defrost system, so while coil frosting is an energy consumer, it does not interfere with normal operation of the warehouse. The real concern is icing on conveyors and floors, which is a significant problem in normal operation because it must be removed by hand.

The manager decides to set the dew point control level in the ice cream room at -10°F. This will prevent icing on the conveyors, but some ice will still accumulate on the evaporator coils. This is an economic decision — it is less expensive to dehumidify to -10° than to -20°, and the ***rate*** of accumulation will still be much less at a -10° dew point than it has been with uncontrolled humidity. Also, the manager reasons that the system design can be arranged to place the driest air on the coldest surfaces. While the dew point in the room is controlled at -10°, the air supplied from the dehumidifier direct to the evaporators, floor and conveyor will be much lower, reducing frosting.

In the candy area, moisture absorption will occur in direct proportion to the relative humidity around the cartons. The carton manufacturer confirms that the cartons will retain 80% of their design strength if kept in a 50% relative humidity condition. Also, the candy manufacturer prefers a maximum 50% relative humidity to preserve product freshness. (See references) So the relative humidity is set at 50% in the candy storage area. Lower might be better, but would also be more expensive in terms of dehumidification equipment and operating cost.

The dew point at 50°F and 50% rh is 33°F. This is well below the surface temperature of both the evaporator coil and the floor — so there is no condensation in the candy room at this control level. This level will achieve three desirable goals: preventing softening of cartons, eliminating condensation on surfaces and even improving the operating efficiency of the refrigeration system by reducing coil frosting.

In the loading dock area, the manager chooses to control at a +33°F dew point. The logic is similar to that used in the ice cream room — it would be very costly to control the whole loading dock to a 0°F dew point just to eliminate all frost on the ice cream conveyor. A +33° dew point will eliminate the major problem of wet floors, and the air distribution system can be designed to put the driest air near the conveyor and the freezer door. This will reduce if not completely eliminate conveyor icing at less cost than controlling the whole dock at a low dew point.

Step Three — Calculate heat and moisture loads

Heat loads are handled quite well by the existing refrigeration system. One important point concerning heat loads is that the desiccant dehumidifiers convert latent heat to sensible heat as they dry the air. After the dehumidifiers are selected and their performance is determined, the refrigeration system capacity should be checked to ensure it can remove this sensible heat load.

The designer employed by the warehouse manager begins her moisture load calculation with the loading dock area. Since the dock is the most open to the weather, the loads will be highest at this point. High loads should be carefully examined since they lead to high costs.

The first moisture load calculation illustrates this point. The internal load is very large — 297 lbs of water vapor per hour. Removing this amount of moisture from a space held at 28 grains per lb would require over 20,000 cfm of dry air. That would mean a rather large dehumidifier, certainly larger than the budget allows.

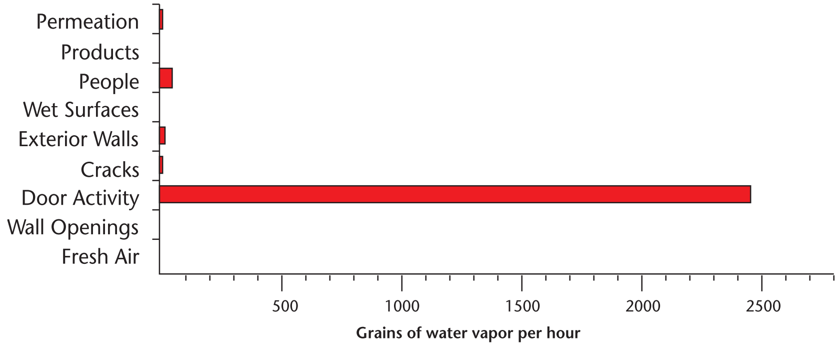


FIGURE 7.11
The moisture load from open doors is by far the largest component of the total load in an active warehouse. Reducing it will save money in both installation and operation.

Now the manager has two choices — reduce the load or raise the control level. Raising the control level defeats the purpose of the project. If the dew point is above the surface temperature of the floors and walls, water will continue to condense and cause problems. The manager and designer re-examine the loads.

The largest load — door openings — is the right place to start. The manager knows that an average of 15 trucks per hour pull out of the warehouse, leaving the loading doors uncovered. He also knows the workers do not quickly close the steel rolling doors over the openings. The designer's calculations assume an open door for three minutes each opening — two minutes after the truck pulls out and one minute when the truck pulls in. Changing this assumption to one minute per opening makes a big difference: the internal load drops from 297 lbs per hour to 44 lbs.

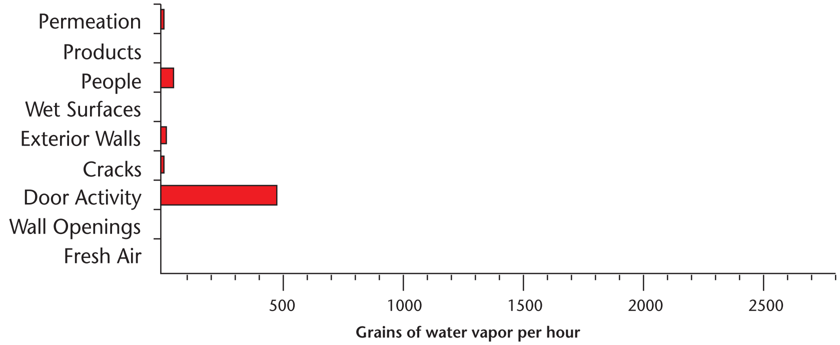


FIGURE 7.12
This shows the effect of closing the doors within one minute of each truck's departure rather than leaving the door open for three minutes — the moisture load is reduced to 25% of its original value.

Also note how little moisture is transmitted through the building walls, floor and ceiling. This load is insignificant compared to the effect of door management.

The volume of dry air necessary to remove the load drops from 20,000 cfm to 3,000 cfm, which means a very large reduction in system cost. This cost reduction is not free — it comes with the knowledge that supervisors will have to watch door openings closely if the system is to work properly. The manager still believes this is worth the effort, since the icing problem is solved at more reasonable cost.

Two other elements of the final moisture load for the dock area are worth discussing: the transmission load and fresh air load. The transmission load is very small — only 4,000 grains per hour, or less than one pound. This is not an error in calculation. Many designers become very concerned about the exact vapor transmission rates of various materials. Although the question is relevant, it is not normally of great consequence. For instance, in this case, the permeance factor of 8" concrete block with vapor barrier paint is 0.38 grains per hour per square foot per inch of mercury vapor pressure differential across the wall. If the wall had not been painted, the permeance would have been 2.4 — six times the permeance of painted block. However, the total permeance load only changes from 4,000 to 16,000 grains per hour, which is less than 5% of the total load. Door opening assumptions and air leakage guesses are far more important than knowing whether a particular paint passes 0.36 or 0.27 grains per hour per square foot.

The assumption concerning fresh air load is immensely more important because that air carries a great deal of moisture. In this case, local codes do not require continuous ventilation. If ventilation had been required, the system would have needed far more capacity because outdoor air is nearly always more humid than a 33°F dew point.

From the perspective of precision humidity control, these assumptions are still not very conservative design. If a moderate wind is blowing against the dock doors, moist air will be forced into the building. (A 15 mph wind travels at 1320 fpm. 5280 ft/mi. x 15 mph = 79,200 fph ÷ 60 min/hr = 1320 fpm) But the designer and manager reason that there is no critical process to be affected by moisture here — the purpose of the project is simply to reduce the problem, not to guarantee it can never occur under any circumstances. The benefit of this decision is a less expensive system — the cost of the decision is that the system will not eliminate the problem on humid, very windy days if doors are frequently open. Another manager and another designer might choose to use pre-dried ventilation air to pressurize the building for a more conservative design — or a more critical requirement.

Step Four — Size the components to remove the loads

The designer faces a decision — should the moisture load be removed by a single dehumidifier or by multiple units? A single large unit costs less to install but two smaller units can save energy when one shuts off at low load conditions. Given his tight construction budget, the manager decides on a single unit — accepting the possibility of higher operating cost in return for a lower initial cost.

So the loading dock will be served by a single system at one end of the dock, mounted on the roof of the building. The system will take air from above the dock doors, dry it and distribute the dry air down near the freezer doors. This arrangement sets up a circular air pattern that sweeps humid air away from the freezer and towards the return duct of the dehumidifier. How much air is needed to remove the load?

Looking at the performance curves in the appendix, it seems reasonable to assume that air could be dried from 28 gr/lb down to about 4.8 gr/lb. The total amount of 4.8 grain air needed to remove 44 lbs (308,000 gr) from a space held at 28 gr/lb would be:

EQUATION 7.1

$$cfm = \frac{315{,}000}{.075 \times 60 \times (27 - 7)}$$

Moisture load (gr/hr)

Difference in moisture between the airstream leaving the dehumidifier and the condition in the loading dock area (gr)

Minutes per hour

Density of air per ft³ (lb/ft³)

cfm = **2950**

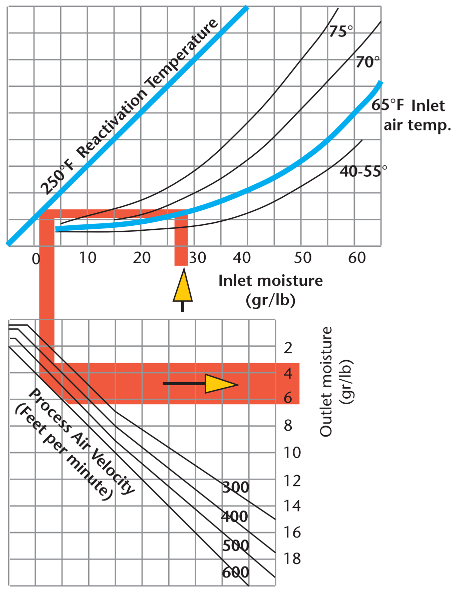


FIGURE 7.13
The volume of supply air necessary to remove the load from the dock area depends on how dry the air can be made — the drier the air, the less is necessary to remove the load.

From the performance charts in the Appendix, air entering a desiccant dehumidifier at 40°F and 28 gr/lb will leave at a moisture condition between 3.5 and 6.5 gr/lb depending on the reactivation temperature and the process air velocity through the desiccant bed. Selecting a unit for 25°F reactivation with a process face area of 7.5 sq.ft., 2950 cfm will travel through the desiccant wheel at 393 fpm. (2950 ÷ 7.5 = 393) The actual leaving condition at 393 FPM is 4.6 gr/lb — low enough to remove the load from the dock area, as can be seen in the following calculation:

EQUATION 7.2

$$\textit{Moisture removal} = .075 \times 60 \times (28 - 4.6) \times 2950$$

Airflow supplied to the dock (cfm)

Difference in moisture between the air-stream leaving the dehumidifier and the condition in the loading dock area (gr)

Minutes per hour

Density of air per ft³ (lb/ft³)

$$\textit{Moisture removal} = \textbf{310,635 gr/hr}$$

The system components are quite simple — a desiccant dehumidifier which pulls air from the dock, dries it and sends it back to the area near the freezer and candy storage doorways:

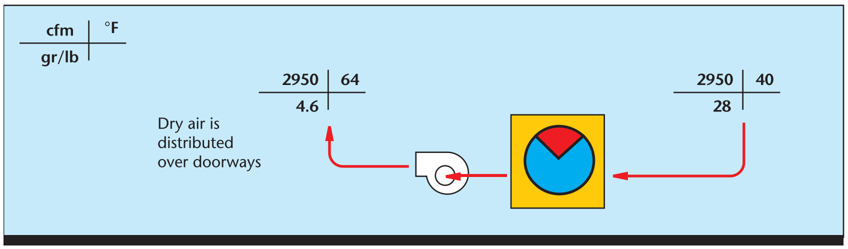


FIGURE 7.14
The dehumidifier takes air from above the loading doors, dries it and distributes it near the freezer doors. That's where the surface temperatures are lowest and where the dry air can do the most good by keeping ice from forming.

The example is arranged to come out neatly, which does not always happen in practice. If the dehumidifier with 7.5 sq.ft. of face area could only dry the air to 8 grains, the designer would have several possible courses of action:

• *Use a larger dehumidifier*
Not an especially attractive alternative in this case, because a larger unit would cost considerably more to purchase, install and operate.

• *Live with the performance of the 7.5 sq.ft. unit selected*
The dehumidifier is always sized at the peak load condition, which will only occur 0.4% of the time — less than 35 of the 8760 hours in a year.

The designer may decide to check moisture load and dehumidifier performance at a more typical summer condition of 80°F, 85 gr/lb, and if that is adequate, simply accept less than perfect performance at the extreme condition. But the manager — or a subsequent manager — may not be pleased with the designer's efforts or the supplier's equipment when the system fails to perform at peak conditions.

• *Reduce the internal load*
A full minute is still a long time to leave a door open — perhaps the doors can be closed ***before*** the trucks pull away from the dock.

Step Five — Select the control system
There are two issues to address in the control system — modulating the system components for maximum operating economy and eliminating the conveyor icing and wet floors.

The dehumidifier control can be an on-off device because there is no need for precision control within a narrow range of humidity values. Essentially, the drier the loading dock the better, since moisture not removed by dehumidifiers at the dock will have to be removed by the cooling coils and dehumidifiers in the ice cream and candy storage rooms. So there is arguably a case for no control at all — the dock dehumidifiers could run constantly and benefit the overall system.

Of course the dehumidifiers consume energy, and for several months the outside air dew point is below what is required to prevent frosting and condensation. Running the units during those dry hours would be wasteful. There are three choices for the dehumidifier control — relative humidity, dew point or surface condensation controllers. A relative humidity control set at 80% rh will start the dehumidifier when the air is above 28 gr/lb as long as the air temperature is 40°F. It is a low-cost device, and perfectly adequate for the project. Its accuracy is not likely to be better than ±2%, and if the air temperature is different than 60°, it will either under or over-dry the dock by a small amount.

A condensation sensor mounted on the floor or conveyor would be ideal, since the dehumidifiers would not run if moisture was not condensing. But as a practical matter, these devices are too fragile to mount on floors or conveyors. They are easily dislodged and crushed.

The designer chooses a dew point control, and sets it at 33°F. This is more accurate than a humidistat, and unlike a condensation sensor it can be mounted in the air rather than on the busy floor or conveyor.

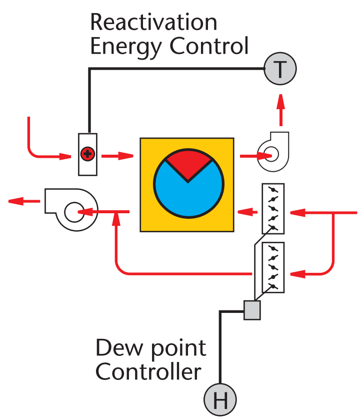


FIGURE 7.15
The dew point control sensor needs to be located so it will have good air circulation. The rear wall of the loading dock is a good spot unless pallets are frequently stacked in front of the sensor.

In almost all cases, it pays to install a reactivation energy modulation control, which saves large amounts of energy at part-load conditions.

Recognizing that the moisture load will vary widely throughout the year, the designer also installs a reactivation energy modulating control. This is a temperature controller, which senses the air temperature leaving the moist reactivation sector and varies the reactivation heater capacity to eliminate excess heat for reactivation. Different desiccants require different reactivation temperatures. In this case, the desiccant should have a minimum of 120°F leaving reactivation to ensure the wheel is fully dried.

Other considerations

In all cases, the air from the dehumidifier should be distributed where it will do the most good. In the dock area, air is distributed along the rear wall near the doors to the storage rooms. By placing dry air near those doors, the load on the dehumidifiers in the storage rooms is reduced, since the largest moisture load comes from door openings.

In many warehouses, there is no solid door between the dock and storage rooms because there is simply too much traffic across the opening. This large moisture load can be minimized by using plastic strips to reduce air infiltration. Strips are particularly important where openings are tall, and where there is a significant temperature differential across the opening. Temperature differences cause air density differences. The relatively heavy cool air falls to the floor, pushing out through the open doorway and creating a lower pressure higher up in the doorway, which pulls in warmer, moist air. Overlapping plastic strips covering a tall doorway can reduce moisture infiltration by blocking this local suction effect.

References: Active Storage

Babbitt, J.D. *The diffusion of water vapor through a slit in an impermeable membrane.* Canadian Journal of Research, Vol.19. 1941 pp 42-55.

Davis, W.J., *Moisture flowing up a moving air stream* 1972. Report on Special Project for engineering course EGR-192, University of Alabama, Birmingham, AL 35201 (Dr. Martin Crawford, Faculty Advisor)

ASHRAE Handbook of HVAC Systems and Applications 1987
Chapter 28 — Industrial Air Conditioning (Table 2 — Regain of Materials)

ASHRAE Handbook of Refrigeration 1986
Chapter 22 — Candies, nuts, dried fruits & vegetables (Table 1 – Optimum design air conditions, and Table 2 - Expected storage life for candy)
Chapter 25 — Refrigerated Warehouse Design
Chapter 26 — Commodity storage requirements (Table 2 – Storage requirements of perishable products)

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Latent loads in low humidity rooms. Final report of ASHRAE research project RP-252. Presented to ASHRAE Technical Committee 9.2 – January 1982. (Including notes on the final report by Wm. Griffiths – Midland-Ross Corporation) American Society of Heating, Air Conditioning and Refrigerating Engineers (ASHRAE) 1791 Tullie Circle, N.E., Atlanta, GA 30329

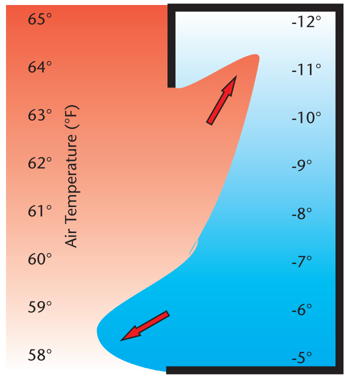
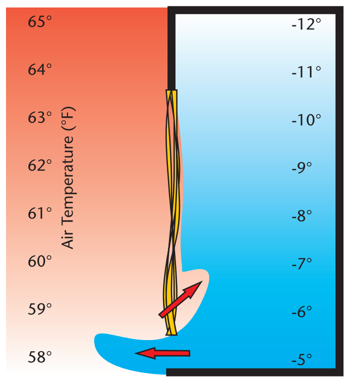


FIGURE 7.16

Tall, thin openings that separate areas of different temperatures leak a great deal of moisture because differences in air density force air through the opening as the cooler, heavier air sinks. Overlapping plastic strips do not eliminate this leakage, but they greatly reduce it by altering the shape of the opening. This reduces the pressure differential across the remaining open area.

FIGURE 7.17



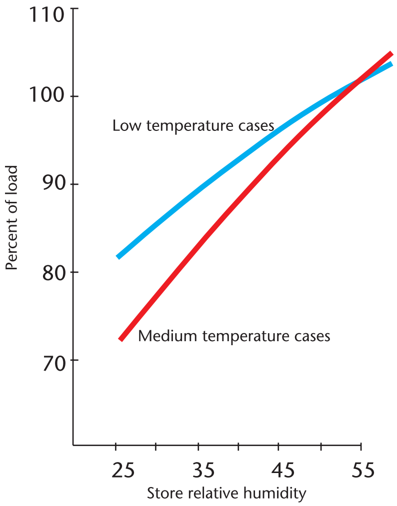


FIGURE 7.18
Commercial and institutional building HVAC systems often profit by the addition of a desiccant sub-system. It removes moisture from the air so the cooling system can operate more efficiently. In a supermarket, lowering the humidity reduces the load on the refrigerated display cases, which saves power costs.

Commercial HVAC - Supermarket Example

Commercial and institutional buildings such as supermarkets, hotels, research laboratories and hospitals often supplement their comfort air conditioning systems with desiccant dehumidifiers. Dehumidifiers allow an engineer to design an HVAC system which handles latent and sensible loads separately. Independent humidity control is useful when the sensible heat ratio (SHR) is comparatively low, which is to say the moisture load is a large proportion of the total heat load on the building. Dedicated dehumidifiers are also advantageous when the building moisture load peaks at a different time than the sensible heat load. Both of these criteria apply when the humidity control level is low, or when the building uses large amounts of ventilation air. We will consider the example of a new 50,000 square foot supermarket.

Step One — Define the purpose of the project

The purpose of the project is to improve the operating economics of the case refrigeration and comfort-conditioning systems by reducing the relative humidity in the store. Supermarkets are unique because their HVAC systems have a great deal of excess sensible capacity because of cool air spilling from refrigerated display cases, which creates a low sensible heat ratio — often below 70%. In addition, there is an economic benefit to maintaining a low dew point.

Step Two — Establishing control levels and tolerances

Since the purpose of the dehumidification system is to remove as much moisture as possible from the cooling systems, the moisture control level should be "as low as possible". Desiccant dehumidifiers can keep the humidity as low as necessary. Industrial spaces are often kept below 1%rh. In the case of a supermarket, maintaining 35 to 40%rh provides the optimum balance between equipment cost and operating cost reduction. (See references)

The engineer sets the temperature control level at 75°F for optimum comfort. Normal air conditioning applications would be set at 70 to 72°, but supermarket display cases spill so much excess cooling into the aisles that an average store temperature of 72° would be quite uncomfortable. The tolerances are fairly broad — ±3% relative humidity and ±3°F for temperature. There is no economic or comfort benefit to tighter control.

Step Three — Calculate heat and moisture loads

The principal heat loads come from heat transfer through the building wall, lights, windows, people, and from ovens in the bakery area and fryers at the deli counter. In fact, the heat loads and make-up air requirements in the deli area are so large — and so intermittent — that the engineer decides to install a separate system to serve that area alone. This relieves the central system of a high intermittent load, and makes both systems more efficient. The remaining sensible load in the rest of the store totals 750,000 Btu/h, but the spill-over cooling effect of the display cases will remove 430,000 Btu/h of sensible heat. So the remaining heat load is 320,000 Btu/h, which must be removed by the central air handling system.

Like the sensible load, the moisture load is partially offset by the dehumidifying effect of the refrigerated cases. Even though the desiccant dehumidifier will keep the store at 40% relative humidity, the case cooling systems will still condense water, since the evaporator temperatures run well below a 40°F dew point. Refrigerated cases are included in the load calculations because they actually remove moisture rather than add it to the store. That total is subtracted from the internal moisture load as shown in the graph.

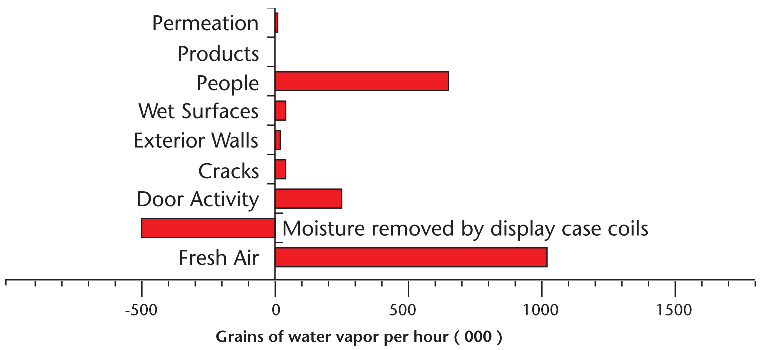


FIGURE 7.19
Refrigerated display cases act as dehumidifiers even when the humidity is controlled at 40%. This dehumidification effect must be accounted for in the moisture load calculation.

The largest element of the moisture load is water vapor in the fresh air, which depends upon the amount of ventilation air the system provides per person, and upon the number of people likely to be in the space. The local building code calls for a fresh air ventilation rate of 7 cfm per person. For 500 people in the store, the engineer designs for a total of 3,500 cfm outside air. Note, however, that ASHRAE Standard 62.1-2019 — *Ventilation for Acceptable Indoor Air Quality* — calls for a ventilation rate of 7.5 cfm per person in supermarkets. A different designer might choose to follow ASHRAE guidelines for a higher standard than required by older local codes. The lower ventilation rate is less costly, but the air quality will not be as high.

Air leakage through building joints is very difficult to determine because the structure is very large, and it is impossible to predict in advance how many cracks of what dimension will be left unfilled in the construction process. Working with the architect, the engineer finds that this particular contractor has an excellent reputation, and the architect has specified foam-sealed joints between different building materials, and around duct penetrations and door frames. So the engineer feels comfortable assuming tight construction, and uses the measured leakage rate of 0.1 cfm per square foot in a 25 mph wind described in Chapter 16 of the 2017 ASHRAE *Handbook of Fundamentals.* After the building is built, it would be possible to validate this leakage assumption through the use of blower-door equipment or the use of tracer-gas tests. These are described in Chapter 5 — Moisture load calculations.

Moisture from door openings presents two important questions — how much and how often? The engineer plans to supply a slight excess of air to minimize infiltration, so in theory, air should move out of the building rather than in whenever the doors are open. In fact, however, doors create local turbulence as they open, and small pressure differences between the bottom and top of each door pull moist air in through the top while dry, cool air pushes out through the bottom. The engineer decides to assume that 30% of the air volume trapped in the vestibule between the doors will enter the store each time the doors are opened.

How often the doors open is also difficult to estimate since the customer load varies so widely. If the engineer uses peak loading — almost 900 customers per hour — the system will be inefficient most of the year, overdrying and overcooling the space at unnecessary cost. A more appropriate calculation uses the average customer loading per hour throughout the week, which in this case is 150 customers, or 300 door openings per hour.

This is a compromise. During peak customer load conditions, the humidity may well be above 40% or even above 45% for short periods. Since people will not be uncomfortable, the engineer reasons that a few hours per year above the control condition will only slightly increase the annual operating cost of the refrigeration systems.

Wet surfaces also contribute moisture to the load. The produce department displays vegetables in open cases, where an automatic sprinkler system periodically sprays the merchandise with a fine water mist. The manufacturer of the spray system tells the engineer that water consumption is 10 lbs per hour for the 120 ft of produce display case which the store will be installing.

There is a certain amount of water vapor brought into the space in the form of moist cardboard packaging materials for merchandise. But most of the package moisture is absorbed into the outer corrugated cartons which leave the store before they can desorb moisture. By cross-checking the number of cartons per day with the store manager, the engineer determines the load from this source is small enough to be neglected.

Moisture from people is the second largest load after fresh air. Fortunately, this calculation is quite simple once the number of people is known. Table 1 of Chapter 18 of the 2017 ASHRAE *Handbook of Fundamentals* estimates the moisture load per person in retail stores at 200 Btu/h. Converting that load to grains per hour, the engineer multiplies by 6.6 for a total of 1320 gr/hr/person. (The latent heat of water is 1060 Btu's per lb, and there are 7000 grains of vapor per lb of water, so 1 Btu = 7000/1060, or 6.6 grains.)

The load from vapor transmission through walls, floor and ceiling is, like many other dehumidification applications, very small compared to loads from other sources. In this case, it is 10,000 grains per hour — less than the moisture added by any eight of the store's customers. With the loads defined, the engineer can proceed to the design of the system.

Step Four — Size the equipment to remove the loads

After removing the oven and fryer loads with a separate system, the remaining sensible heat load in the rest of the store is 320,000 Btu/h and the internal moisture load is 76 lbs per hour (80,560 Btu/h).

In other applications where low-level humidity control is not important, designers can quickly estimate cooling equipment size by dividing the combined latent and sensible loads by 12,000 Btu's per ton of cooling capacity and then estimate system airflow by multiplying the capacity by 400 cfm per ton. This procedure does not work for humidity-controlled supermarkets. It leads to systems with very large airflows and equipment that over-cools and under-dehumidifies the store.

To take advantage of the best characteristics of desiccants and cooling equipment, the engineer must think about temperature and moisture as independent variables. The desiccant sub-system will handle moisture and the air cooling system will control temperature. The sensible load and air circulation requirements will determine system airflow, and moisture need not enter into that calculation. The result will be smaller, more efficient systems than cooling alone can accomplish.

If sensible heat removal was the only concern, the system airflow could be set purely according to the allowable temperature rise across the store. For example, if the minimum allowable discharge temperature was 55°F (to avoid chilling people where the cool air is discharged), the supply air volume could be 14,815 cfm (cfm = 320,000 ÷ (20*1.08). However, in practice, engineers designing supermarkets do not use such small airflows because there might be too much thermal stratification in the wintertime. Typical desiccant-based designs use a minimum air volume of 1/2 cfm per square foot of store area, even though both sensible and moisture loads could be satisfied with less air. In this case, the engineer prefers to use the larger airflow, increasing the minimum supply air volume from 14,815 to 25,000 cfm.

Once the supply air volume and fresh air volume are established, the engineer can calculate the temperature and moisture conditions that will enter the central air handler. The dehumidifier is selected first because it will convert the latent heat in the air to sensible heat, which will affect the cooling system.

Summer design conditions for the ventilation air are 95°F, 120 gr/lb. Calculating the return air temperature from the store is a bit more involved because the air is taken from the aisles rather than from high in the store wall. This is done to remove excessively cold air from the aisles near refrigerated display cases, so it can be recirculated to areas where cooling is needed. This means the return temperature will not be the average temperature of the store — 75° — but somewhat lower. How much lower depends on how much air is returned from the cases and how much from the higher walls.

Empirical data from test stores has established that even when 100% of the return air is taken from the aisles, a maximum of 50% of the case cooling "credit" affects the temperature of the return air. The balance of the cooling effect stays in the store. To calculate this reduction in return air temperature, the engineer solves this equation:

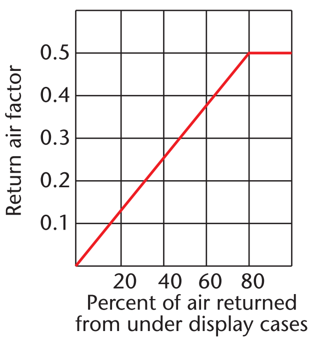


FIGURE 7.20
Supermarket systems are often designed to return air from the aisles underneath refrigerated display cases. The cold air from the cases cools the return air below the average store temperature. The "return air factor" expresses the percent of the case cooling effect captured by the return grills, which allows calculation of the return air temperature.

EQUATION 7.3

$$\textit{Temperature reduction} = \frac{.5 \times 430{,}000}{21{,}500 \times 1.08}$$

Return air factor (dimensionless)
Sensible cooling effect of the refrigerated cases (Btu/h)
Btu/h per cfm per degree F
Return airflow (cfm)

Temperature reduction = 9.2° F

Return air temperature = 75 − 9.2 = **65.8° F**

Since the return air conditions are now known, the engineer can calculate the blended temperature and moisture conditions of the air mixture, and move on to sizing the dehumidifier.

At low humidity levels, desiccant dehumidifiers dry air much more effectively than cooling systems. Therefore, *it is not necessary to dry all the supply air* to maintain conditions at 40% relative humidity. A small portion of air is dried very deeply, and mixed with the rest of the supply air to provide the level of dryness that will remove the load.

The first step is to determine how dry the entire supply air stream must be made. The following equation solves for the necessary difference in moisture between the air as it enters and as it later leaves the store.

EQUATION 7.4

$$\Delta\, grains = \frac{532{,}000}{25{,}000 \times .075 \times 60}$$

Internal moisture load (gr/hr)
Minutes per hour
Density of air (lb/ft³)
Supply airflow (cfm)

$$\Delta\, grains = 4.8$$

$$\textit{Required supply air moisture content} = 56 - 4.8 = \mathbf{51.2\ gr/lb}$$

Now the necessary delivered air condition is known, and the blended air condition entering the drier is calculated as follows:

EQUATION 7.5

Return air (cfm)
Temperature of return air (°F)
Airflow from the weather (cfm)
Weather air temperature (°F)

$$Temperature = \frac{(21{,}500 \times 65.8) + (3{,}500 \times 95)}{25{,}000}$$

Total supply airflow (cfm)

$$Temperature = \mathbf{69.8^\circ\ F}$$

$$Moisture = \frac{(21{,}500 \times 56) + (3{,}500 \times 120)}{25{,}000}$$

$$Moisture = \mathbf{65\ gr/lb}$$

The next question is how dry the air can be made by the drier. The performance curves in the Appendix show that if air enters at 70°F and 65 gr/lb, it can leave between 13 and 19 gr/lb depending on the

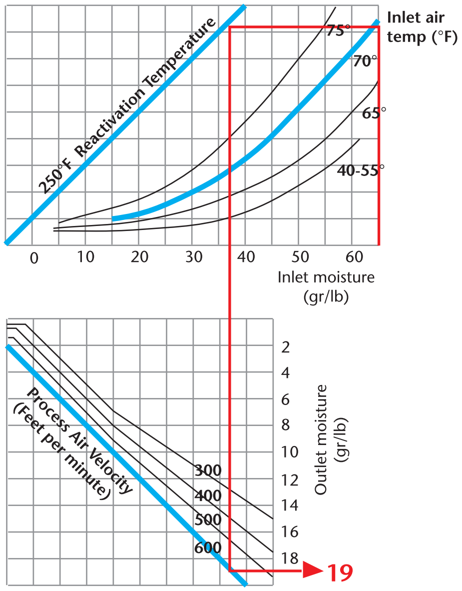


FIGURE 7.21
Desiccant dehumidifier performance depends on the inlet temperature and on moisture — in this case 65 gr/lb and 70°F — the temperature of the reactivation air and the air velocity through the desiccant wheel. High reactivation air temperatures and high air velocities will yield the lowest cost unit selection in this situation.

temperature of the reactivation air and on the speed of the air through the desiccant wheel. High process air velocity and high reactivation air temperature means the dehumidifier will be smaller — therefore less expensive to purchase.

To minimize installed cost, the engineer starts with the assumption that the dehumidifier will operate at 600 fpm process air velocity and 250°F reactivation air temperature, drying the process air to 19 gr/lb. Then the amount of 19 grain air that will bring 25,000 cfm at 65 gr/lb to the necessary 51 gr/lb is:

EQUATION 7.6

$$(cfm \times 19) + ((25{,}000 - cfm) \times 65) = 25{,}000 \times 51$$

Dry air volume (cfm)
Dry air moisture (gr/lb)
Airflow bypassing the drier (cfm)
Air moisture entering drier (gr/lb)
Desired supply moisture (gr/lb)
Total supply airflow (cfm)

$cfm =$ **7610**

A dehumidifier with 15 sq.ft. of process face area will handle 7610 cfm, and the air will be dried slightly lower than 19 grains. The process air velocity is 507 fpm (7610 ÷ 15 = 507) The air will leave at 17 grains, so the system will have more capacity than absolutely necessary.

EQUATION 7.7

$$\text{Total moisture removal} = .075 \times 60 \times (65 - 50.4) \times 25{,}000$$

Airflow supplied to the store (cfm)
Difference in moisture between the supply airstream and the moisture control level (gr)
Minutes per hour
Density of air per ft^3 (lb/ft^3)

$$\text{Total moisture removal} = \mathbf{1{,}642{,}500\ gr/hr}$$

Now that the performance of the dehumidifier is known, the engineer can go on to calculate the cooling that is necessary downstream of the drier after the air has been re-blended. The system flow diagram is now complete.

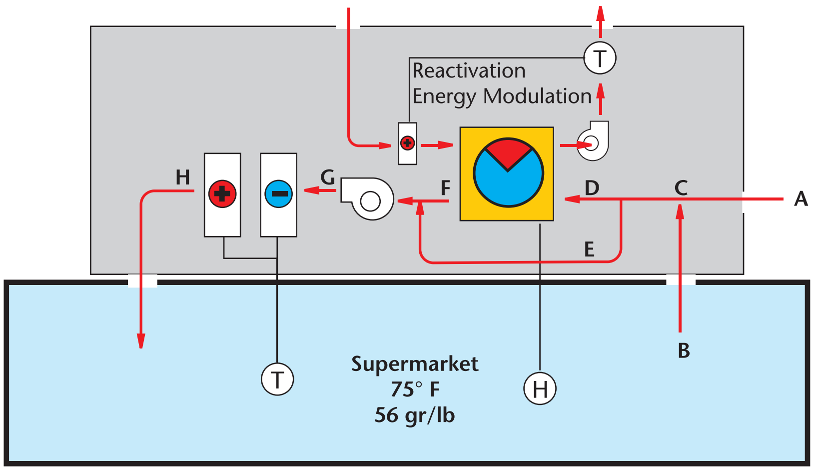


FIGURE 7.22

Location	Airflow (cfm)	Temp (° F)	Moisture (gr/lb)
A	3500	95	120
B	21500	65.8	56
C	25000	69.8	65
D	7610	69.8	65
E	17390	69.8	65
F	7610	112.4	17
G	25000	82.8	50.4
H	25000	55	50.4

Step Five — Select the control system

The temperature and moisture in the store must be controlled, and the dehumidifier and cooler must modulate in response to load changes due to weather and store operation. The main system fan will operate continuously, so air is constantly circulated throughout the space.

The humidity control can be a simple, low-cost, on-off humidistat. There is no need for precision control, and the control level is a comparatively high 40%rh. Some care should be taken, however, in locating the sensor. It should be at least ten feet off the floor to prevent accidental tampering. It should be located in the center of the store in a dry-goods aisle, because the lower aisle temperature created by refrigerated cases will disturb the reading. If the sensor is much higher than 10 feet off the floor, the air temperature will be above the control point, which will again affect the relative humidity reading. Using a dew point sensor avoids the problem of temperature interference with the humidity reading, although the cost of such a sensor-controller is somewhat higher than a simple humidistat.

The cooling system can also be controlled with an on-off thermostat. The engineer reasons that it should be located with the humidistat in the center of the store away from the frozen food so it will sense an average air temperature.

Both the cooler and dehumidifier need to modulate their energy consumption in response to load changes. Cooling can be modulated by equipping the system with compressor cylinder unloading valves.

These open in response to a change in pressure in the suction line from the evaporator coil, reducing the load on the compressor motor and therefore reducing energy consumption.

The dehumidifier reactivation energy is controlled by a temperature controller located in the air stream leaving the reactivation sector. As the moisture load on the unit decreases, less moisture passes out through the reactivation sector, making the air temperature rise. The controller then reduces the energy to the reactivation heater so that a constant 120°F is maintained in the air stream leaving reactivation.

An important control often overlooked by the HVAC system designer is the defrost system for the refrigerated cases. One of the major benefits of the lower humidity in the store is the reduction in frequency of defrost in the display case cooling coils. However, many defrost systems are controlled by a time-clock. In this system, the designer sets a timer according to an estimate of the defrost need rather than actually measuring the frost. Such timers do not allow the store owners to realize the full benefits of lower store humidity since the clock setting will always under or over estimate the need for defrost. A better alternative control uses a light-emitting diode (LED) and photo receptor to measure the frost thickness on the refrigerant evaporator tubes, initiating defrost only when frosting actually reaches an unacceptable level.

Other considerations

Supply air, which is cool and dry, should be distributed at the front of the store. Both the moisture and sensible heat loads are concentrated at the front of the store — where the people, windows and door openings are located. Comfort will be improved, and door opening loads will be minimized if supply air is first distributed at the front of the store over the registers.

Even temperature and humidity control depend on a close fit of the system to the loads. Whenever the heat and moisture loads in an isolated area are much larger or smaller than the loads in the rest of the store, it makes sense to install a separate system to handle that area. The example describes a separate system for the deli and bakery.

In an actual installation, a designer made the mistake of pulling a small amount of air from the central system to cool and heat the store manager's office. The manager was alternately roasted and frozen as the system responded to the needs of the store rather than to the conditions in the office.

The principal cost of operating a desiccant dehumidifier is the cost of energy for desiccant reactivation. In the case of a supermarket or other commercial applications, a key reason for installing a dehumidifier is to reduce operating cost, so the source and cost of reactivation energy is worth considering carefully. Supermarkets all have large amounts of waste heat from display case refrigeration systems which can be used to pre-heat the air before the main reactivation heater. Also, many utility companies provide exceptionally attractive rates for natural gas consumed in the summertime. The engineer can often cut the operational cost of the dehumidifier in half through imaginative use of these low-cost energy sources.

References: Supermarket Example

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Calton, Dean S. *Application of a Desiccant Cooling System to Supermarkets* ASHRAE Transactions 1985 V.91, Pt.1

Harriman, Lewis G. III, Brundrett, G., Kittler, R., 2001. *Humidity Control Design Guide*. ASHRAE, 1791 Tullie Circle, NE, Atlanta, GA 30329 USA ISBN: 1-883413-98-2.

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Industrial HVAC - Pharmaceutical Tableting

Industrial desiccant systems allow faster production of better quality material than would be possible at high moisture levels. Controlling humidity at low levels may mean the difference between profitable and unprofitable manufacturing. Therefore, engineers often think about industrial desiccant systems the same way they think about production equipment. Like any machine tool that runs 24 hours a day, 7 days a week, the dehumidification system must be built to be exceptionally reliable and economical to operate. We'll look at a typical application in the pharmaceutical industry.

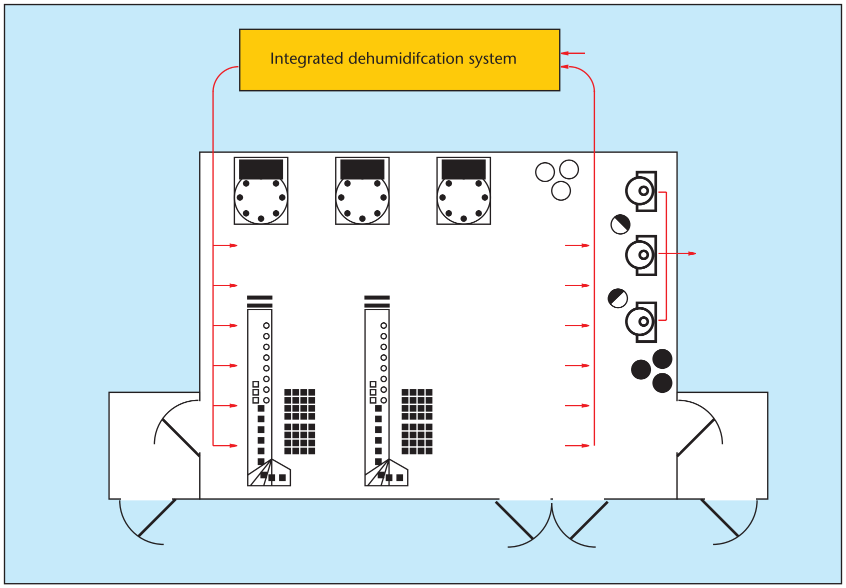


FIGURE 7.23
Pharmaceutical mixing, tableting and packaging operations are often carried out in low-humidity environments. In this example, we examine the decisions an engineer makes as he designs a system to control a room at a moisture level of 10% relative humidity.

Step One — Define the purpose of the project

The purpose of the project is to prevent effervescent powders and tablets from absorbing moisture from the air. Dry powders allow the tableting presses to operate at very high speeds, and dry tablets have a long shelf-life. The room measures 30' x 40' x 8'. The process has three steps: dry powders are mixed in large blenders, then the powder is compressed into tablets which are packaged and stacked onto pallets for shipment.

Step Two — Establish control levels and tolerances

Temperature uniformity is not critical to this process. There are no tight tolerances affected by thermal expansion or contraction. Worker comfort is the main concern, so the project engineer sets the control level at

70°F with a tolerance of ±3°. In fact, he knows many workers prefer a higher temperature in this type of low-humidity environment, but he designs for 70°F to ensure the system will have adequate capacity for all circumstances.

Humidity control is more critical. We will assume product development engineers have found that moisture absorption by the powders during manufacturing is the limiting factor on shelf-life. At this company, they have said "the drier the better, but we can live with 10% rh as a maximum because that eliminates 95% of the moisture absorption". So humidity is set at 10% rh. However, since the temperature will be allowed to swing three degrees above and below the set point, 10% relative humidity could represent between 9.7 and 11.8 gr/lb in terms of humidity ratio. When the humidity level is this low, such a difference can have major cost implications. To avoid confusion, the engineer specifies the dew point absolutely — a 13°F dew point (11 gr/lb).

Step Three — Calculate the heat and moisture loads

The room is built inside a larger building, so its walls, ceiling and floor are not exposed to the weather. Since the rest of the building is air-conditioned, there is almost no temperature difference across the room walls. The main sensible heat gains are from the motors which drive the mixers, tableting presses and packaging machinery. The lights and people also add heat. The total load is 41,000 Btu/h.

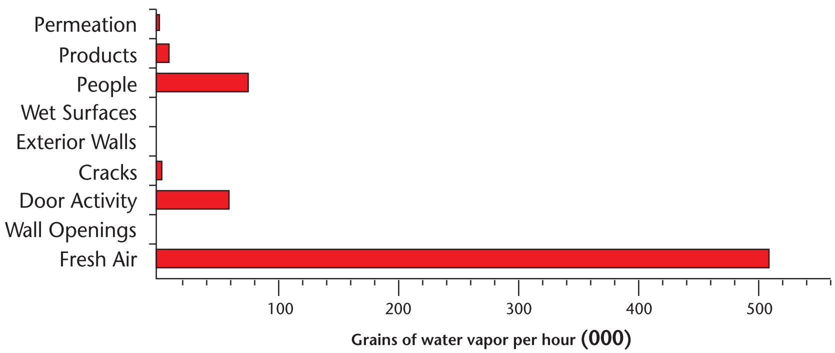


FIGURE 7.24
The exhaust hood over the mixers removes 500 cfm from the room. The air that replaces this exhaust carries a great deal of moisture, and is the largest load on the system.

The loads inside the room consist largely of worker's respired moisture and moisture which enters whenever a door is opened. Vapor transmission through walls, ceiling and floor is almost negligible.

The moisture load calculations show that moisture inside the room is a small proportion of the total load. The load chart shows the impact of fresh air on the room. Its huge relative size demands a closer examination.

The load in this example was calculated using the guidance in Chapter 5. However, equally qualified engineers may differ on calculation assumptions for the same set of circumstances.

For example, one senior engineer working in the pharmaceutical industry has found it very difficult to anticipate how the room will actually be used. Based on long experience, this engineer has found the uncertainty of most pre-construction moisture load calculations has been:

Fresh air volume	**±15%**
Duct leaks	**±100%**
Other air leaks	**±50%**
Dehumidifier leaks	**±50%**
# of Door openings	**±50%**
Wet surface area	**±25%**
Product moisture	**±50%**
# of occupants	**±25%**
Vapor transmission	**±50%**

The engineer finds that the load consists of 800 cfm. He puts the room under positive pressure using 300 cfm (to minimize leaks) then adds 500 cfm to make up for air exhausted from exhaust hood over the mixers. Unfortunately there is no way to reduce those flows. The exhaust is essential and the hood is already at the minimum size necessary to capture particles and solvents emitted from mixers. Although the fresh air flow cannot be reduced, the engineer makes a note to reduce the load by pre-cooling the air before it goes to the dehumidifier. Outside air enters the system at 91°F and with a moisture level of 147 gr/lb. These are ideal conditions for cooling-based dehumidification.

Pre-cooling the air to 50° will remove 60% of its moisture load, and the lower temperature of the air will improve performance of the desiccant unit as well. Very little can be done to reduce the load from people, but the engineer cuts the load from door openings in half by adding air-locked vestibules on the two personnel doors. The product door can be air-locked, but that would require quite a bit of space in the aisle outside the room. Since the door only opens once every hour, the reduction in moisture load would not be worth the cost and inconvenience of a large airlock.

The room is built from standard 3/4" sheetrock painted with industrial latex-based vapor-retarder wall paint. The joints are taped and the seams between walls and floor are caulked, so the load from vapor transmission is only 471 gr/hr — hardly enough to show on the chart.

The engineer calculates a small amount of leakage around the product door. Although he has specified a gasketed door frame, the engineer does not expect the gasketing to stay intact as pallets run over the frame each day.

The final total sensible heat load is 41,000 Btu/h and the internal moisture load is 105,000 gr/hr — 15 lbs of water vapor per hour.

Step Four — Select and size components

Sizing the components begins with the decision of how much air to supply to the room. This airflow must satisfy three criteria. It must be large enough to meet air circulation requirements, and large enough to remove both heat and moisture loads. The engineer must find the minimum airflow that will satisfy all three requirements.

In this case, the supply air quantity is established by company policy. Design practice at this particular company requires a minimum of 20 air turns per hour in production rooms. They find this enhances uniformity of the temperature and moisture throughout the space, improving comfort and production consistency. The room measures 30' x 40' x 8', a total of 9,600 ft^3. Circulating the air 20 times an hour will require 9,600 x 20, or 3,200 cubic feet per minute.

The minimum airflow necessary to remove the heat load depends on the allowable temperature differential between supply and return air. To make sure people are not chilled by the supply air, the engineer does not want to deliver air lower than 60°F. This equation shows the airflow necessary to remove the sensible load based on a 10 degree differential between supply and return.

EQUATION 7.8

$$cfm = \frac{41,000}{1.08 \times (70 - 60)}$$

41,000 — Internal sensible heat load (Btu/h)
(70 − 60) — Temperature difference between supply and return air (°F)
1.08 — Btu/h per cubic foot of air per minute

$$cfm = 3796$$

This means the system supply air must be at least 3800 cfm rather than the 3200 cfm which would satisfy the air circulation requirement.

The air needed to remove the moisture load likewise depends on the differential between the supply and return airstreams. But where the engineer can safely assume a cooling system can cool air by 10°F, he cannot automatically assume the dehumidifier can dry the air by a convenient amount. At high moisture levels, it may be easy to remove 30 gr/lb. At entering conditions of 15 gr/lb, it is difficult to remove more than 14 gr/lb.

So before he can determine the volume of air necessary to remove the moisture load, the engineer must determine how dry the supply air can be made. At this point, he knows three things:

- Supply air volume must be at least 3800 cfm to properly cool the room
- Return moisture condition is 11 gr/lb (70°F, 10% rh)
- Outside airflow is 800 cfm. It will be pre-cooled to 50°F, 54 gr/lb.

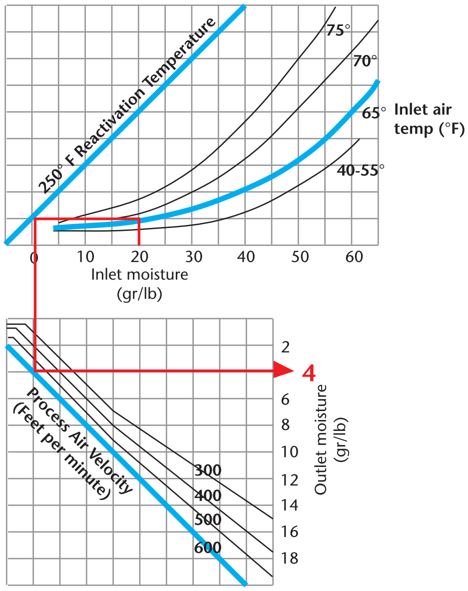


FIGURE 7.25
The condition leaving a desiccant dehumidifier depends on the entering conditions, on the reactivation air temperature and on the velocity of the process air through the desiccant wheel. Higher velocities generally mean smaller, more economical dehumidifiers, so the engineer checks performance at 600 fpm first. In this case, the capacity will be ample at 600 fpm.

With these facts, the engineer can estimate the performance of the desiccant dehumidifier to see how dry the supply air can be made. This will tell him if the airflow that removes the sensible heat will be large enough to remove the moisture load. Assuming a total airflow of 3800 cfm, he calculates the blended air conditions entering the dehumidifier:

EQUATION 7.9

Estimated return airflow (cfm)
Air temperature in the room (°F)
Airflow from the weather (cfm)
Weather air temperature after pre-cool (°F)

$$\textit{Temperature} = \frac{(3000 \times 70) + (800 \times 50)}{3800}$$

Total airflow to the dehumidifier (cfm)

$$\textit{Temperature} = \mathbf{66^\circ\ F}$$

$$\textit{Moisture} = \frac{(3000 \times 11) + (800 \times 54)}{3800}$$

$$\textit{Moisture} = \mathbf{20\ gr/lb}$$

Consulting the dehumidifier performance curves for a rotary lithium chloride dehumidifier, the engineer sees that air entering the dehumidifier at 72°F and 20 gr/lb can exit between 1.5 and 5 gr/lb, depending on the velocity of the air through the desiccant bed. A unit with 7.5 sq.ft. of free face area will process 3,800 cfm at a velocity of 506 ft/min. Air entering at 20 gr/lb leaves the unit at a moisture condition of 4.0 gr/lb. The engineer checks the capacity of this selection against the moisture load:

EQUATION 7.10

Proposed supply airflow (cfm)
Density of air (lbs/cuft/min)
Minutes per hour (min)

$$\textit{capacity} = 3{,}800 \times .075 \times 60 \times (11 - 4)$$

Moisture difference between supply and return airstreams (gr/lb)

$$\textit{capacity} = \mathbf{119{,}700\ gr/hr}$$

Since the load inside the room is 105,000 gr/hr, the airflow of 3,800 cfm delivered at a moisture condition of 4.0 gr/lb will be sufficient to remove the load, with 15% extra capacity in reserve.

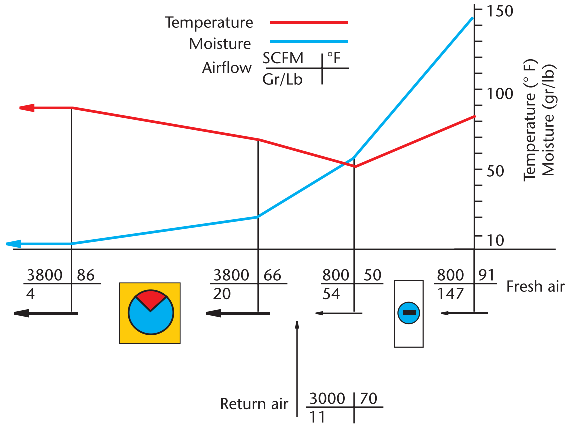


FIGURE 7.26
Pre-cooling fresh air to reduce its moisture load before it goes to the desiccant unit is almost always a very cost-effective decision. Cooling dehumidification works best at high temperatures and moisture contents. Desiccants perform more efficiently at lower temperatures and drier conditions. Combining the two technologies to dehumidify fresh air is less costly than using either one alone.

This graphic shows the changes in air temperature and moisture as it moves through the system.

Note that if the air from the weather had not been cooled and dehumidified, the blended air would have entered the dehumidifier at a condition of 74°F and 37 gr/lb. In that case, the air could only have been dried to 7 gr/lb. This would have required a dehumidifier twice the size of the unit selected, and much more than 3,800 cfm of supply air to remove the 105,000 gr/hr load. If pre-conditioned air is not available from a central system, it pays to dehumidify the weather air with a cooling system before it enters the desiccant dehumidifier.

The load in the room is treated separately from the load in the fresh air. The engineer concentrates on making the air dry enough to remove the internal load. He knows that the load in the ventilation air will be removed by the dehumidifier before it gets into the room. If the untreated weather air was somehow fed directly into the room, the dehumidifier would have to be much larger. For example, if the 500 cfm for the hood was not supplied by the system but simply allowed to leak into the room from the weather, it would add 306,000 gr/hr — almost quadrupling the room load. Therefore the system would require a dehumidifier almost four times the size of the 7.5 sq.ft. unit.

At this point, the engineer has established the supply airflow and the size of the dehumidifier. Now he can size the cooling downstream of the dehumidifier, knowing that the air must be delivered to the room at 60°F.

The air leaves the desiccant unit at 86°, so the post-cooling requirement is:

EQUATION 7.11

$$\textit{post} - \textit{cooling tonnage} = \frac{3{,}800 \times 1.08 \times (86 - 60)}{12{,}000}$$

$$\textit{post} - \textit{cooling tonnage} = \textbf{9 tons}$$

Proposed supply airflow (cfm)
Btu/h per cubic ft of air per min. per °F
Temperature difference between air entering and leaving the cooling coil (Btu/lb)
Btu/h per ton of air cooling capacity

The coil which treats the fresh air entering the system from the weather must dehumidify as well as cool, so the load at that point is considerably more per cfm than the post-cooling coil, which only removes sensible heat:

EQUATION 7.12

$$\textit{pre} - \textit{cooling tonnage} = \frac{800 \times 4.5 \times (45 - 20.2)}{12{,}000}$$

$$\textit{pre} - \textit{cooling tonnage} = \textbf{7.5 tons}$$

Fresh air from the weather (cfm)
Lbs. of air per cfm per hour
Enthalpy change between air entering and leaving the cooling coil (Btu/lb)
Btu/h per ton of air cooling capacity

Since cold air will be entering the system in winter, the engineer checks the winter temperature design extreme to size a heating coil. Air will enter from the weather at -5°F in the winter, which makes the temperature of the blended air 54°.

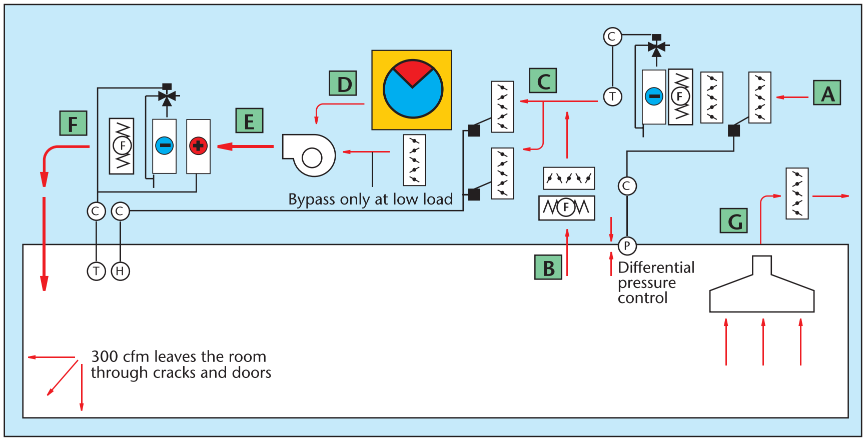
During the summer, the supply air temperature to the room is 60°F, but in the winter, that may not be high enough. Supplying air at 60° assumes there will be a full sensible heat load in the space — all the equipment is running, all the people are in the room and all the lights are on. In the winter , the system must be able to keep the room at 70° even if there is only a partial sensible load in the space. So the engineer sizes the heater to bring the supply air temperature from 54° to 70°.

EQUATION 7.13

$$\textit{winter heating} = 3{,}800 \times 1.08 \times (70 - 54)$$

$$\textit{winter heating} = \textbf{65,665 Btu/h}$$

Proposed supply airflow (cfm)
Btu/h per cubic foot per minute
Temperature difference between the blended air and the supply air to the room in winter (° F)



Location	Airflow (cfm)	Temp (°F)	Moisture (gr/lb)
A	800	91	147
B	3800	70	11
C	3800	66	20
D	3800	86	4
E	3800	86	4
F	3800	60	4
G	500	70	11

FIGURE 7.27
The system is arranged to remove moisture first, and adjust temperature second. The two variables are controlled independently because the sensible heat and moisture loads vary independently.

Fixed-position dampers are placed at key entry and exit points to set the basic system airflow. A variable-position damper at the fresh air intake then opens and closes automatically to allow enough air into the system to maintain a positive pressure inside the room.

The mixing operation is hooded and vented, but past experience with similar installations suggests there is still a need for filtration to remove particulate from the main air handling system. The engineer places roughing filters (35% removal efficiency) where air enters the system: one on the weather air and another at the return. These will provide as much protection as the system components require. Then he places a bank of high efficiency filters (95% removal efficiency) downstream of all system components. This will protect the process from particles which bypass the roughing filters and any particles generated inside the system itself.

Air regulation is another important aspect of system design. Airflows through the system will change with changes in weather temperature and pressure unless airflow can be fixed. The engineer places fixed-position, flow-setting dampers at the return from the room and the fresh air intake.

It is particularly important to control the exhaust airflow from the hood. If too much air is pulled from the room, untreated air will leak into the room, making it difficult to maintain moisture control. The designer specifies an exhaust hood with automatic airflow control to minimize the potential problem.

Step Five — Select the controls

The control system must modulate the capacity of each component so the system can respond to load changes. It must also ensure each component uses only the minimum energy necessary to meet those loads. In addition, since the room must be maintained under a positive air pressure, the airflow from the weather must be increased or decreased automatically as the room pressure changes throughout the day.

Room air pressure is maintained by a differential pressure controller which automatically adjusts a damper in the fresh air intake. As air pressure in the room rises above a set differential, the damper closes to reduce the incoming air. When the pressure falls — like when the product door is opened to let the pallet-jack through — the damper opens to allow more fresh air to pressurize the room.

The dehumidifier capacity is controlled by passing air around the unit when less dehumidification is necessary. The dehumidifier dries less air, and the blended condition of the supply air rises, so the system does not over-dry the room at part-load conditions. This works well as long as the resistance to airflow is the same through the bypass as through the dehumidifier, so the engineer places a fixed-position, flow-setting damper in the bypass. Equalizing the bypass and desiccant wheel resistance allows for smooth modulation of capacity.

The system controls room humidity by closing and opening the bypass dampers when a dew point controller signals a humidity change. That control does not, however, automatically reduce energy consumption as the load changes.

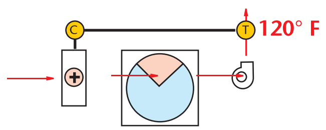


FIGURE 7.28
Reactivation energy can be reduced whenever there is less moisture being absorbed on the process side of the dehumidifier. If the temperature of air leaving reactivation is more than 120°F, the desiccant is dry. Power to the reactivation heater can be reduced.

When steam or hot water is used for reactivation, the heater temperature can remain constant, and the reactivation airflow is reduced by means of inlet vanes on the fan, or motor speed control. This method has the advantage of reducing fan energy as well as heater energy. It would not be appropriate for electric or gas reactivation since reduced airflow could lead to heater failure.

To minimize energy use, the engineer installs a temperature controller in the airstream leaving reactivation. The controller senses air temperature, reducing the energy of reactivation heaters whenever the temperature rises above set point. Air leaving reactivation at 120°F indicates that all the moisture absorbed on the process air side has been desorbed in reactivation. If the temperature falls, there is more moisture being absorbed in process, so more energy is required in reactivation. If the temperature rises, there is less moisture on the process side and the reactivation energy can be reduced.

The cooling coils are supplied with chilled water from a central system. The flow of chilled water through the fresh air coil is controlled by a three-way valve which responds to a controller located well downstream of the coil. The controller modulates the chilled water flow rate to maintain a constant 50°F air temperature leaving the coil. The post-cooling coil is controlled the same way, but the temperature sensor is located in the room, since its function is to maintain a constant room temperature.

The supply air heating coil is electric. It responds to the same sensor that controls the post-cooling coil, and its proportioning controller increases the power to the heating element whenever the room temperature drops below 68°F.

Other considerations

Weather design conditions

Early in a project, the engineer must decide what weather air temperature and moisture conditions to use as design points. In this case, the engineer chose the 1% summer design for both temperature and moisture. This means that over the course of the four summer months, the weather conditions will only exceed these values for 1% of the time — 26 hours. In this case, the decision was easy. Production is continuous, and if the system does not perform to specifications, the product loss is very costly. In less critical applications, the engineer may prefer to use less extreme conditions; perhaps the 2.5% conditions from this handbook, or even the much less extreme ASHRAE design data. But he should do so with the clear understanding that for many hours of the year, such a system cannot maintain 10%rh in the room.

Cooling system controls

In the example, we conveniently assumed the cooling would be accomplished with chilled water coils. If direct expansion refrigeration had to be used instead, the engineer would have had to decide whether to feed both coils from the same condensing unit, or install separate units for each coil. Installed cost will favor a single system, and control simplicity will favor separate systems.

The load on the post-cooling coil will be constant any time the outside air temperature is above 50°F — more than half the year. But the load on the pre-cooling coil will change as the weather changes. The engineer must provide for this load change in both the refrigerant piping and compressor control systems.

Energy recovery

Even with energy modulation, the cost to run this system will be high because it operates continuously. The engineer could further reduce this cost by recovering waste heat from either of two sources — refrigeration compressors and reactivation air exhaust. This waste heat can be used to warm the reactivation air before it enters the heaters.

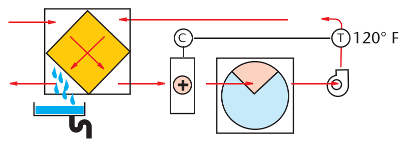


FIGURE 7.29
Operational cost can be reduced by recovering heat from the reactivation exhaust, and using it to pre-heat the air going to the heaters.

It is important to remember that the hot, wet reactivation air will condense gallons of water inside the heat exchanger, so the condensate must be properly drained.

When designing system ductwork, remember to allow for condensation inside the system. At one installation the duct that carried the moist air leaving reactivation was installed vertically without a drain and left without insulation. Water condensed in the duct and ran back to the unit, ruining the desiccant and reactivation fan.

In another case, there was no provision for draining the post-cooling coil since it does not condense moisture when the dehumidifier is running. However, when the system starts up after maintenance, the coil runs wet until the system reaches operating equilibrium. In that case, the coil was installed horizontally above the room supply grill. Water condensing on the post-cooling coil ran down into the room, to the considerable embarrassment of the designer and system supplier.

Waste heat from reactivation is substantial in this case. The unit uses 1,300 cfm of reactivation air, which leaves the unit at 120°F all year long. When the outside air temperature is 55°, an air-to-air heat exchanger could easily recover 60% of that temperature difference, or 91,200 Btu/h. It is important to remember, however, that a heat exchanger in the hot moist reactivation will condense a great deal of moisture — in this case several gallons per hour. The heat exchanger and ductwork must be arranged so the condensate runs to a drain and not back to the dehumidifier.

Another energy-saving option is to use waste heat from the refrigeration system to pre-heat air entering reactivation. This benefits the refrigeration system as well, since the coil transferring heat to reactivation increases the normal surface area available for refrigerant condensation.

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Product Drying - Candy Coating Example

Dehumidification systems which dry products often provide their owners with exceptionally large profits. In many cases, using dehumidified air to dry products can double the processing speed of a production line, and increase product quality as well. Perhaps even more than industrial building room control systems, a product drying dehumidification system is like a machine tool — it can pay back its installation cost in months or even weeks rather than decades. For our example, we will look at a “bin drying” system for covering candy pieces with a hard, shiny coating.

Desiccant dehumidifiers are used to dry products when there is a benefit from either a very low dew point, or a very low drying temperature, or both.

For example, the emulsions in photographic film are “set” by cold air and chilled rolls at temperatures between 0°F and 20°. Dry, cold air is necessary to set the emulsion, to remove some of its water and to insure no condensation occurs on the cold film surface.

In contrast, PET plastic resin is very hygroscopic. 300°F air with a dew point of -40 (0.5 gr/lb) is used to remove the last vestiges of moisture from the resin before processing. Film manufacturing is continuous, and plastic resin drying is typically done in batches. Both processes are possible without dry air, but the speed is at least doubled by using dehumidifiers.

Step One — Define the purpose of the project

The purpose of the project is to reduce humidity of the drying air, so the fast production rates of winter can be maintained through summer, spring and fall. Historically, the process goes out of control during the summer. Moist weather makes the air too wet to dry the coating at high speed. The coating gets blotchy, uneven and candy pieces stick together. We can imagine an existing system which has previously used air from other parts of the building to dry the coating. That air was cool, but very moist.

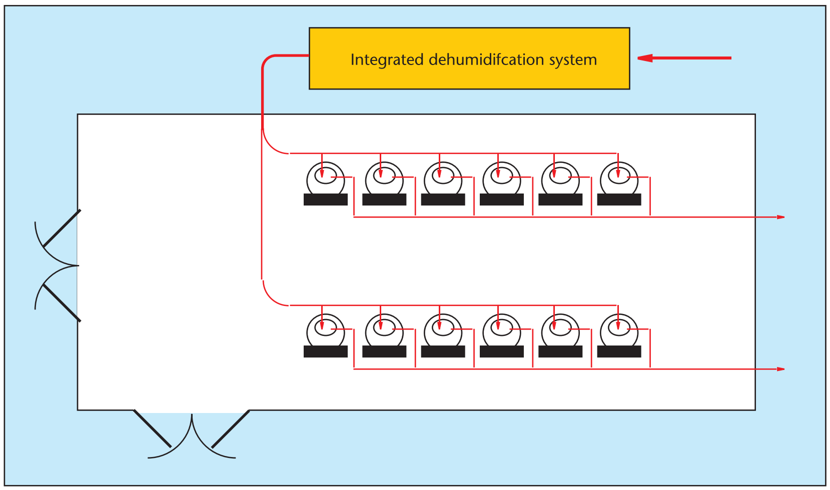

FIGURE 7.30
Confectionery, pharmaceutical and chemical manufacturing frequently use dry air in coating and drying operations. This example describes how fresh air is taken from the weather and conditioned to meet two different drying requirements.

Step Two — Establish control levels and tolerances

Depending on the time of the year and on product demand, the company runs one of two products through these particular drying bins.

- Chocolate-covered nuts with a thin sugar shell coating
- Hard-coated sugar candy — similar to “Jawbreakers”

The products are not run at the same time, and air temperature and moisture requirements for each product are different. The chocolate product needs cold air, because otherwise the chocolate loses its "temper" (recrystallizes above its melting point) from the heat generated by the tumbling action of the product in the bin. On the other hand, the hard candy needs hot air to set the sugar coating and to form an especially smooth and shiny surface which is part of the product's primary marketing identity.

Past processing has shown that 65°F is adequate for the chocolate product, but winter production has shown that 50° is even better because that keeps the pieces quite hard, reducing dents in them caused by tumbling in the bin. The hard candy product will need 120° air to set the shiny coating.

The drier the air, the faster the product coating will dry. Unfortunately, two factors limit how dry the air can be as it is fed to the drying bins — system cost and over drying. The drier the air, the more it will cost to buy and run the system. Also, if the air is too dry, the sugar coatings crystallize in an amorphous pattern, which results in weak layers which crack easily. Without resources to have laboratory tests conducted, the engineer is confused about how to establish the humidity level that will give the fastest speed without over drying. This delays the project past the deadline for capital budget approvals, so the project is put on hold for a year.

During the following summer and winter seasons, the engineer decides to rent a desiccant dehumidifier to make some experiments on the drying system itself. The results of these experiments confirm that the following conditions are optimum for each product:

- 50°F, 50%rh (27 gr/lb) for the chocolate product
- 120°F, 9%rh (44 gr/lb) for the hard candy
- 250 cfm airflow for each drying bin

Step Three — Calculate heat and moisture loads

The existing system must be entirely changed. It simply takes cool air from the room, blowing it into each of 12 rotating drying bins. There are exhaust fans in the ceiling that pull moist air out of the building, but there is no make-up air system to supply treated air to the room. This means air has to be pulled into the room from other production areas by the exhaust fans. The engineer needs an entirely new system concept before he can calculate loads.

First, the engineer must decide where the drying air will come from, and whether it will continue to be exhausted to the weather or returned to the dehumidification system. If it returns to the system, will it do so directly, or will it be pulled through the entire room after it leaves the drying bins? If the air is returned from the drying bins directly, it will carry only the moisture load from the coating. If it comes back through the room, it will contain the moisture loads from the room as well as the product.

The engineer makes two fundamental decisions: take air from the weather rather than the building and exhaust it directly from the drying bins to the weather, returning no air to the dehumidification system. He reasons that the old practice of pulling air from the building simply places the cooling load on systems in the rest of the building which are already overloaded. By putting the drying bins on an independent system that treats its own outside air, he can improve conditions in the rest of the building as well as improve the coating operation.

By not returning air to the dehumidification system after it leaves the bins, the engineer greatly simplifies his maintenance work. If the air returned to the system it would have to be heavily filtered to remove sticky, unsanitary sugar dust. Filter systems are available, but the engineer knows his maintenance technicians do not have time for current, critical tasks. He knows he will get no increase in maintenance budgets to hire people to carefully maintain an expensive filtration system.

In making this decision, the engineer has also looked at the moisture conditions in the weather compared to the moisture of the air leaving the coating pans, recognizing that if the air from the pans is drier, the dehumidifier will use more energy than if the system takes air from the weather. He found that for half of the year, the air from the weather is actually drier than the air leaving the bins, so weather air is less expensive to process. An optimized system would switch the intake automatically between the bins and the weather, whichever is drier — but the engineer rejects that alternative in the interest of simplicity and reduced maintenance.

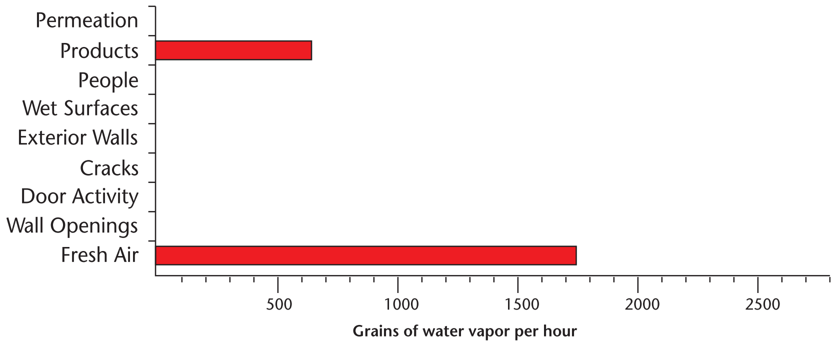

FIGURE 7.31
Since air is not returned to the system after it picks up moisture from the bins, the only moisture loads are the water vapor contained in the fresh air and the moisture from the wet product itself.

These decisions have certainly simplified the moisture load calculation. The only heat and moisture loads on the system are contained in air taken from the weather. There are no relevant room loads, people or door openings. The system must simply take fresh air and deliver it at 120°F and 44 gr/lb for the hard candy, and 50°, 27 gr/lb for the chocolate-covered nuts.

The sensible heat load consists of three elements:

- Sensible heat generated by product pieces tumbling in the drying bin
- Sensible heat from the fresh air
- Latent heat in the fresh air, converted to sensible heat by the dehumidifier

Experience has shown that heat generated by the product can be removed by supplying air at 50°F for the chocolate. Sensible heat from the outside air is the difference between the summer design temperature and the 50° delivered air condition to the bins — 133,000 Btu/h. Latent heat in the fresh air will be converted to sensible heat by the dehumidifier, in proportion to the amount of moisture the unit removes. The more moisture removed, the warmer the air will be following the dehumidifier. When the engineer sizes the dehumidifier, he will know the sensible load from the dehumidification process.

There is also a winter sensible heating load, since the hard candy needs air at 120°F. The winter design condition in the area is 11°, so the system must supply:

EQUATION 7.14

$$\textit{heating load} = \underbrace{3000}_{\text{Total airflow to the drying bins (cfm)}} \times \underbrace{1.08}_{\text{Specific heat of air (Btu/h per cfm per °F)}} \times \underbrace{(120 - 11)}_{\text{Temperature difference between the winter fresh air and air delivered to the bins (° F)}}$$

heating load = **353,160 Btu/h**

In theory, there might also be a need to humidify the fresh air in winter, since the moisture outside will be lower than the 27 gr/lb and 44 gr/lb that seem to be ideal for the products. But in fact, the engineer has never noticed a problem with excessive dryness in winter, so he does not bother to calculate a humidification load.

Step Four — Size the components to remove the load

Since desiccant dehumidification converts moisture to sensible heat, the dehumidifier must be selected and its performance determined before sizing the cooling system which follows the dehumidifier. The size of the dehumidifier depends on the temperature and moisture conditions of the entering air. Economically, it makes sense to pre-cool and dehumidify the fresh air before it enters the desiccant unit. The question is — how much pre-cooling is appropriate?

The answer depends on whether the thermal energy used for reactivation is less expensive than the electric power that runs the cooling system. When reactivation energy is inexpensive, the dehumidifier can remove moisture at low cost. The unit will cost more to install, but operating cost savings can offset that additional cost in a matter of months. Conversely, if reactivation energy is comparatively expensive, then cooling dehumidification should take as much of the load as practical — the installation will be less costly to install and run.

In this case, the engineer plans to use waste heat from a cogeneration plant for reactivation, so the cost of running the desiccant unit will be almost negligible. Therefore the only appropriate pre-cooling is the minimum necessary to let the desiccant unit deliver the 27 gr/lb condition required by the chocolate product.

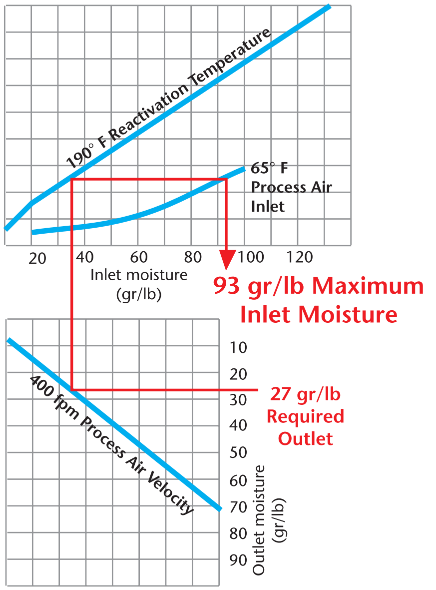

FIGURE 7.32

In this example, the engineer needs to minimize the amount of pre-cooling before the air enters the dehumidifier. He starts with the moisture condition he knows must be sent to the drying bins, and works backward to the maximum temperature and moisture that can enter the dehumidifier for that outlet condition.

The engineer consults the performance charts in the Appendix for rotating Honeycombe® dehumidifiers. Based on his known requirement for 3,000 cfm, he selects a model with 7.5 sq.ft. process face area, then he runs the performance curves in the reverse of the normal sequence. 3,000 cfm will move through this unit at 400 feet per minute. (3,000 ÷ 7.5 = 400 fpm) Starting with the desired end condition of 27 gr/lb, he moves left to the 400 feet per minute velocity curve, then vertically to the 190°F reactivation temperature line and right to the 65° entering temperature line and down to the inlet moisture condition, which reads 95 gr/lb. This means that if the air enters the desiccant unit no warmer than 65° and no wetter than 95 grains, the 7.5 sq.ft. unit can deliver the required 27 gr/lb condition.

The engineer uses the 190°F reactivation temperature because the cogeneration system provides 200°F water, so the hottest practical temperature for reactivation is 190°. Finally the engineer assumes a process air inlet temperature of 65°F because that allows the wettest possible inlet condition. In fact, if air is pre-cooled to 65°, it will be saturated and have a moisture content of 93 gr/lb.

This tells the engineer what the pre-cooling coil must accomplish — it must bring the air from a summer design condition of 91° and 146 gr/lb to a condition of 65° and 92 gr/lb.

EQUATION 7.15

Fresh air from the weather (cfm)

Lbs. of air per cfm per hour

Enthalpy change between air entering and leaving the cooling coil (Btu/lb)

$$\textit{pre-cooling tonnage} = \frac{3{,}000 \times 4.5 \times (45 - 30)}{12{,}000}$$

Btu/h per ton of air cooling capacity

$$\textit{pre-cooling tonnage} = \mathbf{16.9\ tons}$$

The cooling requirement downstream of the desiccant unit will depend on the temperature rise through the process side of the dehumidifier, which can be calculated as shown in the Appendix.

EQUATION 7.16

$$\text{temperature leaving process} = 65 + .625(92 - 27) + 0.1(190 - 65)$$

$$\text{temperature leaving process} = \mathbf{118\ ^\circ F}$$

Temperature of air entering process (°F)
Constant which converts moisture to temperature (° F per gr/lb)
Moisture removed by the dehumidifier (gr/lb)
Temperature difference between air entering process and air entering reactivation (° F)
Heat carryover factor (dimensionless)

EQUATION 7.17

$$\text{post-cooling tonnage} = \frac{3{,}000 \times 1.08 \times (118 - 50)}{12{,}000}$$

$$\text{post-cooling tonnage} = \mathbf{18.4\ tons}$$

Airflow to the drying bins (cfm)
Btu/h per cfm of air per °F
Temperature difference between air entering and leaving the cooling coil (° F)
Btu/h per ton of air cooling capacity

Step Five — Select the controls

Temperature and humidity must be controlled to meet the needs of both products. Equipment energy consumption must modulate to follow the loads, so energy is not wasted.

Humidity control is achieved through a set of modulating dampers which bypass air around the dehumidifier when the moisture load is low. Humidity must be controlled at two fixed points: 44 gr/lb and 27 gr/lb. While these two conditions are quite close together in absolute humidity, they are very far apart in relative humidity. Since the 27 grain condition is delivered at 50°F, it represents 50% relative humidity. The 44 grain condition is delivered at 120°, therefore it represents 9% relative humidity. Relative humidity sensors control well at a single point, where the sensor can be calibrated and used constantly, but they would need recalibration when the control point shifts. Also, relative humidity sensors in general are less accurate than other types when measuring below 10% relative humidity.

The engineer decides to use an optical dew point sensor-controller and to control the system based on dew point rather than relative humidity. Because the air temperature is quite high, the optical sensor is ordered with extra cooling capacity to ensure accuracy. Also, the sensor is placed well downstream of the desiccant unit, since the outlet moisture conditions vary widely across the wheel face. A better location for the

Location	Airflow (cfm)	Temp (°F)	Moisture (gr/lb)
A	3000	91	146
B	3000	65	92
C	3000	118	27
D	3000	50	27

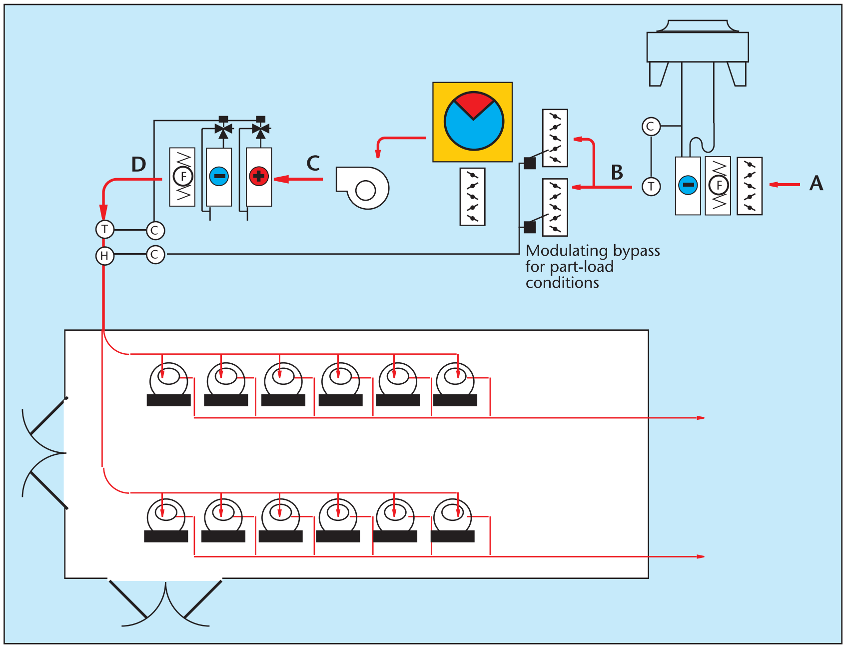

FIGURE 7.33
The dehumidifier is installed with a bypass so its capacity can reduce when fresh air is drier than the design condition. The fixed-position damper in the bypass creates a resistance equal to that of the desiccant bed. This lets the bypass dampers modulate airflow smoothly through both dehumidifier and bypass.

The fixed-position damper at the inlet to the system allows the engineer to set the system airflow exactly, so there is less variation in flowrate with changes in weather air temperature and pressure.

sensor is immediately before the air enters the short distribution ducts to the bins — at that point the air is reasonably uniform in temperature and moisture across the duct.

Airstream temperature is controlled in two places — before and after the dehumidifier. Air is pre-cooled for partial dehumidification. That coil is the direct-expansion type, served by its own condensing unit. This frees capacity in the central chilled water system. The pre-cooling system is controlled by a temperature sensor/controller immediately following the cooling coil. It is set at 65°F. Anytime the air temperature is above 65, the system will cool and dehumidify the incoming air.

The post-cooling coil is served by the central chilled water system. The engineer placed the cooling coil downstream of the heating coil to make sure the water in the coil will not freeze during winter operation. In theory, the maintenance department can prevent coils from freezing by draining their water in wintertime. But the engineer knows that the weather changes faster than the hard-pressed maintenance technicians can drain and refill the coil. The post-heating coil is provided with water at 200°F from the cogeneration plant, just like the reactivation heating coils for the dehumidifier.

Both post-heating and post-cooling coils are controlled by three-way valves which modulate fluid flow in response to a temperature sensor included with the optical dew point controller. Another advantage to this more costly instrument is that temperature and humidity set points can be reset at the process control console rather than having to send technicians into the ductwork.

Because there is such wide variation in moisture loads from the weather, the dehumidifier energy requirements will also vary widely. The unit should reduce energy automatically in response to these changes. This is less important during the summer months, because there is an excess of heat from the cogeneration system. But during the winter, all available thermal energy is needed to provide comfort and process heating.

The dehumidifier reactivation energy is controlled by changing the hot water flow through the heating coils in response to a temperature sensor/controller located in the airstream leaving reactivation. The controller is set at 120°F. As the air temperature rises above that point, it means there is less moisture being absorbed on the process side, and reactivation energy can be reduced.

Other considerations

Freeze-protection

This project uses hot water from a cogeneration system for reactivation and supply air heating. In the winter there is a danger of freezing the water in the coils, which would burst the tubes. To avoid this potential problem, the engineer could take air from a warm mechanical room rather than directly from the weather. Alternately, an electric or natural gas pre-heater could be placed on the outside air intake to the system. In any case, hot water-based heating systems for weather air are subject to freezing in cold climates, and the engineer must take precautions to avoid the problem.

Supply air fan location

In the example, the supply air fan is located downstream of the desiccant dehumidifier. This provides for good mixing between the dehumidified and bypassed air, but it places the ductwork between the fan and the dehumidifier under a strong negative pressure compared to the air outside the duct — which is moist. The negative air pressure can lead to air leaks through duct joints, and leakage within the dehumidifier casing, so both should be well sealed if the fan is in this location. Placing the fan upstream of the desiccant unit avoids the leakage problem downstream of the unit, but the designer must take care to avoid blowing the air directly at the face of the desiccant wheel. Otherwise air will not move through the bypass duct when necessary.

Pre-cooling

One of the reasons it makes sense to use minimum pre-cooling in product drying applications is the need for a warm or a hot temperature combined with dry air. Since the dehumidifier is converting the moisture load to sensible heat, the process benefits. However, if this project dealt only with the heat-sensitive chocolate product, the operating and installation costs could be reduced by doing as much pre-cooling as the refrigeration system will allow. Refrigeration capacity is almost always less costly to buy than desiccant dehumidification capacity until the cooling system encounters a freezing problem. The practical lower limit for chilling outside air is debatable. Most engineers would agree that 45°F for direct expansion cooling and 40°F in glycol-water systems are achievable, although the cooling coils will be very deep and the resistance to airflow will be quite high.

Optimum conditions can change in mid-cycle

In this example the airflow, temperature and moisture can stay constant because product batches are identical in size, shape, type of coating and degree of dryness required. However, in other temperature-sensitive, batch-oriented drying applications such as investment castings, fish filets or seeds, it may be important to be able to change temperature, air velocity and humidity throughout the drying cycle to avoid over-drying. When this is necessary, the engineer may wish to invest in more complex controls to automatically re-set the air volume, temperature and moisture in mid-cycle for best results.

Total system airflow

In many product drying applications, the total system airflow is large, because of the need to maintain high velocity across all surfaces of the product. But the dehumidifier need not handle all that air. For example, consider that instead of small candies in a small drying bin, the engineer is trying to dry large investment casting “Christmas trees” in a room measuring 50' wide x 75' long x 18' high. The moisture load may only be 50 lbs of water vapor per hour. The dehumidifier will be small in comparison to the airflow required to create high velocity on all the complex product surfaces. In such cases, it is best to add large circulation fans to move the air within the room, with the dehumidifier only drying a small portion of the total air in circulation.

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8

OPTIMIZING MIXED SYSTEMS

In this chapter, we will examine dehumidification systems for larger rooms and buildings in more detail, since there are many alternate means of achieving the same dehumidification result.

The central issue is to determine the optimum mixture of cooling and desiccant dehumidification that will result in the lowest possible first cost and operating cost for a given set of circumstances. We will consider five possible system schematics for a single circumstance, and show the operating cost and first cost consequences of each of the alternatives.

The equations and calculations in this chapter assume the reader is familiar with basic heating and air conditioning terms and numerical relationships. If this is not the case, we suggest that the reader examine Chapters 2 and 7 before proceeding with this material.

Example Case Description

Humidity control systems for buildings often involve heating and cooling as well as humidification and dehumidification. Since cooling will generally be accomplished with refrigeration, the engineer can often use that equipment to dehumidify as well. The questions then become: how much cooling equipment and how much desiccant equipment is appropriate in a mixed system, and how should the components be arranged for maximum economy? We will consider the following specific case:

FIGURE 8.1

Project Purpose	Prevent clogging of packaging machinery for hard candy during peak summer production
Control levels	70°F ± 3° 35°F Dew point (30 gr/lb) +0 gr, -10 gr
Internal sensible heat load	100,000 Btu/h
Internal moisture load	181,800 gr/hr
Make-up air	400 scfm @ 95°F, 140 gr/lb (44.9 Btu/lb enthalpy)
Minimum delivered air temperature	55°F
Cooling available	Chilled water @ 45°F 10°F approach of the air to the coolant temperature

In this example, the moisture control level is below the practical limitations of a system based only on cooling dehumidification — a desiccant unit will be necessary to meet the specification. There are many possible system configurations, and we will examine the four shown on the facing page.

System 1 – Dry the make-up air only

This alternative can be quickly assessed without detailed calculations or selecting specific components. Assume the make-up air could be dried very deeply — down to a moisture content of 1 grain per pound. Then it could remove the following amount of moisture from the room:

Moisture removal =400 x 4.5 x (30-1) = 52,200 gr/hr

FIGURE 8.2

System 1

Pre-cool the make-up air and dehumidify it with a desiccant unit before the air blends with return air from the room. When the system requires a large proportion of make-up air, this schematic is generally the most economical to install and operate.

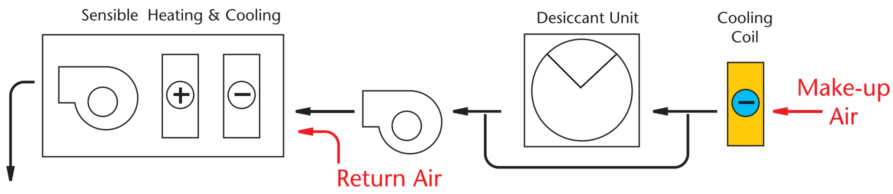


FIGURE 8.3

System 2

Pre-cool the make-up air, then blend it with the return air before dehumidifying with a desiccant unit. If the system does not use enough make-up air to provide complete dehumidification using system 1, this configuration is often the most economical choice.

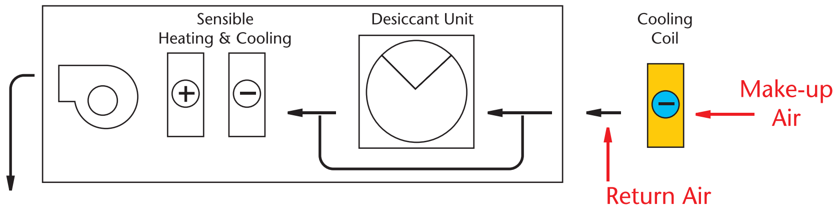


FIGURE 8.4

System 3

Pre-cool the blended make-up air and return air before dehumidifying with a desiccant unit. Cool air improves the performance of the dehumidifier, increasing the capacity of the system, and in some cases allowing the designer to use a smaller unit.

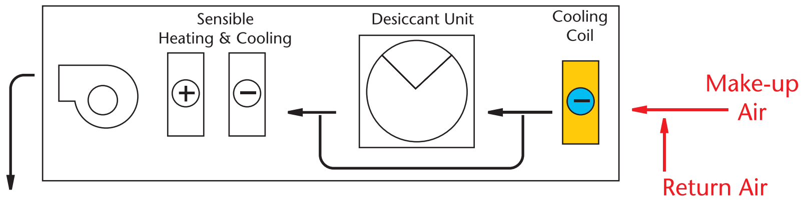
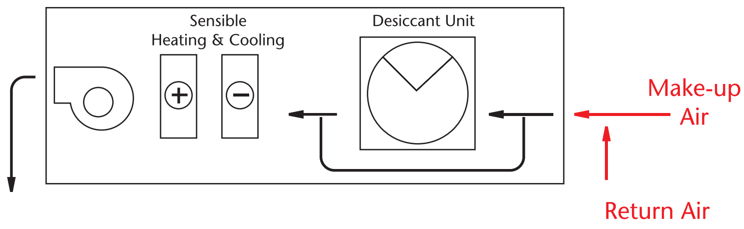


FIGURE 8.5

System 4

Blend the make-up air and return air before dehumidifying with a desiccant unit—no pre-cooling. If the make-up air does not carry a significant proportion of the moisture load, it may not be useful to pre-cool in front of the desiccant unit.



Since the internal moisture load is 181,000 gr/hr, it is obvious that dry make-up air alone cannot meet the design requirements. In other cases — such as a clean room where the make-up air may be a larger volume and the internal moisture load smaller, drying the outside air is often the best alternative. This is also true in many commercial buildings, where the moisture control level might be higher, which gives each cubic foot of dry make-up air more moisture removal capacity. In any event, drying the make-up air alone will not suffice in this case.

System 2 – Pre-cool the make-up air and dry the blend with a desiccant unit

Using the available chilled water, we pre-cool the make-up air to 55°F, 64 gr/lb and blend it with return air. The desiccant unit must then remove the internal load as well as the remaining moisture from the make up air. This total load is the sum of the internal load plus the moisture remaining in the make-up air after the cooling coil:

Total load = 181,800 + [(4.5 x 400 x (64-30)] = 243,000 gr/hr

To determine dehumidifier performance, we must know the temperature and moisture content of the air entering the unit, and to determine those values, we need to know how much return air is being blended with the make-up air. This combined air quantity generally depends on the space cooling requirement and on the minimum allowable temperature that can be supplied to the space. To avoid uncomfortable drafts, we will set the minimum supply air temperature at 55°F, which allows us to calculate the total air quantity needed to remove the internal sensible heat load:

Total supply airflow = 100,000 ÷ [1.08 x (70-55)] = 6173 scfm

So the blended air condition to the dehumidifier will be at the following condition:

Make-up air	*400 scfm @ 55°F, 64 gr/lb*
Return air	*5773 scfm @ 70°F, 30 gr/lb*
Total	*6173 scfm @ 69°F, 32 gr/lb*

Selecting the appropriate size dehumidifier is essentially an iterative process — the designer must determine the smallest unit that will remove the 181,800 gr/hr load from the air stream. Using the rotary Honeycombe® dehumidifier performance data given in the appendix, the designer selects a unit with a nominal face area of 3.75 sq.ft. to dry 2250 scfm of air to a moisture content of 6.2 gr/lb. The unit performs as follows:

Inlet conditions: 2250 scfm @ 69°F, 32 gr/lb

Face velocity = 2250 ÷ 3.75 = 600 fpm

Outlet moisture content = 6.2 gr/lb

Moisture removal = 4.5 x 2250 x (32 - 6.2) = 261,225 gr/hr

Process air temperature rise = .625 x (32 - 6.2) + [.07 x (250 - 69)] = 28.8 °F

Process air outlet temperature = 69 + 28.8 = 97.8 °F

Reactivation air volume = 2250 x [(97.8 - 69) ÷ (250 -120)] = 500 scfm

Reactivation energy at design = 500 x [(250 - 95) x 1.08] = 83,700 Btu/h

The total sensible heat load consists of the internal load plus the heat from the dehumidification process:

Total sensible load = 100,000 + [1.08 x 2250 x (97.9 - 70)] = 167,797 Btu/h

The total cooling includes this sensible heat, plus the pre-cooling for the make-up air:

Total cooling = 167,797 + [4.5 x 400 x (44.9 - 23.1)] = 207,037 Btu/h

We can see this design calls for the following investments:

Cooling Capacity = 17.2 tons

Dehumidifier size = 3.75 sq.ft process air face area

Required desiccant reactivation heat = 83,700 Btu/h

System 3 – Pre-cool the blended air before it enters the desiccant unit

Without pre-cooling the make-up air, we allow it to blend with the return air, and then cool the mixture before it enters the desiccant dehumidifier. The moisture load on the dehumidifier is larger, since the outdoor air moisture has not been removed from the make-up air:

Moisture load = [181,000] + [4.5 x 400 x (140 - 30)] = 379,800 gr/hr

The conditions of the air entering the cooling coil in front of the dehumidifier are calculated as before:

Make-up air	*400 scfm @ 95°F, 140 gr/lb*
Return air	*5773 scfm @ 70°F, 30 gr/lb*
Total	*6173 scfm @ 71.6°F, 37 gr/lb*

The designer iterates as before, finding that a larger dehumidifier must be used. A unit with 7.5 instead of 3.75 sq.ft. is selected, and it performs as follows:

Inlet conditions: 2500 scfm @ 55°F, 37 gr/lb

Face velocity = 2500 ÷ 7.5 = 333 fpm

Outlet moisture content = 1.5 gr/lb

Moisture removal = 4.5 x 2500 x (37 - 1.5) = 399,375 gr/hr

Process air temperature rise = .625 x (37 - 1.5) + [.06 x (250 - 55)] = 33.9°F

Process air outlet temperature = 55+ 33.9 = 88.9°F

Reactivation air volume = 2500 x [(88.9 - 55) ÷ (250 -120)] = 652 scfm

Reactivation energy at design = 652 x [(250 - 95) x 1.08] = 109,145 Btu/h

The total sensible heat load now consists of the internal load plus the heat from the dehumidification process plus the pre-cooling of the blended air entering the dehumidifier:

Total sensible load = 100,000 + [1.08 x 2500 x (88.9 - 70)] +
[1.08 x 6173 x (71.6 - 55)] = 261,670 Btu/h

Summarizing this system, we can see the following investments:

Cooling Capacity = 21.8 tons

Dehumidifier size = 7.5 sq.ft process air face area

Required desiccant reactivation heat = 109,145 Btu/h

System 3 uses a desiccant unit twice the size of the unit in system two, and a cooling system that is 25% larger than the cooling in system two. This is the result of not removing any moisture with the cooling system. However, system three has a much greater “reserve” moisture removal capacity, because the desiccant unit — currently handling 2500 cfm — could dry up to 4500 cfm if necessary.

System 4 - Eliminate all pre-cooling and remove all moisture with desiccants

Since the desiccant unit selected in system three has substantial extra capacity, the engineer can examine the possibility of using the same unit without pre-cooling of any kind, and use low-cost, low-temperature heat for desiccant reactivation. In this case, we will assume that waste heat from a cogeneration system is available to heat the reactivation air to 200°F.

The moisture load is the same as the previous cases:

Moisture load = [181,800] + [4.5 x 400 x (140 - 30)] = 379,800 gr/hr

The conditions of the air entering the dehumidifier are calculated as before:

Make-up air	*400 scfm @ 95°F, 140 gr/lb*
Return air	*5773 scfm @ 70°F, 30 gr/lb*
Total	*6173 scfm @ 71.6°F, 37 gr/lb*

The goal is to use the smallest possible amount of process air that will result in removing 379,800 gr/hr from the total airflow of 6173 scfm @ 71.6°F, 37 gr/lb. If reactivation air enters at 200°F, the process air will leave the unit between 8 and 11.5 gr/lb, depending on process air velocity. {*"The slower, the lower"* as a maintenance technician once observed}. The designer can make an estimate of the necessary process airflow by assuming the worst case (wettest outlet):

Estimated process airflow = 379,800 ÷ [4.5 x (37 - 11.5)] = 3310 scfm

3310 cfm actually moves through a unit with 7.5 sq.ft. process face area at a rate of 441 fpm (3516 ÷ 7.5 = 441), so performance will be better than the 11.5 grain "worst case", which occurs at 600 fpm. Consulting the unit performance chart for a velocity of 441 fpm, the designer concludes the air will leave the unit at 10 gr/lb.

Moisture removal = 4.5 x 3310 x (37 - 10) = 402,165 gr/hr

The capacity is ample. Complete performance data can now be calculated:

Inlet conditions: 3310 scfm @ 71.6 °F, 37 gr/lb

Face velocity = 3310 ÷ 7.5 = 441 fpm

Outlet moisture content = 10 gr/lb

Process air temperature rise = .625 x (37 - 10) + [.08 x (200 - 71.6)] = 27.1°

Process air outlet temperature = 71.6 + 27.1 = 98.7 °F

Reactivation air volume = 3310 x [(98.7 - 71.6) ÷ (200 -120)] = 1121 scfm

Reactivation energy at design = 1121 x [(200 - 95) x 1.08] = 127,121 Btu/h

The total sensible heat load consists of the internal load plus the heat from the dehumidification process:

Total heat load = 100,000 + [1.08 x 3310 x (98.7 - 70)] = 202,597 Btu/h

Summarizing this system, we can see the following investments:

Cooling Capacity = 16.9 tons

Dehumidifier size = 7.5 sq.ft face area

Required desiccant reactivation heat = 127,121 Btu/h

Interestingly, this alternative results in less total cooling than either system three or system two, and still has quite a bit of reserve capacity. The designer could decide to install two reactivation heaters in series — the first using low-temperature waste heat, to heat the air to 200°. The second heater could boost the air temperature to 250°, which would improve moisture removal for future increases in moisture load, or allow the room to operate at a lower humidity control level for different products. It would also be necessary, however, to add cooling capacity in that situation. Sensible heat load is partly a function of the amount of water removed by the desiccant unit.

Comparing Alternatives

The table and graphic on the facing page compare the four alternate systems. Each has advantages and disadvantages. For example:

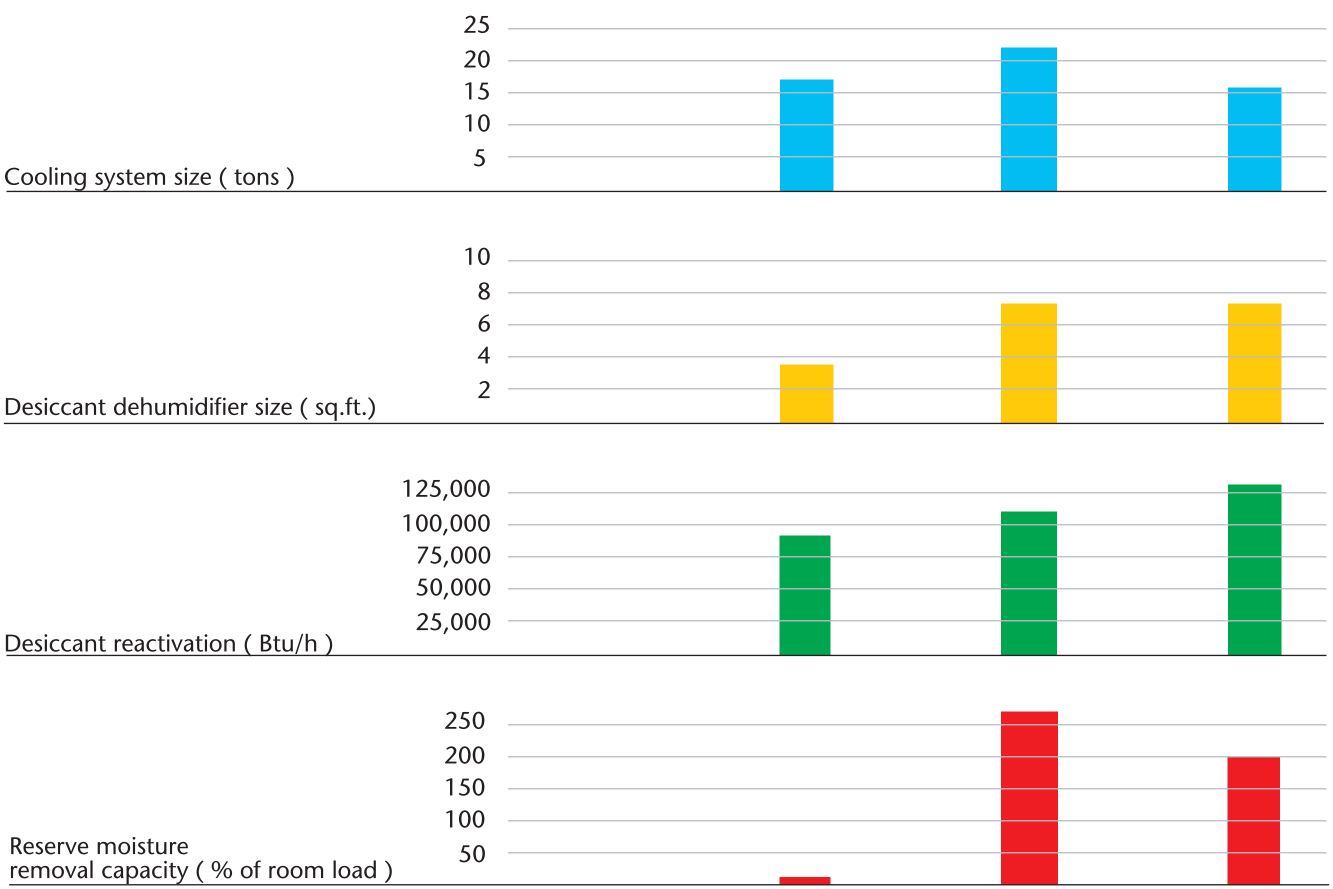
• **System 1** has enough capacity to remove the load from the make-up air, but cannot remove the internal load from the room. If the volume of make-up air were larger, the internal load smaller or the control dew point higher, this system would be the smallest and most economical to operate. However, in this case, such an arrangement cannot work.

• **System 2** is the least costly to purchase. It has the smallest desiccant unit, a small cooling plant and the least amount of reactivation energy. However, it just barely removes the load from the room. The designer may want to re-check the moisture load calculations if this system is chosen, or perhaps use the next size desiccant unit.

• **System 3** provides more moisture removal capacity than system 2 by increasing the desiccant dehumidifier size and by improving desiccant performance with cool inlet air at the same time. This system will be the most costly to install and operate. However, if the original load calculations were perhaps more of a guess than a firm calculation, or if there is a need for future expansion, this system may be a wise choice.

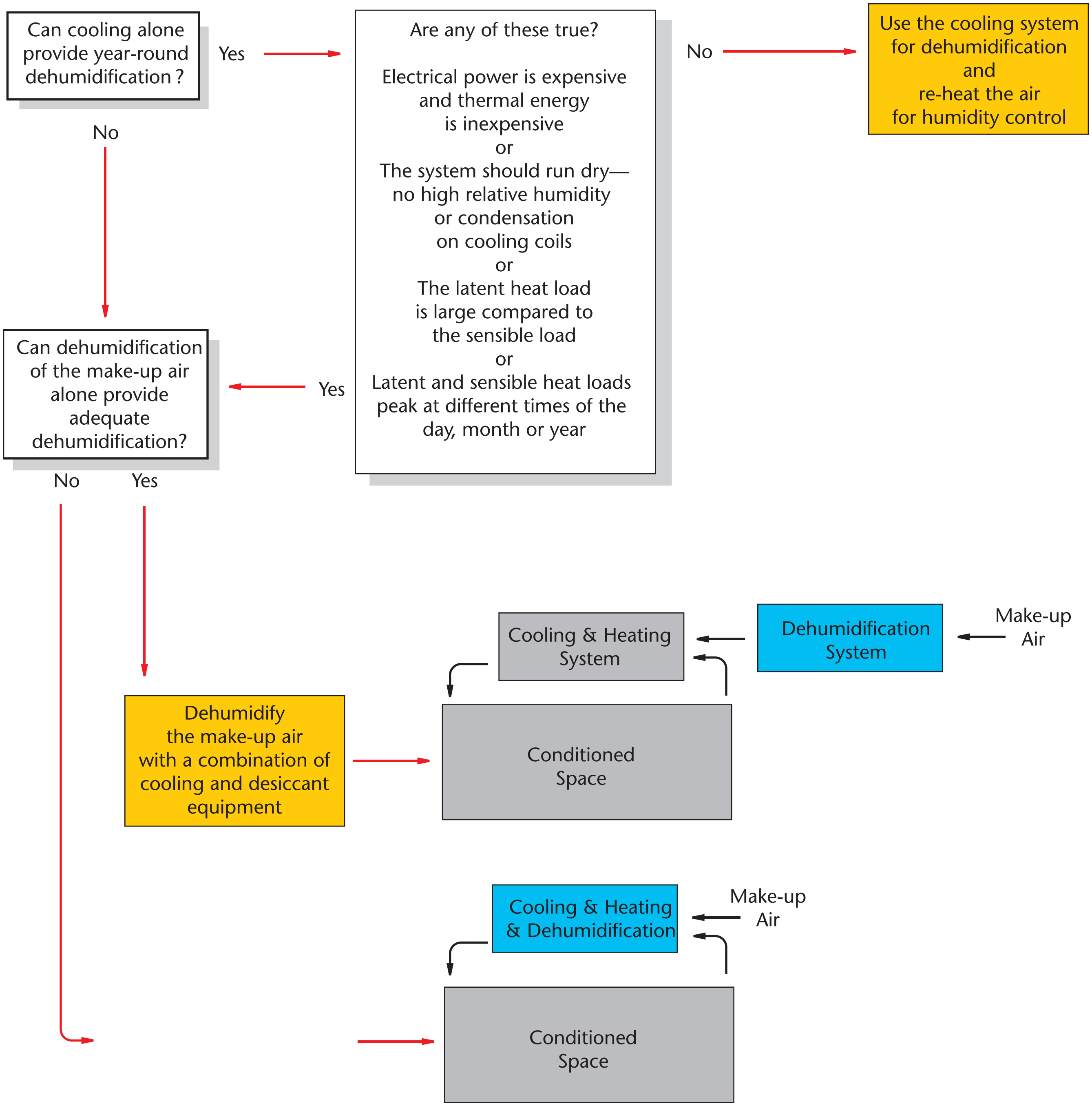
• **System 4** is more costly to install than system 1, but less costly than system 2, and it will certainly be the least expensive to operate. Using waste heat to reactivate the desiccant keeps energy costs to a minimum. The cooling coil only uses the minimum chilled water necessary to remove the sensible heat load — not to pre-cool to improve desiccant performance. On an annual basis, this system will probably cost less than half the operating cost of system 3, but will still have the same reserve capacity if the designer installs a high-temperature reactivation heater and extra cooling for future use.

	System 1	System 2	System 3	System 4
Cooling system size (tons)	Inadequate	17.2	21.8	16.9
Desiccant dehumidifier size (sq.ft.)	Inadequate	3.75	7.5	7.5
Desiccant reactivation (Btu/h)	Inadequate	83,700	109,145	127,121
Reserve moisture removal capacity (lb/hr , % of load)	Inadequate	3.4 (13 %)	41.8 (161%)	28.8 (101%)



Note that any change in these circumstances would change the advantages and disadvantages of these different configurations. In particular, increasing the make-up air quantity can have a major effect — making system 1 the best choice to keep both first cost and operating cost to a minimum.

The diagram on the facing page shows a logical thought-path to the dehumidification system that will cost the least to install and operate. But as the preceding exercise demonstrates, there can be many circumstances that alter the general case. As we described in Chapter 7, it is essential that the dehumidification system designer understand the basic purpose of the project. This understanding provides a map through the maze of trade-offs between the capacity, energy and first cost of dehumidification systems.


Can cooling alone provide year-round dehumidification ?
Yes
No
Are any of these true?
Electrical power is expensive and thermal energy is inexpensive
or
The system should run dry— no high relative humidity or condensation on cooling coils
or
The latent heat load is large compared to the sensible load
or
Latent and sensible heat loads peak at different times of the day, month or year
No
Use the cooling system for dehumidification and re-heat the air for humidity control
Yes
Can dehumidification of the make-up air alone provide adequate dehumidification?
No
Yes
Cooling & Heating System
Dehumidification System
Make-up Air
Dehumidify the make-up air with a combination of cooling and desiccant equipment
Conditioned Space
Cooling & Heating & Dehumidification
Make-up Air
Conditioned Space

9 HUMIDITY & MOISTURE INSTRUMENTATION

Instrumentation to measure and control moisture and humidity is very diverse. For a complete discussion, humidity instrumentation would need a separate and equally large handbook. However, there are some fundamental issues and basic operating principles which are useful to know when designing dehumidification systems. This chapter deals with those basics.

Since there are so many options for instrumentation, it is important to avoid becoming confused by details, and to maintain a perspective on what is truly important in a given application. If a designer is laying out an archival storage room for paper documents, there is no need to be concerned about $\pm 1\%$ repeatability, dew point measurements accurate to $\pm 0.5°F$ or response times under 4 seconds — these issues are irrelevant. But conversely, if an engineer is designing a photographic film manufacturing line producing $5,000 worth of product every thirty seconds, he or she should not be overly concerned about the cost of the control — small errors or machine downtime will cost more in a few minutes than the cost of the most expensive instruments available.

In this chapter, we will discuss a framework for making decisions about instrumentation, and basic operating principles of different devices.

Moisture and humidity instruments are differentiated according to the *type of measurement* they take, the *duty cycle* on which they take it and the *functions they perform* once the measurement is taken. The system designer can quickly choose from the bewildering array of instruments once these variables have been defined for the application in question.

Types of Measurements

Instruments can measure the *relative humidity* of air, its *specific humidity* and the *moisture content of solids.* Instruments for each of these functions are different.

• **Relative humidity** sensors generally measure the change in a material that results from absorption of moisture from the surrounding air. Many materials are moisture sensitive and suitable for the purpose. As early as 1550, Leonardo Da Vinci observed that a ball of wool weighs more on a rainy day than on a dry day. Human and animal hair also gets longer as it absorbs moisture, which is what causes hair to curl on a muggy day. And the electrical characteristics of plastics change as they absorb moisture. Instruments can measure these changes and make a numerical correlation to changes in relative humidity in the surrounding air.

• **Absolute humidity** sensors either use a chemical reaction to “count” water molecules, or cause condensation on a cold surface with a known temperature. The first recorded use of a condensation hygrometer was in 1751, when the French naturalist Charles LeRoy added ice to cool a water-filled, polished-steel container while taking the temperature of the water. When drops of condensation formed on the outside of the container, he was able to define the dew point of the surrounding air. The same phenomenon is still in use today, and with electronic controls, such “condensation hygrometers” are very precise.

• **Moisture content of solids** can be determined by measuring some property of the material that differs with moisture content, or by removing all the water from a sample of the material and measuring the weight change. For instance, water absorbs infrared energy very efficiently. When a solid material is exposed to infrared radiation, sensors can measure the difference between the reflected radiation and what is absorbed by the material. The difference is proportional to the moisture content of the material surface.

Accurate primary measurements — those which measure the specific amount of moisture independent of another variable like air or material temperature — are more accurate, but more costly. Often, less expensive measurement systems can, through electronics, produce a specific moisture reading by sensing two or more variables and calculating the primary moisture value. For instance, sensors which measure temperature, pressure and relative humidity can use these values to calculate dew point or humidity ratio.

Generally, however, it is good practice to measure the variable of interest directly rather than converting from other values when project economics allow. For example, consider an application to reduce moisture absorption by hygroscopic powders in a pharmaceutical tableting operation. In most cases, moisture absorption depends on the relative humidity in the air surrounding the material, not on the air dew point or humidity ratio. In this case, it would be wise to look for instruments which respond to changes in relative humidity rather than air dew point.

Conversely, an application to control condensation on cold pipes in a water treatment plant cannot be controlled efficiently by measuring relative humidity on a wall ten feet from the pipes. Condensation can occur at any relative humidity in the air — it depends on the difference between the air dew point and the pipe surface temperature. In that case, the designer needs an instrument which senses either air dew point, or the difference between the dew point and the surface temperature.

Duty Cycle & Operating Environment

Another fundamental difference between instruments is the duty cycle they were designed to satisfy. Some are suitable for batch samples, while others provide a continuous measurement.

For instance, the “gravimetric train” is generally agreed to be the most accurate known moisture measurement device. It is used by national standards agencies throughout the world as the primary standard by which other devices are calibrated. However, the instrument is huge, slow and very costly, requiring several trained technicians for operation, and capable of measuring only one or two small air samples each day. While it is accurate, the gravimetric train is clearly not suited for controlling a continuous industrial process.

Likewise, instruments designed for continuous duty are not generally as accurate as instruments which can take a long time to measure moisture of a small sample. Also, sensors designed for continuously measuring small changes very accurately may be less accurate and less repeatable when they must measure very different conditions in a short period of time. For instance, an aluminum oxide dew point sensor is an accurate device for continuous measurements in a narrow range of low dew points, but if it must measure an environmental simulation chamber cycling rapidly between dry and saturated conditions, its accuracy and repeatability will suffer.

The operating environment for a sensor may also influence the selection. For example, if there are a number of condensible gases in the air other than water, a condensation-type hygrometer may read a dew point of a contaminant rather than water vapor. Likewise, if there are corrosive vapors in a hot airstream, an aluminum oxide sensor may well react chemically with the corrosives, altering its electrical characteristics and hence the water vapor reading.

Strategies to deal with difficult operating environments are as diverse as the possible circumstances, and instrument manufacturers can provide guidance in each case. In general, if the environment is extreme in any sense — near saturated or near zero humidity, or very hot or very cold, or highly corrosive — the system designer should be prepared to seek expert advice, and share all the circumstances of use with the advisor. Often, complex problems can be solved simply — for instance it may be more practical to specify a low-cost, replaceable sensor in a corrosive environment than to take extraordinary precautions to protect a more costly device.

Accuracy of measurement, short response time, wide measurement range and tolerance of extreme environments are not mutually exclusive characteristics in a humidity sensor, but when ***all*** are necessary, the device will be expensive.

Instrument Functions

There are four basic functions a moisture instrument can perform, including:

- Sensing humidity or moisture content
- Indicating the measurement
- Recording the measurement
- Controlling a dehumidification or humidification system

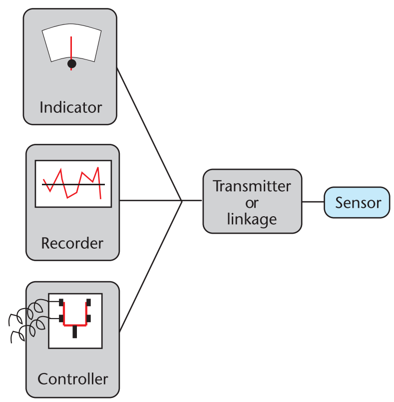


FIGURE 9.1

Instrument functions

Moisture and humidity instruments perform one or more functions, providing the designer with a wealth of alternatives which can be confusing until he or she determines which functions are necessary for the project.

FIGURE 9.2

Indicators vs. controllers

The dial indicator in the top photo shows relative humidity. But to control humidity, a different type of instrument must turn mechanical equipment on and off — which is the function of the humidistat in the bottom photo. Both instruments have the same type of sensor, but they perform different functions.

FIGURE 9.3

Indicator-recorder

A hygrothermograph measures and records the dry bulb temperature and relative humidity. Most such devices do not control either variable, they simply indicate its value.

As a measurement signal passes through different functions in series, inaccuracies accumulate, just as a story changes as it is repeated from person to person. This is because each function has four limits which affect the final result:

- Repeatability (precision)
- Range
- Response time
- Durability in the specified operating environment

As the number of instrument functions increase and their performance specifications tighten, the instrument becomes more costly and there is more potential for error. This is not an argument for single-function instruments, but since cost is always an issue, the designer should realize that if the instrument seems expensive or difficult to use, it may be because it has more functions than required, or because the specifications for each function may be too restrictive.

For example, a fertilizer storage shelter may have a pressing need for humidity control, since moist fertilizer can become as hard as concrete. But there may not be any need to actually know or to record the humidity. It may be enough simply to *sense* the humidity and *control a switch* which turns on the dehumidifier. That device is a simple commercial humidistat, costing less than $100 in 2002 dollars. If there is a need to *indicate* the humidity, the instrument will cost more, and a chart to *record* the humidity will in turn add further cost.

Purchasing several single-function instruments is an alternative to adding functions to the original instrument. A thermometer to measure dry bulb temperature and a plastic-element dial hygrometer are often less costly than an electronic instrument that combines both functions. The multiple-instrument alternative is especially attractive when the specifications for one function are much tighter than for another. For instance, in a Lithium battery dry room, the dew point is critical, and the temperature is much less so. The hygrometer must have a tolerance of ± 0.05 grains per pound because moisture affects the product, but the temperature measurement can accommodate a wider tolerance.

While using separate instruments for different functions is often less costly, it can also be confusing if two instruments read the same variable. For example, if a chart recorder has its own humidity sensor, it will never agree precisely with the reading taken by a separate humidistat that controls the dehumidifier. In fact, two apparently identical sensors will each have a slightly different response to the same environment.

Repeatability vs. Accuracy

Every instrument has its own accuracy, tolerances and response times, which are unlikely to coincide exactly with any other instrument. They will disagree much like equally intelligent people, who may describe the same event from different perspectives. Instrument discrepancies can be minimized by calibrating both instruments at the same time, in the actual range where both will be used. Where cost is more important than exact agreement between instruments, separate instruments may be an economical alternative.

It is useful to understand the relationship between accuracy and repeatability. Accuracy is the ability of the instrument to indicate the true value of the humidity. Repeatability is the ability of the instrument to indicate the original value when returning to the original humidity. If an instrument is highly repeatable, it can be made accurate through calibration. If it is not repeatable, the instrument can never be accurate. Recognizing this, instrument manufacturers generally define the repeatability of their products in their standard product data sheets.

Low-cost sensors are sometimes described in terms of "accuracy" alone. System designers evaluating such devices might want to ask the manufacturer for more detailed information about repeatability within the expected temperature and moisture range if accuracy is truly important in a given application. Repeatability, and therefore accuracy, is seldom uniform through the whole range of an instrument's possible operation. Generally, repeatability is better in the middle of the range than at the extremes.

Now we will examine the operating principles of the three types of moisture sensors.

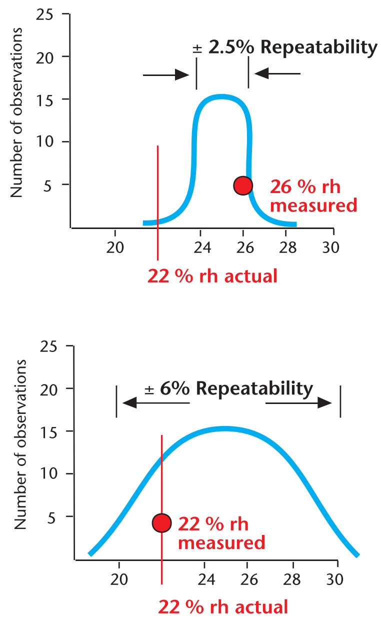


FIGURE 9.4

Repeatability vs. accuracy

The first instrument is not accurate at the moment, since it does not indicate the actual humidity value. but it can be made accurate through calibration, because it is repeatable — for any given actual value, it will consistently indicate a number within 2.5% rh of its original indication.

The second sensor happens to be accurate in this one case, but since its repeatability is poor, the next observation is unlikely to be accurate, and calibration will not improve its accuracy.

Relative Humidity Sensors

In general, relative humidity sensors are based on the change of mechanical or electrical properties of materials that result from changes in the surrounding humidity. We will look at a representative sample of such sensors individually.

Mechanical expansion hygrometer

Perhaps the oldest form of humidity measurement device is based on the fact that human hair changes length in proportion to the relative humidity of the surrounding air. The higher the humidity, the longer the

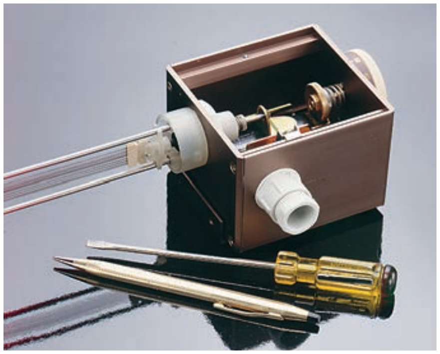


FIGURE 9.5
Expansion sensor
Most common humidistats sense humidity by the expansion or contraction of a hygroscopic material. In this case, the element is a flat bundle of animal hair, which responds to increasing relative humidity by lengthening. Other expansion sensors are based on plastic strips or filaments which respond like animal hair, changing dimension with relative humidity.

hair. Manufacturers generally use a bundle of hairs. This averages the individual responses of each strand, since different hairs responds at slightly different rates of expansion and contraction.

Sources of measurement error include the response of different hairs, and the response of the mechanical linkage which connects the hair to the indicating scale. An accuracy of ±5 to 7% rh is a common result for hair hygrometers, but like many other instruments, they can be more accurate in the middle of the relative humidity scale than at the very high or low ranges. Accuracy may be closer to ±2 to 3% rh between 40 and 60% rh at room temperatures. Outside of that range, accuracy will decline.

To improve accuracy, the device should be calibrated in the room where it will be used, and calibrated at a condition in the humidity range expected for the room. Designers should maintain a healthy skepticism concerning hair hygrometer readings, including those marketed as "certified". Often a small tap with a finger is enough to change a humidity reading by three percent, as the mechanical linkage can seize up over time. They are best used as a general indication of humidity rather than for important readings.

A contemporary adaptation of the hair hygrometer is the plastic expansion hygrometer. In this popular and economical instrument, the hair is replaced by a hygroscopic polymer like nylon, polyimide plastic or cellulose. Humidistats controlling residential furnace humidifiers, and many inexpensive industrial humidistats use this type of sensor. While the hygroscopic polymer is more uniform than human hair, the same advisory cautions apply — do not expect accuracy greater than ±7% rh, calibrate them regularly in the environment where they are used and one should not expect accuracy if wide swings in relative humidity are common.

Electronic expansion hygrometer

In the mechanical hygrometer, the change in expansion of material is measured and indicated by a mechanical linkage of gears, levers and dials. A modification of this concept replaces the linkage with electronics. Hair, plastic and in one case, a desert plant seed case is connected to an electronic strain gauge which measures the pressure exerted as the sensing element contracts. This is often an improvement over mechanical hygrometers, since electronics tend to be more repeatable than linkages, particularly over long periods of time.

Electronic capacitance sensor

Briefly described, a capacitor stores an electrical charge. In its simplest form, it consists of two metal plates separated by an insulator. If other factors are equal, the charge of the capacitor and the resistance it presents to a circuit depend on the insulating capacity of the separator between the plates, a property called its *dielectric constant*.
Capacitors used in most electronics are sealed because moisture changes the dielectric constant of the separator. If the capacitor is open to the air, its resistance changes with humidity as the separator absorbs and desorbs moisture. This provides the operating principal of the electronic capacitance sensor. Resistance of an open capacitor is measured electronically and converted to a corresponding relative humidity.

FIGURE 9.6

Capacitance sensor

Electrical characteristics of many materials can change with humidity. Electric circuits connected to the material can measure the change and convert the signal to a value for relative humidity.

The variable capacitor used in moisture sensors can be constructed with many different kinds of separators. Hygroscopic polymers and aluminum oxide are the most common. The ideal separator is one which has a very high dielectric constant when dry and very low when moist. The larger the difference, the easier it is to measure.

Like any sensor, there are some sources of error in capacitance sensors, principally in linearity of the signal and aging of sensor materials. Dielectric constants of even the best materials are not strict linear functions of relative humidity. The change in dielectric constant is smaller at high or low humidity extremes than in the middle of the range, so the signal must be linearized before it is converted to a relative humidity value. Otherwise, the signal underestimates the humidity changes at the top and bottom of the scale. Normally this is accomplished within the sensor circuitry, but linearization is sometimes accomplished at the destination rather than the source of the signal.

FIGURE 9.7

Resisitive sensor

These instruments measure the resistance of a treated polymer as it changes with increasing relative humidity. The bulk resistance rather than surface resistance is measured, which helps make the sensors quite reliable at high relative humidities.

Also, many of the separator materials best suited to capacitance sensors can change characteristics over time, particularly if they become saturated with condensation or have to measure airstreams with heavy chemical contamination. This does not imply that they are less suitable than other types of sensors — simply that like any other instrument, regular calibration can improve accuracy, particularly when the instrument is calibrated at the midpoint of its expected measurement range. For example, it would not be wise to rely on an instrument calibrated at 75°F and 65% rh if it must function accurately at 70°F and 10% rh — the moisture levels are too different for reliable measurements.

Electronic resistive sensor

The electrical resistance of certain polymers can change according to the water vapor they absorb, and the rate of absorption is a function of relative humidity of the surrounding air. In one such system, two

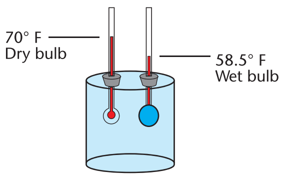


FIGURE 9.8

Wet & dry bulb thermometers
The difference between the wet and dry bulb temperatures of the air provide an indication of its moisture content. The wet bulb temperature is lower when the air is drier. When both readings are the same, the air is saturated — 100% relative humidity.

FIGURE 9.9

Sling psychrometer
Wet and dry bulb thermometers mounted on a swivel are spun in a circle, and the difference in temperatures noted. While inexpensive, using the device takes experience and many repeated readings to achieve accuracy beyond ±7% rh.

electrodes are coated with, and connected by a thick polymer which contains a quaternary ammonium salt. As the polymer absorbs water vapor, mobile ions are created in the material, reducing impedance between the electrodes.

Since the phenomenon takes place in the bulk of the polymer rather than only on the surface, this type of sensor tends to be more accurate and more stable than capacitive devices when measuring relative humidities above 90%. Capacitive devices, in contrast, have better sensitivity than resistive sensors at relative humidities below 15% since they detect surface absorption of small amounts of moisture more readily than the bulk polymer.

Psychrometric instruments

One of the most common methods of measuring relative humidity uses a pair of matched thermometers to sense the difference between the wet and dry bulb temperatures of the air. These instruments use two thermometers, one dry and the other covered with a wet wick. The wet thermometer shows a lower temperature than the dry one, because the evaporating water takes the heat required for its vaporization from the body of the thermometer itself. This lowers the temperature around the sensing bulb.

The amount of cooling depends on the rate of evaporation, which depends on the amount of moisture already in the air — the drier the air, the more cooling takes place, and the lower the wet bulb temperature will be. The wet bulb temperature drops until an equilibrium is reached between the heat loss due to evaporation and the heat supply due to heat transfer from the surrounding air. When the wet bulb and dry bulb temperatures of the air are known all other properties including relative humidity and dew point can be determined by reading a psychrometric chart as described in Chapter Two.

This principle is the basis of the common *sling psychrometer*, an inexpensive, lightweight instrument for measuring air moisture. The wet and dry bulb thermometers are mounted in a sling. After the wick on the wet bulb thermometer is wetted, the operator spins the sling rapidly in a circle. The water evaporates from the wick, and the wet bulb reading drops until all the water evaporates — then the wet bulb temperature begins to rise towards the dry bulb temperature. Generally, the operator repeats the process five to seven times to make sure he has seen the lowest wet bulb reading, which will be the most correct.

There are many limitations to the technique since there are so many uncontrolled variables in using a sling psychrometer. The operator must read the thermometer accurately at the point of maximum wet bulb depression. And there is no guarantee that the water on the wick lasted long enough to read the true wet bulb — it may have evaporated entirely before the evaporation rate came into equilibrium with the heat transfer rate. The wick must be absolutely clean — even the natural skin oils from light handling can change the wet bulb reading by two or three degrees Fahrenheit. The wick must be completely and uniformly wetted. The rate of spin — therefore the air velocity across the wetted wick must be a minimum of 600 feet per minute. And the water must be very pure — distilled water is recommended to avoid contamination of the wick. In short, the instrument is useful for approximations, but only the most experienced and meticulous operators can obtain consistent results.

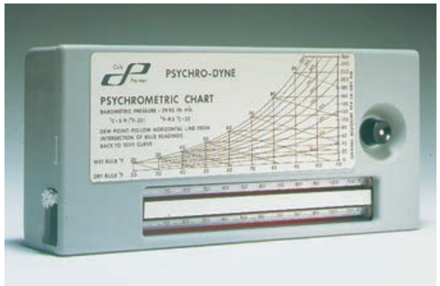


FIGURE 9.10

Aspirated psychrometer

This instrument improves on the sling psychrometer because air velocity and water evaporation are constant.

Some instruments use wet and dry bulb measurements under more controlled conditions. The *aspirated psychrometer* is an example. Wet and dry bulb thermometers are mounted in a case with a battery-powered fan and a liquid reservoir to feed water to the wet bulb wick. The fan draws air first across the dry bulb and then across the wet bulb at a constant rate. Because the wet bulb wick has a reservoir, the temperature stays relatively constant, making it much easier to read. Wick cleanliness is still very important, and thermometer accuracy and readability affect the accuracy of the reading.

ASHRAE (The American Society of Heating, Refrigerating, and Air Conditioning Engineers) suggests that with careful operation, aspirated psychrometers can yield accuracy within +5% relative humidity at dew points above 32°F.

The largest errors occur when the difference between wet and dry bulb temperatures is the greatest — low relative humidities. Also note that the error is always positive, never negative, which is to say the technique will always overestimate the amount of moisture in the air rather than read below the true moisture content. This is because the source of the error will always be inadequate cooling due to limitations in evaporation rates, and a higher wet bulb temperature reading overestimates air moisture content.

Absolute Humidity Sensors

Some of the relative humidity sensors described previously also display specific humidity readings, which they derive mathematically by combining dry bulb temperature with relative humidity. The sensors described in this section, however, measure absolute humidity directly.

Gravimetric train

This instrument is not used to control or indicate moisture in dehumidified rooms, nor is it found in instrumentation catalogs. However, it is useful to understand because it is the standard for determining accuracy of all other moisture measurement devices. When U.S. instrument manufacturers speak of “measurements traceable to National Institute of Standards and Technology”, they are referring to a chain of measurements that ends at the gravimetric train.

In the gravimetric technique, a technician weighs a small amount of a powerful drying agent, typically phosphoric anhydride (P_2O_5). The drying agent is exposed to the sample of moist air, from which it removes all the water vapor. Then the drying agent is weighed again. The difference in the two measurements is the weight of moisture removed from the air sample. As one might imagine, the procedure is highly complex and time-consuming. The apparatus fills a room, and operation requires several skilled technicians. A single measurement can take hours, days or weeks to perform — the lower the moisture, the longer the time required to take the measurement.

The National Institute provides dew point certification of instruments, which are then used to calibrate other devices used in commerce and industry. But for this purpose, the Institute uses a “Two-pressure Generator”, which creates air at a controlled dew point which is then sent to the equipment to be calibrated.

FIGURE 9.11

Dewcup

When condensation forms on the polished surface of the inner container, an observer estimates the air dew point by measuring the temperature of the liquid inside the cup.

Condensation hygrometers

A more common instrument to measure specific humidity is based on the dew point phenomenon. If the temperature and total pressure of the air is known when its moisture condenses, the specific humidity is known as well. At constant pressure, each dew point has only a single value of vapor pressure and moisture content.

In its simplest form, called a dewcup, the condensation hygrometer duplicates the experiments of Charles LeRoy in 1751. A thin, polished metal container is filled with liquid, which is progressively cooled by adding ice or dry ice. A thermometer measures the liquid temperature, which is assumed to be the same as the air temperature next to the outside surface of the container. When condensation forms on the outside of the container, the observer notes the liquid temperature, which is assumed to be the air dew point temperature. Although far more accurate and repeatable than taking wet and dry bulb readings, dewcup readings are equally operator-dependent. There are many uncontrolled variables, including the observers vision, surface illumination, possible surface contamination, airflow past the surface, temperature uniformity of the liquid solution, and re-evaporation of condensate. These all make it difficult to achieve accurate dewcup readings in the field.

FIGURE 9.12

Electronic condensation-based hygrometer

Such instruments automate the condensation observation and surface cooling through electronics, making the technique exceptionally accurate and useful for calibrating other instruments.

Manufacturers have improved on this basic principal by automatically controlling the cooling of the surface, its illumination, the airflow rate and the coupling of the temperature sensor to the surface. Controlling these variables make optical dew point observations far more repeatable.

In addition, the observation of the condensation surface is automated through electronics. A light-emitting diode shines on the cooled surface. The reflected light is received by a sensitive photocell. When the photocell senses a reduction in illumination, the instrument assumes moisture has condensed on the cool surface, scattering the light. The instrument controls the cooling mechanism to maintain the surface exactly at the air dew point, so it can be displayed continuously.

The electronic dew point hygrometer is held by many to be the most consistently accurate device for continuous measurements across a broad range of conditions, and is often used to calibrate other instruments. However, there are a few sources of measurement error. One is contamination on the chilled surface. Most equipment is designed to compensate for normal light dusting, but if corrosive or hygroscopic material settles on the polished surface, the reading becomes less accurate. Also, there is a limit to the amount of cooling available in each device, so if the temperature is very high and the measured dew point very low, the user must be careful to specify adequate cooling capacity for the expected air temperature range.

FIGURE 9.13

Aluminum oxide sensor

Water vapor is strongly attracted by aluminum oxide, which makes such sensors particularly useful for measuring low dew points which occur at high temperatures.

Aluminum oxide sensors

These instruments operate on the same principle as capacitance sensors used for indicating relative humidity. The capacitance of a thin film of aluminum oxide varies with adsorbed water vapor. Sensors which indicate dew point rather than relative humidity are often optimized for operation at very low relative humidities, and can be quite accurate. A typical specification is ±3°C of dew point. Aluminum oxide is very hygroscopic. This is an advantage when measuring very low relative humidities, and such instruments are often used to measure dew points of -40°F at air temperatures above 300°F which comes from desiccant dehumidifiers used to dry plastic resins.

Some caution is necessary because the oxide binds water vapor quite strongly. If an airstream changes from very wet to very dry, the sensor may take a long time to respond to the change. One should not expect to take a reading downstream of a cooling coil and then immediately expect an accurate reading in a dry room held at 70°F and a -20°F dew point. Also, repeated wetting of sensors operating close to saturation can cause chemical changes in the oxide film, which may change response characteristics. These limitations can be mitigated by keeping the sensor very dry, and by taking intermittent readings of the air in question, so the sensor is always "approaching" the correct reading from below rather than above the true value.

Salt equilibrium sensors

Another specific humidity sensor uses the equilibrium characteristics of a hygroscopic salt. Lithium chloride is dry when the surrounding air is approximately 11% relative humidity. When the salt is not in a liquid solution, its resistance is very high.

FIGURE 9.14
Lithium chloride salt equilibrium sensor
Sometimes called a dewcell, this sensor is based on the moisture equilibrium characteristics of a hygroscopic salt. A variable heater keeps the salt dry, and the salt temperature is directly proportional to the specific amount of moisture in the air.

A saturated salt sensor heats a layer of lithium chloride until it is completely dry, as determined by measuring its electrical resistance. The temperature of the salt is measured, which can then be converted to the specific humidity of the air. For instance, if the salt must be at 100°F to be dry, the sensor assumes the moisture content of the air must be 26 grains because 26 grains represents 11% rh at 100°.

Such instruments are quite repeatable as long as the salt layer is clean and uniform, and the temperature sensor is stable. The sensors have a reputation for requiring maintenance, because their lithium chloride layer can be depleted if the sensor loses heat at saturated conditions — the salt becomes saturated and drips off the sensor. Although other sensors may have the appearance of needing less attention, practicing engineers have found that all sensors need regular attention and calibration for accurate results.

Electrolytic hygrometers

These instruments are generally used for measuring extremely low moisture levels — dew points below -50°F. They use an electric current to break water molecules into their constituent atoms of hydrogen and oxygen.

A very small and carefully controlled airsteam is directed across a sensor coated with phosphoric anhydride, a powerful desiccant. Two electrodes wound through the desiccant electrolyze the absorbed water vapor. The current required to keep the desiccant dry is directly proportional to the number of water molecules originally in the air sample. The technique is very accurate at low dew points, where it would be difficult to use other methods because the amount of moisture is so small. Electrolytic instruments can distinguish between 100 and 125 parts per million by weight — less than one quarter of one percent relative humidity at 70°F.

Since electrolytic sensors act on any molecules absorbed by the phosphoric anhydride, the reading can be distorted by other contaminants. Also, since a higher airflow rate will bring a disproportionate number of water molecules to the desiccant, the sample flow rate must be carefully controlled.

FIGURE 9.15

Electrolytic hygrometer

These instruments break water molecules apart using electrolysis. The necessary current is proportional to the number of water molecules in the air sample.

Material Moisture Content Sensors

In many dehumidification applications, the goal of the project is to either dry a solid material or to prevent moisture regain. Either project can require a measurement of the amount of moisture in a solid rather than in the air.

There are several methods for measuring moisture content of solids. In all cases, good sampling methodology is essential, and most require knowledge of the material's behavior when perfectly dry. Also, these methods measure the change in material properties from a known dry material, so if the identity of the material itself is in question, it will be difficult to determine moisture content. The one exception is a laboratory technique known as Carl Fischer titration.

Coulombic Karl Fischer titration

This method is based on measuring the electrical current required for the reaction of water with iodine. The reaction is very predictable, requiring 10.71 coulombes for every milligram of water present in a solution of water, iodine and sulfur dioxide.

FIGURE 9.16

Coulombic titration

Moisture is extracted from the material using a solvent solution. The instrument then measures moisture by the current required for its reaction with iodine.

FIGURE 9.17

Infrared analyzer

Water absorbs infrared energy. The difference between the energy absorbed and reflected by the surface of the solid is measured by the analyzer, and displayed as the corresponding percent moisture value.

A representative sample of the solid product is placed in the solution, which also contains non-reacting hygroscopic solvents to extract the water from the material. Then the current required for the iodine reaction is measured. The current is proportional to the water extracted from the material.

While the method is quite accurate, it requires time for the solvents to extract all the water from the material, so coulombic titration is generally used to develop baseline data on materials, which is then used to calibrate other instruments.

Infrared absorption

Water absorbs infrared radiation quite efficiently. The more moisture in a material, the more infrared radiation it will absorb. If the absorption characteristics of the dry material are known, the increased absorption of wet material is directly proportional to the increase in moisture.

Making use of this effect, an infrared emitter shines on solid material and a photocell measures the reflected energy in two wavelengths — 1.9 microns, for instance, which is absorbed by water, and 1.7 microns, which is not. Electronic circuits compare the strength of the two signals, and the difference is proportional to the water content of the material. While the absorption rate can vary with temperature, most such instruments are equipped with temperature sensors and automatically correct their reading for this variation.

The technique is best used where the product layer is both homogeneous and thin. Large bulges in product contain more moisture because they have more mass. Also, if the top layer exposed to the infrared energy has a higher or lower moisture content than the layer below, there can be a false reading of moisture content of the total sample. Product structure and color can also affect absorption, so optimum wavelengths are generally different for different materials. But when sensor arrays are engineered to a single, specific product application, remarkable accuracy is possible — 0.1% of true surface-moisture content in many cases.

FIGURE 9.18

Equilibrium sensor

Relative humidity sensors are used for fast estimates of product moisture. The sensor must be in "dead air" and in close contact with the product surface to insure reliable estimates.

Equilibrium moisture detectors

Sensors which measure relative humidity of air are also used to make approximate measurements of moisture in materials. Solids which contain moisture will release it if the surrounding air is very dry, and absorb more moisture if the air is humid. At some point between the two extremes, the product will be in equilibrium with the moisture in the surrounding air, neither giving up nor taking on more water vapor. The equilibrium point depends on the amount of moisture in the material, its particular structure and chemical composition, its temperature and the surrounding relative humidity.

If an observer knows the product is neither taking on or giving up moisture, and also knows the relative humidity of the air, he can estimate the amount of moisture in the material. Since the materials structure and chemical composition also influence the moisture equilibrium characteristics, there are separate tables for different materials that show the moisture content of a given product at different relative humidities.

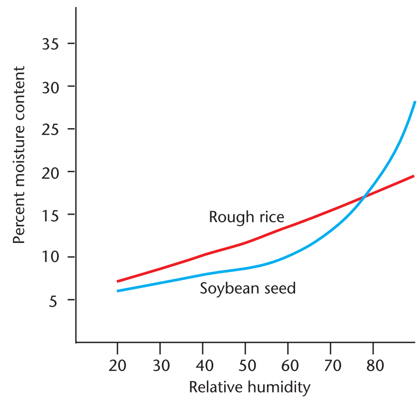


FIGURE 9.19

Moisture equilibrium

For every product moisture content, there is a corresponding relative humidity in which the product will neither gain nor lose moisture. If the relative humidity at the product surface is known, the observer can estimate product moisture content.

There are important practical considerations about using a relative humidity sensor to measure product moisture. First, the sensor must be in air which is truly in equilibrium with the product. The sensor must be placed in non-circulating air very close to the product surface. For instance, in paper storage applications, the sensor is inserted deep into a stack between the sheets rather than simply read in the warehouse around the pallets. And in grain, the sensor is thrust deep into the bin rather than the space above the product. Also, the moisture estimate is only as good as the equilibrium data for the material in question. If the equilibrium data for barley is used for soybeans, there will be a significant measurement error. In fact, if the data for one corn species is used for another, the reading may be misleading.

Further, like all material moisture measurements, the usefulness of the technique depends on the uniformity of the material and the statistical validity of the sample. For example, even if grain starts out at a perfectly uniform moisture content, several months of storage in a large bin will redistribute moisture because of minute convection currents and temperature differences in the bin. The grain at the top of the bin may be several percent more moist than grain at the bottom. Several readings within a large batch are necessary to gain an accurate reading. Also, large product pieces can be moist on the inside and dry on the outside, or vice-versa. The relative humidity technique only measures surface characteristics of the material.

The equilibrium technique is frequently used where the real question is not the exact moisture content, but rather the change since the last reading. It allows a relatively fast method of monitoring changes when an observer knows and understands the probable behavior of the product.

FIGURE 9.20

Resistance sensor

Electrical resistance of solid materials depends on their structure, composition, temperature and moisture content. If other variables are known, changes in resistance correspond to different moisture contents.

Resistance moisture sensors

Solid materials containing moisture will generally conduct electricity. Their resistance to current flow is inversely proportional to their moisture content — the drier the material, the greater the resistance.

Resistance moisture detectors penetrate the material with two sharp needles which act as electrodes. A current passes through the material from one electrode to the other, and the observer notes the resistance reading. The resistance of the material when totally dry and completely wet has been pre-established, so the instrument reading indicates roughly what percent moisture remains in the material.

Different materials require different electrode shapes for proper moisture measurements. For instance, an electrode for paper can be made in the shape of two flat paddles which contact the paper at a fixed distance from each other. Electrodes for wood can be needles and for bulk powders and grain a probe can be designed with electrodes at the end of a long shaft.

Like other solid moisture measurement techniques, resistance instruments assume the observer has reference data which allows him to convert a resistance reading to a corresponding moisture value. Also, the instrument measures resistance between electrodes, which are basically at the surface of the material. If there are large, thick pieces to measure, the electrodes must penetrate deep into the material to avoid a misleading result.

Wood provides a good example, since moisture content of lumber is frequently measured with resistance instruments. The end-grain of lumber absorbs and desorbs moisture faster than the edge grain, and lumber on the outside of a stack may be much drier or wetter than boards in the middle. So, measurements should be taken at several points along a board; and several boards from different places in the stack should be measured if the observer must establish the average moisture content of the whole pile.

Finally, the resistance of moisture-laden materials increases with temperature, just as happens in a copper wire. So when the material is significantly higher or lower than 70°F, the observer must make a temperature correction to the moisture content reading.

Microwave absorption

Another absorption moisture measurement technique for continuous product flows uses the fact that microwaves are absorbed by water much like infrared energy. But where infrared techniques are really limited to surfaces, microwaves can penetrate into deeper thicknesses of material.

FIGURE 9.21
Microwave absorption
Like infrared energy, microwaves are absorbed by water. Microwave systems are used to measure deep beds, or used where product color variations might make infrared techniques less accurate.

A microwave generator is mounted on one side of the material. On the opposite side, a receiver measures the change in microwave amplitude and the extent of the phase-shift. The amplitude loss divided by the phase shift is proportional to the moisture in the material.

Since the weight percent of moisture in the material will also vary with differences in product density, a gamma-ray sender and receiver continuously measure product density and use the value to correct the moisture reading. Temperature also affects microwave absorption, so a third circuit continuously samples product temperature and makes the appropriate correction.

Microwave and infrared absorption techniques are equally accurate, but since infrared instruments are generally less costly, microwaves are typically used in deep beds or dark materials where infrared techniques have limitations.

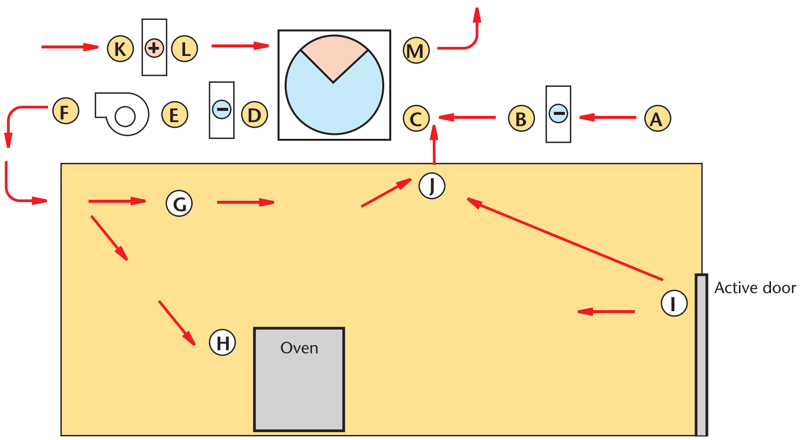
Radio frequency (capacitance) sensors

Since the dielectric characteristics of water are very different from most solids, measuring the characteristics of a given material provide a means of determining its water content. Water has a high dielectric constant — which is to say it is easily charged. But it also has a high dielectric loss, so it quickly loses that charge when the current is removed.

In this technique, a radio frequency generator and electrical oscillating circuit are combined so that they resonate. When a product is brought near the instrument, the moisture in the product absorbs energy from the radio frequency field. This weakens the oscillating circuit so its voltage peak declines in proportion to the amount of moisture in the field. Like other absorption techniques, radio frequency attenuation is affected by product temperature, so the circuit must adjust for that variable.

Ideally, the field should be presented with the most uniform and reproducible moist sample possible. Therefore, for best results the actual shape of the RF generator must be fitted to each application.

FIGURE 9.22
Radio frequency sensor
Moisture attenuates radio waves in proportion to the amount of water in the field. Such instruments measure this attenuation and display the reading as a moisture content value.



Sensor Location	Temp (°F)	Moisture (gr/lb)	Dew point (° F)	Moisture
A	95	120	72.5	48
B	55	65	55.0	100
C	69	40	42	38
D	109	10	12	3
E	55	10	12	15
F	58	10	12	14
G	60	15	20	20
H	82	15	20	9
I	70	35	39	32
J	72	15	20	12
K	95	120	72.5	48
L	250	120	72.5	<1
M	120	258	95	49

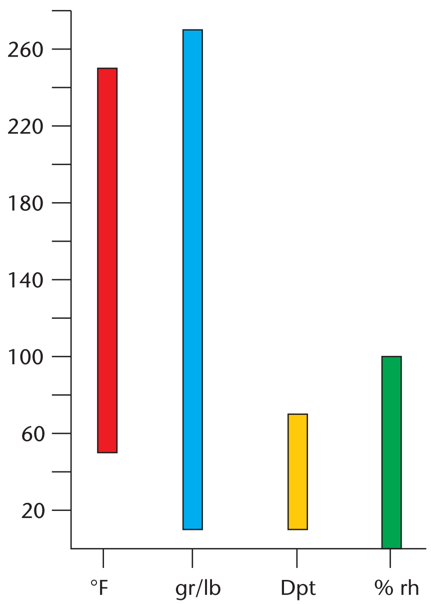


FIGURE 9.23

Sensor placement and selection

Different operating environments and expected measurement ranges suggest different sensor selections. For example:

A & K. Measuring weather air calls for a sensor which tolerates high humidity-including condensation. Of all the sensor points in this diagram, this location calls for the broadest range of measurements.

B. The air immediately following a condensing cooling coil must tolerate condensation and liquid water droplets. In this location, a temperature sensor may provide adequate information about humidity since the air will be saturated for much of the year.

C. Temperature and humidity levels at this location are comparatively easy to measure. The problem will be to make sure the air is well-mixed at the sensor location.

D. Immediately downstream of a Honeycombe® desiccant dehumidifier, the air is highly stratified in both temperature and moisture, the average temperature is high and the humidity very low. An aluminum oxide dew point sensor might do well in this location as long as the air is well-mixed before the sensor. The typical measurement at this point is well outside the range of relative humidity sensors.

E & F. Taking a humidity measurement downstream of a fan or dry cooling coil is easier duty than immediately following a dehumidifier, but on system start-up, the coil may well run at saturation for a time, as it will if the dehumidifier should stop. The sensor must tolerate saturation in this location, and good air mixing after a cooling coil is also essential.

G, H, I, J. Note how air distribution and concentrated heat and moisture loads can affect the conditions immediately surrounding a sensor. Perfect mixing throughout the room is not possible, so sensors should be placed away from obvious load extremes, unless control at those locations is essential.

M. The reactivation air leaving a dehumidifier seems moderate in terms of relative humidity, but when the unit shuts off, the massive amount of moisture in the duct will condense as the air cools. This is an extremely harsh environment for a humidity sensor — high temperature, intermittent condensation and possible corrosives in the airstream. Most systems require only temperature measurements at this location.

General Observations

Sensor placement

Moisture and humidity sensors are only as useful as the results they achieve in practice. Product specifications are meaningless unless the units are properly applied. Most common difficulties with moisture sensors are avoided by calibrating them in the environment where they will be used, and within the range of expected measurement. The second most common problem is poor sensor placement.

In a humidity-controlled space or product, there may be a wide range of possible moisture conditions. Figure 9.24 illustrates these points. The caption for that figure offers some suggestions on sensors for different locations.

In taking measurements of product moisture content, the same caution applies. Recognize that moisture content is not necessarily uniform across the product surface or through its depth. If accuracy or uniform dryness is important, it will be essential to take several readings within the product to understand its moisture characteristics.

Sensor placement is even more important than sensor accuracy. For instance, a dehumidifier installation to protect steel from rusting failed to prevent the problem. The technician investigated, and found the dehumidifier was located 75 feet from the storage racks, but the humidistat controlling the unit was mounted near the dry air discharge. The dehumidifier kept the discharge duct free of rust, but $50,000 worth of steel had to be reworked before the humidistat was relocated onto the storage racks.

Measuring moisture below 10% relative humidity

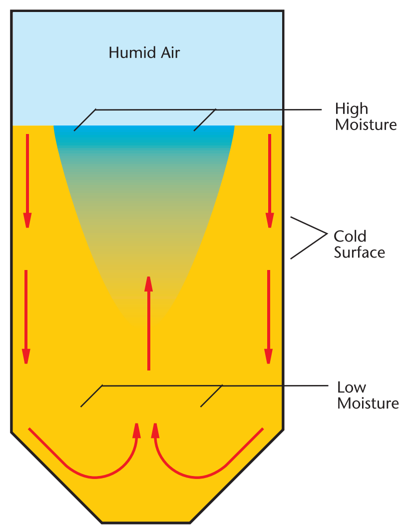
Cautions concerning sensor placement are especially important when the control level is very low, because moisture differences can be large between different points in the space. For instance, people's respiration will raise moisture levels around work spaces, and the air outlet from the dehumidification system will be exceptionally dry. Additionally, since there is very little moisture in the air, an instrument may take longer to register a humidity change.

For instance, the moisture difference between 50% relative humidity and 55% at 70°F is six grains per pound. However, in a room manufacturing surgical sutures, the difference between an acceptable -20°F dew point and an unacceptable -15°F dew point is less than 0.06 grains per pound, which is 100 times less than the difference between 50 and 55% relative humidity. This means the sensor may take longer to register the change. The instrument must be much more sensitive to begin with, and details which might appear minor become more significant — such as the sampling system.

In many low-humidity rooms, a costly instrument is installed in a central location, and an air pump pulls several samples from the room through a manifold, so several points can be measured in turn. But at

FIGURE 9.24

Moisture migration in storage
Small differences in temperature cause convection currents in storage bins. Moisture can be carried along, creating large differences in moisture content in a product which was originally quite uniform.



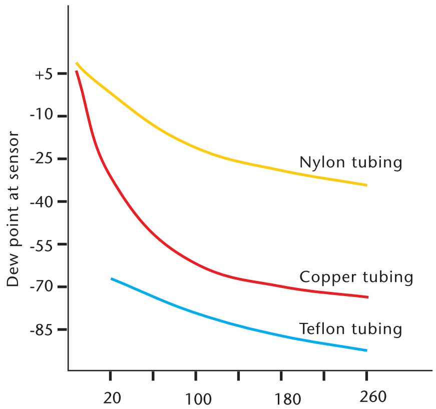


FIGURE 9.25

Sampling tube dryout time
At very low humidity levels, small variables can distort moisture readings. This shows how long it takes to dry tubing used for humidity sampling systems. Until the tubing is dry, there is no possibility of accurate measurements of room humidity at these dew points.

Because of this problem, instrument manufacturers use stainless steel tubing exclusively when accurate measurements are required below -10°F.

very low levels, it may take many hours or even days for the sample tubing to dry out so it does not distort the room reading. Figure 9.25 shows how a metal sampling tube can dry within hours, but nylon tubing may take days to dry low enough for dew point readings below -40°F. For this reason, instrument manufacturers generally insist on stainless steel sample tubing for measuring below -20°F dew points.

Environmental chambers

One of the most difficult tasks in moisture measurement is indicating and controlling humidity in an environmental simulation chamber. Such devices create a very wide range of temperature and moisture conditions, and they may change frequently and rapidly between different points. This is very heavy duty for moisture measurement instrumentation, and if accuracy is essential, it is important to calibrate the sensors regularly. Also, when the sensors must span a wide range, it is best to calibrate them at several points within that range for best results.

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10

MINIMIZING COSTS & MAXIMIZING BENEFITS

Dehumidification equipment and systems can provide very large benefits for a wide variety of applications — a fact which has been discussed in Chapter 4. However, there is no benefit without investment, and in many cases, the investment in dehumidification equipment can be substantial. The design engineer and equipment owner are quite naturally concerned with minimizing the equipment investment, and maximizing its benefits.

The costs and benefits of any expenditure are never absolute — they are relative to the costs and benefits of other courses of action. The most common alternative to a dehumidification project is making no investment at all — which in many cases is the most costly of all possible courses of action.

The first section of this chapter offers suggestions on how to identify and quantify economic benefits of a dehumidification project. The second section provides ways to keep project and operational costs to a minimum.

Identifying & Quantifying Economic Benefits

Economic benefits of a dehumidification project are as diverse as the number of applications for the equipment. But in general, they fall into four principal categories:

- Operational cost reduction
- Capital equipment cost reduction
- Product quality improvements
- Operational response improvements

All of these translate directly to either improved profits, or the ability to reallocate expensive resources to other uses. The effect of economic benefits multiplied over the 20-year life of dehumidification equipment can be quite astonishing. We will look at each of the areas in turn.

Operational cost reduction

Global competition has put a premium on keeping operational costs to a minimum. Dehumidification projects often assist in this goal. For example:

Interruption costs

Many dehumidification projects are initiated to reduce interruption costs. Where capital equipment is a large cost of operation, it must be in use constantly to assure profitable operation.

When a ship is in drydock for painting, the cost of "demurrage charges" can be over $20,000 per day. A dehumidification project to speed painting may save several days or even weeks of these charges. Semiconductor and pharmaceutical manufacturing is often so sensitive to moisture that excessive humidity forces a halt to operation. Lost production time can cost over $40,000 per day.

Businesses often invest in "interruption insurance" to cover the lost production in case of fire, flood or other disaster. Again, the cost to the insurance company can exceed $10,000 per day. Using a dehumidifier to speed drying of an office or production equipment can return the business to operation quickly, avoiding very large insurance claims, and speeding resumption of profitable business operation.

Production freezers must occasionally halt operation to defrost machinery and conveyors. If this happens in the middle of a shift, dozens of workers are made idle. The cost of labor and capital cost of

FIGURE 10.1

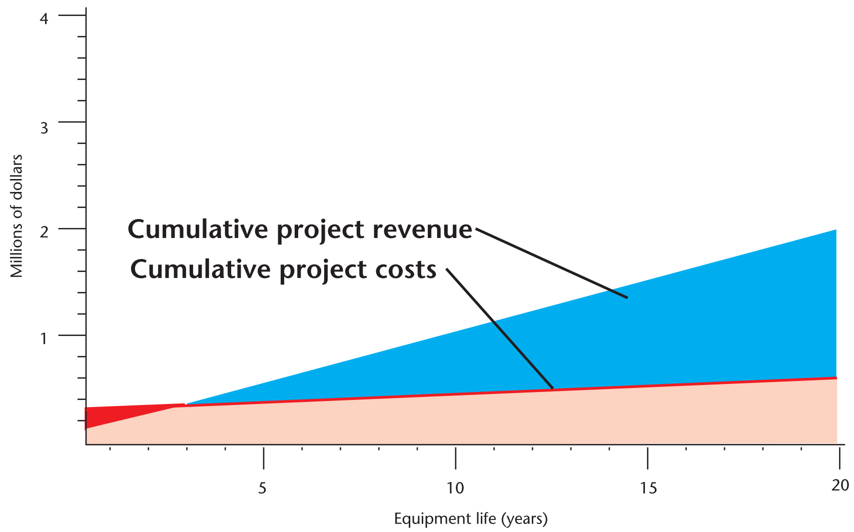
Dehumidification project costs & benefits

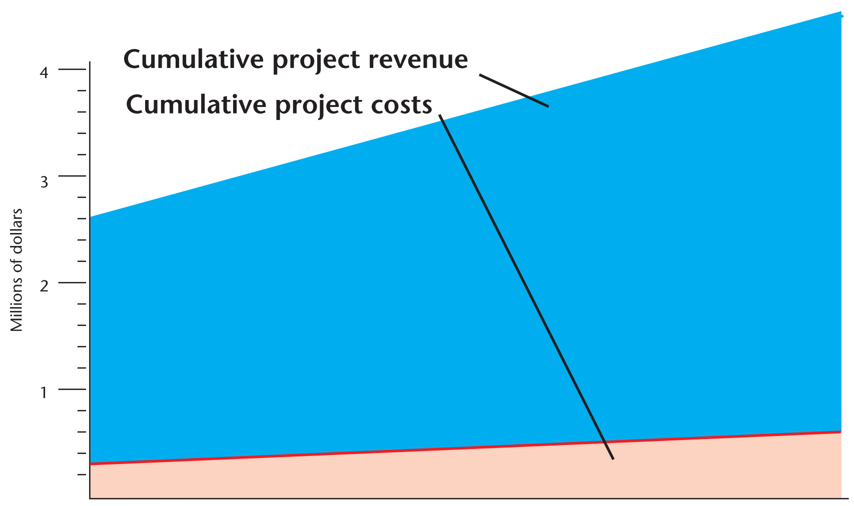
Dehumidification projects can present an exceptional investment opportunity, often recovering their costs in a short period.

These charts illustrate two ways of looking at the same project - with and without consideration of reduced capital costs.

The chart above shows the costs and benefits of a project to dehumidify a plant producing injection-molded, PET plastic preforms for beverage bottles. Dehumidification doubles processing speed, which recovers the cost of the project in less than three years.

The chart below shows the same project, but considering the costs avoided. The dehumidifier has doubled the plant capacity, which avoids the need for an entire second injection-molding plant.





equipment does not decline, even though there is no production. A dehumidifier which allows a manager to schedule defrost may save over $1,000 per hour of downtime.

Reducing re-work

If material becomes damaged by high humidity, the product or machine may have to be re-worked before it can be used or sold. In one classic case, the edges of coated steel coils corroded in storage, damaging the electrical properties of the steel for use in transformers. The dimension was critical, so it was not possible to simply grind off the corrosion. Coils had to be re-slit to a different dimension, loosing material and disrupting production. The cost saving from dehumidifying the warehouse was $250,000 per year.

Reducing energy costs

Desiccant dehumidifiers can utilize low-cost sources of energy to remove moisture from air, which can save money compared to other methods. In supermarkets, refrigerated cases remove moisture from the store, but not efficiently. Desiccant units can save over $30,000 per year in store energy costs.

Desiccant units are often combined with vapor-compression cooling systems to provide operating efficiencies. Refrigeration systems work more efficiently at high evaporator temperatures. When a desiccant dehumidifier feeds a cooling system with dry air, the cooling system can be set to a relatively high temperature because it need not dehumidify the air. This improves the operating efficiency of the refrigeration system, saving thousands of dollars each month on large systems.

Reducing need for skilled labor

In many situations, labor costs for maintaining equipment are high because of the effect of humidity. For example, aircraft avionics can change calibration with minute corrosion of circuits and connectors. The personnel needed to re-calibrate such instruments are scarce and expensive. Controlling humidity can free this resource, saving thousands of dollars per month in labor costs.

Maintenance costs

Maintenance budgets are often based on past practices that may be unnecessary with proper humidity control. For example, it can cost over $1,000,000 to blast and paint a marine tanker or process chemical plant. A dehumidifier improves coating adhesion, which can double the life of the coating. This may mean a maintenance cost saving of several million dollars over the 20 to 40 year plant or ship's useful life.

Packaging machinery processing hygroscopic material like candy, fertilizers and powders must frequently be cleaned by maintenance personnel. The cost of the operation gets lost in maintenance budgets, but it can be quite substantial. A dehumidification project which reduces cleaning time also reduces operational costs.

Reduced worker's compensation insurance costs

Refrigerated warehouses with icy floors can have larger than normal costs for worker's compensation claims. A dehumidification project can reduce such uncontrolled costs, and benefit the operation through improved worker morale.

Indoor air quality problems in some buildings stem from microbial growth in cool, moist air distribution systems. Drying this air with a desiccant dehumidifier can reduce the problem, saving the cost of sick time and worker's compensation claims that result from humidity-related health hazards.

Reducing cost of capital investments

Many organizations are limited in their access to funds for capital expenditures. While dehumidification projects generally fall into this category, such equipment is often far less costly than other capital expenditures to accomplish the same goal. For example:

Reduced need for plant expansion

Some dehumidification projects speed production enough to eliminate the need for a complete second factory. For instance, dropping the coolant temperature in the "chill box" of a photographic film coating line can double the coating speed, essentially doubling production from the same equipment. A dehumidifier that prevents condensation on cold surfaces allows this capacity increase, which avoids the need for a second coating line. Coating equipment may cost between $10 and $50 million to install, without including the cost of additional personnel and operating expenses.

Equivalent productivity improvements are possible in plastic injection molding operations, which have similar costs for new plants and equipment. A dehumidification project for existing equipment can avoid both the cost and time required for major plant expansions.

Reduced equipment replacement costs

In many industries, it is common practice to replace equipment on a regular basis. However, the cost of such policies can be difficult to bear when access to capital becomes limited. For instance, hotels and motels assume that mold and mildew as well as normal wear and tear will require replacement of room furnishings on a regular basis. Preventing mold and mildew has allowed certain chains to extend the useful life of furnishings, reducing the need for capital expenditure.

Military equipment provides an even more dramatic example. Rusted ammunition, humidity-damaged missile motors and corroded airframe components can force premature replacement of multi-million dollar supplies and equipment. Armed forces do not always have the resources to re-purchase such costly gear. A dehumidification project which eliminates these problems can create a major opportunity for resource reallocation.

Reduced HVAC system installed costs

Standard heating, ventilating and air conditioning systems can increase in cost when humidity control is important. For example, if a building has only a few spaces which must be humidity-controlled, but a central chilled water system serves all spaces, it will be necessary to run the chiller at a low temperature to provide dehumidification. Adding a desiccant dehumidifier to control only those humidity-sensitive operations allows the designer to use a much smaller chiller to control sensible temperature and dehumidification for comfort in the balance of the building, for a net installed system cost savings.

Also, in process applications, solvents are often used to speed drying rates for products. Investment casting operations provide a good example. Either solvents or water can be used in ceramic casting slurries that form the investment casting molds. But if solvents are used, an expensive emission-control system is necessary to comply with air pollution regulations. Using water-based slurries — with a low-humidity level to speed drying — a plant can avoid the high equipment and operational costs of emission-control systems.

Reduced commercial building retrofit costs

In recent years, businesses have made large investments in office automation in the form of personal computers, printers, copiers and FAX machines. In many cases, this has doubled the sensible heat load inside commercial buildings. Some owners have also concluded they must increase outside air ventilation rates to avoid loss of worker productivity through health problems. This also increases the total latent and sensible heat loads on older buildings, overloading the original mechanical systems beyond their design limits.

Desiccant dehumidifiers are used to off-load the latent heat load from the system by pre-drying the ventilation air to very low levels. The dry air has a large capacity for dehumidification, which means the existing ductwork does not have to be replaced to gain latent capacity. Also, since thermal energy can be used to regenerate the desiccant and remove the latent heat load, the owner can often avoid the large cost of upgrading the electrical distribution system to add cooling capacity.

Improved profits through improved quality

The value of many products and services is increased through the use of dehumidification equipment. This value can be converted to higher profits, which may not be achievable under other circumstances, or if possible, may cost more than using dehumidification equipment. Some examples include:

Improved market image through product appearance

Premium chocolate products have a fine, polished surface for maximum product appeal. These are worth more to the consumer than those with a blotchy finish caused by condensation in cooling tunnels or in coating pans. The difference can mean millions of dollars in profit opportunity for manufacturers who invest in dehumidification technology. Also, the cost of marketing to build brand image can be vast, and less than optimum product appearance can make these expenditures useless.

Reduced scrap rate

In some production operations, it is not important to manufacture high-quality goods. However, many international manufacturers have the opinion that “quality is free”. This is to say that reducing scrap and smoothing out manufacturing with production tooling investments pays its cost back very rapidly. Like other tooling investments, dehumidification projects can be assessed against the cost of the scrap they eliminate.

For example, semiconductor manufacturing is sensitive to fluctuations in thousands of production variables, one of which is atmospheric humidity. Even a 2% improvement in chip yield due to improved humidity control can mean thousands of dollars in reductions in scrap costs in a single day.

Higher product value

The mechanical, chemical or biological properties of some products can be enhanced through the use of dehumidification equipment. For instance, enzymes used in food processing and bio-engineering applications deteriorate with heat. Using a dehumidifier to dry enzymes at low temperatures allows the material to retain full potency. This means that a given amount of the enzyme will accomplish more work than material dried using high temperatures. The more potent enzyme is worth more in the market than the less potent material.

The germination rate of seeds also depends to large extent on drying temperature. Seeds dried at low temperatures using dehumidifiers germinate at a higher rate, so they are worth more than seeds dried at high temperatures. The value of spices, tea and other high-value agricultural products can be enhanced in the same way. The increased profit potential is a direct benefit of the dehumidification equipment. Such equipment can provide over 20 years of higher profits for product drying operations.

Improving operational responsiveness

Many companies operate in a business environment which puts a premium on fast response to sudden changes in market demand. Dehumidification projects can contribute to fast response, which has a direct relationship to improved profits. For example:

Power generation

Electrical demand can vary widely throughout the year, and even within a month or week. Producing excess power is not profitable, so certain peak load generators are often shut down. However, key components can corrode when out of use. Dehumidifiers which prevent corrosion without the need for greases or vapor-phase inhibitors allow a generator set to be instantly restarted at full capacity in response to changing load conditions, which avoids high costs for peak power purchased from other sources for short periods.

Un-scheduled maintenance of civilian aircraft

A large proportion of maintenance problems in aircraft are humidity-related, including avionics failures, structural corrosion and turbine-blade and bolt corrosion. The cost to pull an aircraft out of service is very large, particularly when there is no replacement aircraft available. Dehumidification that prevents even one non-scheduled maintenance event can greatly improve operational profitability.

Military mission-capability

At any given time, some percentage of military equipment is not fully mission-capable. Dehumidification projects to increase this availability mean fewer tanks, planes or artillery pieces can perform the same mission with great reductions in both capital expenditure and maintenance costs. Improved mission-readiness also has the obvious advantage of saving lives in times of conflict.

Crop storage

Most agricultural products are worth less at time of harvest than later, when supplies have been depleted. Protecting such products with dehumidifiers until market demand is high increases the product value greatly in excess of the cost of the equipment even in a single season.

Minimizing costs

Assuming everything possible has been done to maximize the benefits of the dehumidification project, the system designer's attention shifts to insuring the project will use the minimum resources necessary to achieve the intended benefits. There are two areas of concern — first cost and operating cost.

Often the designer must make difficult decisions when the goal of low operating cost may conflict with a limited project budget. It is painful to be presented with a situation where the organization simply cannot afford the cost of an additional control which might pay back its cost within a month of installation. However, dealing with those situations are beyond the scope of this handbook. Our discussion will center on what can be done — once resources have been made available — to make sure overall costs are minimized.

Minimizing first cost

In principle, first costs can be minimized by removing only the minimum moisture necessary to achieve the desired benefits, and by removing it in the most efficient manner possible. In practice, this means the designer should concentrate on:

- Minimizing moisture loads
- Optimizing the control levels and maximizing tolerances
- Drying outside air before it enters the controlled space
- Combining refrigerant and desiccant-based dehumidification equipment

Minimizing moisture loads

The refrigerated warehouse example in Chapter 7 provides an example of how to minimize moisture loads to minimize first cost. In that example, the designer is faced with the fact that existing warehouse loading door practices will create an enormous moisture load. The project could never hope to proceed because the company could not afford the equipment no matter how great the potential benefit. In the example, the designer makes a change in the maximum allowable time a door can be open, and the moisture load drops to less than half the original calculation, reducing the capital cost by more than 50%.

The example is fictitious, and most designers do not have direct control over door activity, but the principle is sound — when capital is limited, the designer should go through the moisture load calculation sheet very carefully. It may be possible to limit door activity, reduce exhaust fan airflow or tighten the building envelope at very low cost compared to building a larger system to remove a large moisture load.

Laboratory exhaust hoods are particularly costly because they generate a requirement for dry make-up air to replace the exhausted air. Often, the open area of the hood can be reduced without compromising experimental or process requirements.

This is a difficult tightrope for the designer to walk — one must not be overly optimistic about *possible* future changes to existing buildings and established practices. Still, some large loads can be easily reduced by low-cost application of building sealants, aluminum foil tape and vapor-retarders.

Optimizing control levels and maximizing tolerances

As the moisture control level reduces, the engineer should be more and more careful not to over-specify the dew point or the tolerance. This is particularly true at dew points below 0°F (5.5 gr/lb). Even if air could be delivered at zero moisture content, the amount of dry air necessary to remove a given moisture load increases as the control level goes down. This means much larger dehumidification systems for the same load.

For example, if the space has an internal moisture load of one pound per hour, and the control level is 5.5 gr/lb, it would require 283 cfm of perfectly dry air to remove the load and to maintain control. If the control level is reduced to a -15°F dew point (2.5 gr/lb), the system will need 622 cfm, and at a -27°F dew point (1.2 gr/lb), the airflow must be 1296 cfm — all this represents a larger first cost for no increase in moisture load.

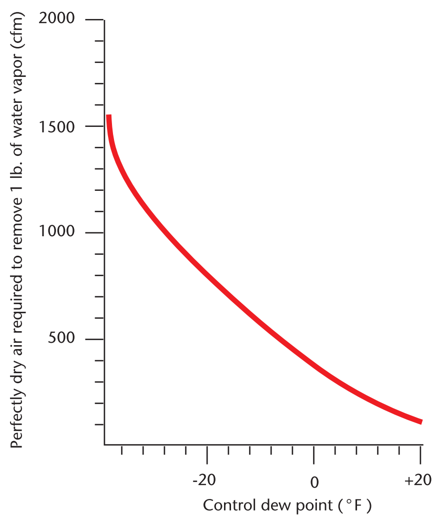


FIGURE 10.2

Low humidity requires large airflows

As the control level declines, it takes more dry air to remove the same load. System cost is roughly proportional to air quantity. This example shows the amount of totally dry air necessary to remove one pound of water vapor at different control levels.

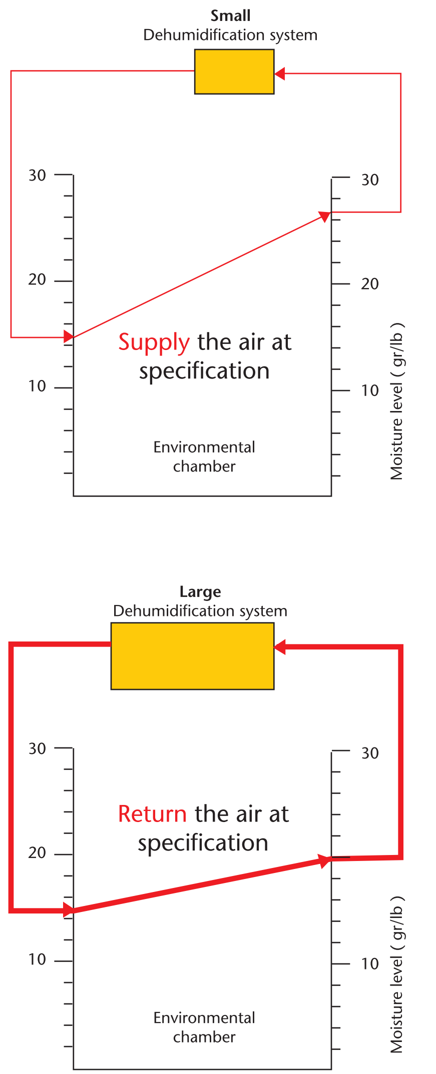


FIGURE 10.3

Specification confusion

The specification calls for a moisture control level of 15 gr/lb ±5 gr/lb, but it fails to define where that will be measured — in the room, at the air supply diffuser or at the return air grill. As the control level declines, such ambiguities can lead to different system designs from different suppliers.

Tolerances are another area for scrutiny by the cost-conscious designer, and like control levels, the cost of over-specifying rises dramatically as the control level goes down and the tolerances tighten. Also, confusion in a tolerance specification can be very costly.

For example, consider an environmental simulation chamber specification calling for a room control level of 15 gr/lb, with a tolerance of ± 5 gr/lb. The specification can have two meanings. One supplier may interpret the designer's intention as "the moisture condition at the discharge where air enters the space must not vary from the specified level by more than ± 5 gr/lb." A second supplier may interpret the specification to mean "the moisture condition throughout the room must be uniform. It cannot vary by more than 5 gr/lb from any one point in the room to any other."

The first supplier can satisfy the specification by delivering any amount of air — probably a small amount — at a condition of 15 gr/lb. The control level in the rest of the room may rise well above the 20 grains implied, but not made explicit by the specification. The second supplier will provide enough air to remove the specified load while maintaining no more than a 5 grain differential from system outlet to system inlet. This will be a much more costly system.

The designer can save a great deal through conversations with equipment suppliers concerning the effect of different specifications on system cost. This is particularly true as the moisture control level moves down to very low dew points.

Combining refrigeration and desiccant-based dehumidification equipment

Often, refrigeration-based dehumidification systems are cost-effective for moisture removal above a 50°F dew point. At lower levels, desiccant equipment is probably more efficient. However, most humidity control systems cross this line as wet ventilation air is brought into a building with a low moisture control level. The designer should consider using a combined system. In many cases, a system which uses both technologies will be less expensive to install and operate than one which uses either desiccants or refrigeration exclusively. This is particularly true for large building air conditioning systems. Chapter 8 described this optimization process in detail.

Minimizing operating cost

Reducing operating cost to a minimum is — like reducing installation cost — a matter of limiting load, and using the least expensive energy sources to remove it. Some aspects of this task are beyond the control of a system designer, since they involve the way a system is operated, or the way a production process must change from original assumptions. The designer, however, does control the most important methods of minimizing operational costs — modulating energy in response to load changes, and using the lowest-cost energy sources available.

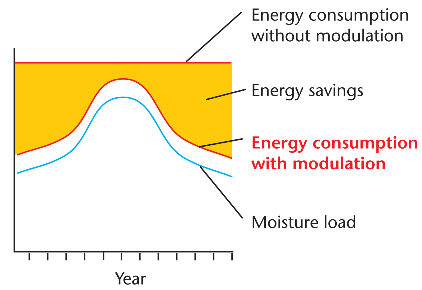


FIGURE 10.4

Modulating energy in response to load

Dehumidification systems are generally designed with enough capacity to meet maximum loads. But the maximum loads only occur for a short period. Unless the system is equipped with modulating controls, it will consume the same amount of energy regardless of load changes - rather like an automobile with an on-off switch but no accelerator pedal.

Energy modulation in response to load changes

The idea of purchasing a car without an accelerator pedal or transmission gearbox is almost inconceivable. But with imagination, one can picture the comic effect of lurching down the street with only an on-off switch and brake to control the engine's power. Yet until recent years, it was quite common for dehumidification systems to be designed without controls to modulate dehumidifier capacity, reactivation energy and cooling.

The importance of energy modulation depends on the size of the system, and on the nature of the load changes. Some systems — like those for long-term storage — may need nothing more than an on-off switch, because the loads are small and relatively constant, and there is no economic benefit to control precision in a narrow range. Other systems — such as those for semiconductor manufacturing — place a high premium on control precision, and the majority of the load comes from the ventilation air, which changes radically over short time spans.

Even when control precision is not an issue, deciding when to be concerned about energy modulation is fairly straightforward. Compare the annual cost of running the non-modulated system with its installed cost. When annual energy cost is a high percentage of the installed cost, modulate energy in response to load. Such control will probably reduce the annual energy cost by 25 to 50%.

In a rotary desiccant dehumidifier, reactivation energy can be modulated by reducing either the flow rate or the temperature of air entering the reactivation sector. The signal to reduce energy comes from a temperature sensor mounted in the air leaving reactivation. Since air picks up moisture in reactivation, its sensible temperature goes down — just as air temperature around a sprinkler is reduced in summertime as the air picks up moisture. The temperature drop is proportional to the moisture pick-up. If there is less moisture to pick up, the reactivation air

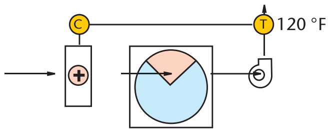


FIGURE 10.5

Reactivation energy modulation

In rotary dehumidifiers which use lithium chloride for the desiccant, reactivation energy is reduced when the temperature of the air leaving reactivation rises above 120°F. The higher temperature means there is less evaporation taking place in reactivation, which means less moisture is being absorbed on the process air side of the unit.

temperature will remain high, which indicates less energy is needed. So, as reactivation outlet temperature rises, the temperature sensor signals the control system to reduce reactivation energy.

There are two levels of energy modulation for desiccant systems. The first level could be called "reactivation load following" control. This is the control system described above — the reactivation heat is reduced as the moisture load from process declines. This control system is the lowest in cost and the highest in benefit.

The next level could be called "equipment reconfiguration" control. When loads are lower than design, moisture removal efficiency (measured in Btu's per lb of water removed) declines. This is similar to using a large dump truck to deliver a bag of groceries — the mass of the machinery must still be moved a given distance, but the payload is smaller. But unlike the truck, it is possible to reconfigure a dehumidification system for improved performance at low-load conditions by using a variety of sensors, microprocessors and variable speed motors for fans, refrigeration components and desiccant drives. The cost of such a system is quite high compared to the benefits, but extremely large systems may benefit enough in absolute dollar terms to justify the cost. For smaller installations, the strategy outlined in the next paragraph is often more cost-effective than installing such an "equipment reconfiguration" control system.

Multiple systems for disparate loads

Another way to reduce operating cost on a large installation is to match the equipment more closely to the loads in each part of the system. Borrowing another analogy from transportation, imagine the benefits of a railroad compared to a fleet of trucks. The railroad has advantages when loads are heavy, continuous and always travel the same path. The trucks have advantages when loads and their locations are very diverse and intermittent. In the second situation, the railroad would be prohibitively large and expensive to meet diverse needs. Likewise, large, centralized dehumidification systems are cost-effective when loads are constant and continuous, but where there are diverse circumstances, the designer may want to consider separate smaller systems.

For example, imagine a cool, humidity-controlled brewery cellar with a constant moisture load from ventilation air brought into the space to remove carbon dioxide gas from fermentation. Occasionally, the brewing equipment is washed down and cleaned, temporarily adding a massive internal moisture load for the system to remove. If the designer sizes the main dehumidification system to remove both peak weather moisture and peak internal moisture, the system will be enormous and difficult to control at low-load conditions. It would be very cost-effective to design the basic system for the maximum weather load, and add a separate, much smaller dehumidifier to operate only when the basic system cannot handle a simultaneous peak of internal and ventilation loads.

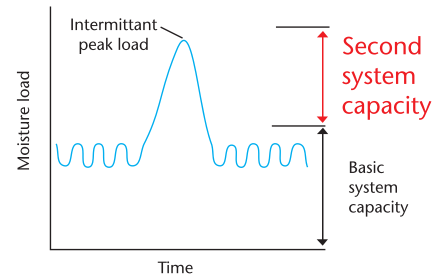


FIGURE 10.6

Intermittent loads

When large loads occur intermittently, it is often cost-effective to have two smaller systems than a single larger one. The second system operates only when necessary. The basic system runs more fully loaded, which can save energy over a single, partly loaded large system.

Utilizing low-cost reactivation energy

Energy consumed by a desiccant dehumidification system is used in three places: electrical power for fans and pumps, electrical or thermal energy for cooling, and thermal energy for desiccant reactivation. Reactivation is the largest energy consumer (in terms of Btu's), and many creative methods have been developed to minimize its cost.

Beginning with the least-cost source, a system designer can re-use energy that has already been paid for by recovering heat from cogeneration systems, refrigeration condensers or the reactivation process itself. In a rotating desiccant unit, the air leaving reactivation is quite hot and moist. The designer can place an air-to-air heat exchanger between the air entering and leaving reactivation to recover this exhaust energy for re-use. For a very modest investment in a heat exchanger, the engineer can recover 60 to 80% of the energy the system has already used once — surely the lowest-cost energy source available.

In a liquid desiccant unit, a liquid-to-liquid type heat exchanger can be placed between the warm desiccant leaving the regenerator and the cooler desiccant entering the regenerator. This has a doubly beneficial effect. Less energy is needed to regenerate the desiccant, because it is warmer than when it left the conditioner. And less energy is needed to cool the process air, because the desiccant returning to the conditioner is cooler than when it left the regenerator.

Refrigeration condensers reject heat from the air-cooling process. This energy can be used to pre-heat the air entering the reactivation sector of a dehumidifier. Refrigeration systems often include a "de-superheating" coil to cool the refrigerant as it moves between the compressor and condenser. This coil can be arranged so the rejected heat passes to the air entering the reactivation sector of the desiccant unit.

Cogeneration of electrical power also rejects heat that can be used to reactivate — or partially reactivate — a desiccant dehumidifier. Such generators are often powered by engines that reject heat to a water-based cooling system. The hot water from this system can be used to

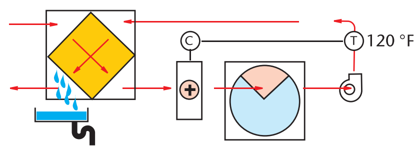


FIGURE 10.7

Reactivation heat recovery

The lowest-cost energy source for desiccant reactivation is the waste heat from reactivation itself. In this design, an air-to-air heat exchanger is installed between reactivation entering and leaving airstreams. The waste heat pre-heats the incoming air, and excess moisture in the leaving air condenses out into the drain pan on the other side of the heat exchanger.

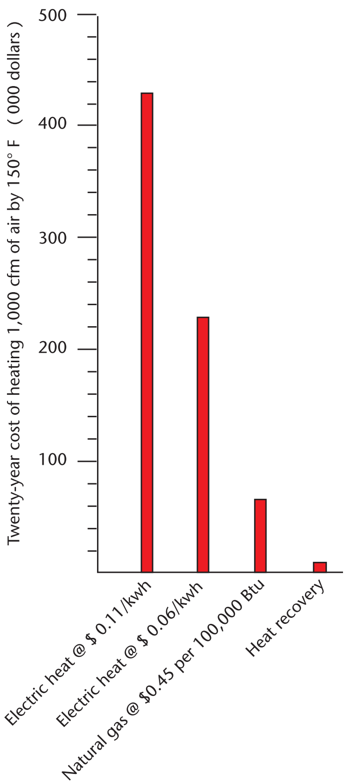


FIGURE 10.8

Reactivation energy costs

Heat recovered from either reactivation, refrigeration or cogeneration is by far the least expensive source of energy over the typical twenty year life of the equipment. This graph includes the cost of the heat recovery equipment, and the fan energy to push air through the heat exchanger.

pre-heat air before it enters reactivation heaters in a solid desiccant dehumidifier. In a liquid desiccant system, a liquid-to-liquid heat exchanger can use the engine jacket water to heat the desiccant before it is pumped to the regeneration heater.

In short, any heat rejected from another process can be used to minimize the cost of regenerating a desiccant dehumidifier. Also, the designer can seek out other low-cost sources of heat. For example, utility costs can vary throughout the year, depending on the supply and demand at the time. In some European countries, electrical power is very inexpensive during the summer — just when desiccant regeneration demand is high. In parts of the U.S. and many developing countries, there is an abundant supply of natural gas in summertime, so the cost to use that energy for desiccant reactivation can be very low.

Many industrial plants as well as commercial and institutional buildings have boilers which must be operated year-round to support process heating needs. Since there is virtually no comfort-heating requirement in summertime, this “nearly-waste” energy can be used to regenerate a desiccant system at very low cost. Also, clean, warm exhaust air can be used for desiccant regeneration directly. For example, many large mechanical rooms are cooled by ventilation fans even during winter months. This air can be used for desiccant regeneration instead of colder weather air, saving considerable energy cost.

Staged reactivation

To fully reactivate a desiccant, the material must be raised to a relatively high temperature. But this high temperature is only necessary to remove the last portion of the water, which is tightly bound by the desiccant. Most of the water can be removed from the material by lower temperature heat, which is generally less costly than higher temperature energy. Equipment manufacturers can sometimes use a two-stage reactivation process to remove 70 to 80% of the water with low-grade heat, and can accomplish only the final drying with high-cost heat.

Desiccant energy storage

In a liquid desiccant system, it is possible to purchase extra desiccant and holding tanks beyond regular requirements, regenerating the liquid when energy is inexpensive and storing it until there is a need for more moisture removal capacity. This stores the latent capacity of a building HVAC system in the form of dry desiccant just like sensible cooling capacity can be stored by making ice at off-peak rates with a refrigeration system.

Utilizing low-cost cooling

A desiccant system often needs cooling where temperature as well as humidity must be controlled. The designer can minimize this energy cost by using indirect evaporative cooling, by using thermal energy to drive an absorption cooling system, or by increasing cooling system efficiency through raising the refrigerant evaporator temperature.

Indirect evaporative cooling cools air by adding water to it by passing it through a spray, mist or through contact with a wet surface. The cool air in turn cools either water or another airstream through a heat exchanger. Large buildings often use cooling towers — indirect evaporative coolers — to obtain low-cost cooling for industrial processes or refrigeration systems. Dehumidification systems can also make use of existing cooling towers to cool air or liquid desiccant solutions.

Absorption cooling systems use low-cost thermal energy to cool air or water. Such systems form a closed loop like a vapor-compression refrigeration system. Water is the refrigerant gas instead of the more common halocarbon gases. Inside the system, water evaporates from one section, because it is attracted to a desiccant in another. This evaporation removes heat from air on the other side of a cooling coil. This cooling effect can be used to remove heat from the desiccant dehumidification process, just as vapor-compression refrigeration removes heat from air and fluids.

Desiccant dehumidification systems can also use cooling from more-efficient vapor-compression systems. Such systems often cool air to low dew points for dehumidification. When a desiccant unit removes the moisture, the cooling system may be able to operate at higher evaporator temperatures, which improves efficiency and lowers costs.

Operational considerations

If the system designer is also in charge of its operation, there is much he or she can do to minimize operational costs. Even when that is not the case, the designer has an obligation to the organization to make clear the economic consequences of operating the system in different ways.

If a freezer door were open, the homeowner would not expect the ice cream to stay frozen. Likewise, if the door to a humidity-controlled space is opened frequently, the humidity level can be expected to rise, and the dehumidification system must use more energy to remove the load. Devices as simple as a door sign reminding personnel to keep the door shut tight have effectively reduced operational costs.

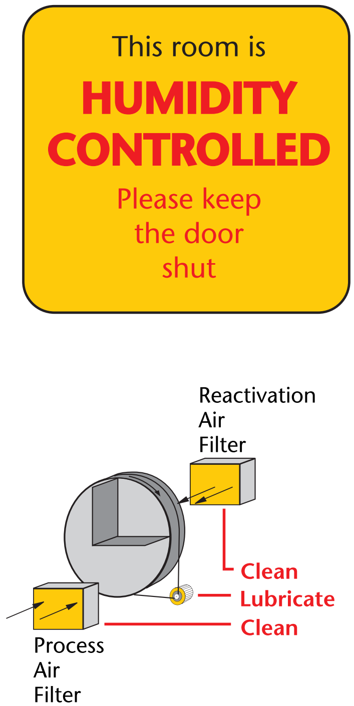


FIGURE 10.9

Operations management

Operations personnel can minimize system cost by two simple methods — keeping the door closed, which minimizes moisture load, and performing regular filter changes and drive lubrication. Desiccant dehumidifiers are surprisingly durable and trouble-free when minimal maintenance is performed on a regular basis.

Adding a fume hood to a humidity-controlled room means more air will have to be brought into the room to replace the air exhausted through the hood. This increases the moisture load, which increases the operating expense. If exhaust air can be avoided, it will minimize operational costs.

Desiccant dehumidifiers are such basically simple and reliable devices that normal preventive maintenance is often forgotten. Lack of maintenance creates very high costs when key components eventually fail through lack of simple attention — just like a car's engine will fail when operated 24 hours a day, 7 days a week without oil changes. Regardless of the type of dehumidifier, two tasks must be accomplished regularly — changing dirty filters and lubricating drive motors and fans or pumps.

Without this minimal attention, desiccants clog, heaters burn out and motors seize up, costing thousands or tens of thousands of dollars where less than a hundred dollars worth of maintenance time over a year could have avoided the problems.

Summary

Dehumidification projects are not without expense. Like any effort, there is an investment of time and money required before the benefits will be achieved. The designer's work is made easier if he or she can identify and quantify the benefits, and minimize the costs of a project. The concepts and suggestions outlined in this chapter are summarized in the following checklists.

Dehumidification Benefit Worksheet

Operational cost reduction

Interruption costs
_______ x _______ x 20 years = _______
Cost per hour × Hours per year × 20 years = Interruption savings

Reducing re-work
_______ x _______ x 20 years = _______
Cost per piece × Pieces per year × 20 years = Re-work savings

Reducing energy costs
_______ x _______ x 20 years = _______
Average hourly savings × Hours per year × 20 years = Energy savings

Reducing need for skilled labor
_______ x _______ x 20 years = _______
Average hourly cost × Hours per year × 20 years = Labor savings

_______ x _______ x 20 years = _______
Recruitment costs × Employees/yr × 20 years = Employee aquistion saving

Reducing maintenance costs
_______ x _______ x 20 years = _______
Average hourly cost × Hours per year × 20 years = Labor savings

_______ x _______ x 20 years = _______
Replacement part cost × Parts per year × 20 years = Parts savings

Reducing worker's compensation costs
_______ x _______ x 20 years = _______
Average claim cost × Claims per year × 20 years = Compensation savings

Reducing cost of capital investments

Reducing need for plant expansion
_______ x _______ = _______
Cost of new plant & equipment × Number of plants = Plant & equipment savings

Reducing equipment replacement costs
_______ x _______ x 20 years = _______
Equipment cost × Pieces of equipment/yr × 20 years = Replacement savings

Reducing HVAC system installed costs
_______ + _______ - _______ = _______
Equipment cost + Installation cost - Project costs = HVAC system savings

Improved product value

Value of improved market image
_______ x _______ x 20 years = _______
Value increase per unit × Units per year × 20 years = Value improvement

Reduced scrap rate
_______ x _______ x 20 years = _______
Profit per unit × Units per year × 20 years = Scrap reduction value

Material property improvements
_______ x _______ x 20 years = _______
Value increase per unit × Units per year × 20 years = Value improvement

Improved operational response

Avoid expensive peak capacity purchases
_______ x _______ x 20 years = _______
Cost per unit purchased × Units per year × 20 years = Reduced cost of operation

Avoid unscheduled maintenance
_______ x _______ x 20 years = _______
Cost per unscheduled event × Events per year × 20 years = Reduced cost of operation

Less equipment required
_______ x _______ = _______
Cost per item of extra equipment × Number of items = Reduced equipment cost

Value increased by storage
_______ x _______ x 20 years = _______
Change in product value × Units per year × 20 years = Peak demand value increase

Total dehumidification project economic benefits = _______________

Guide To Mimimizing Project Costs

Minimizing first cost

- ◯ Minimize moisture loads
  - ◯ Tape all joints and seams to avoid air leaks
  - ◯ Avoid excessive door activity
  - ◯ Use vapor retarder film and paints on the building envelope
  - ◯ Supply slight excess of ventilation air to reduce infiltration
  - ◯ Design doors with air-lock vestibules
  - ◯ Use plastic strip curtains, over-pressure air and ducted openings to reduce infiltration
  - ◯ Minimize the open face area of exhaust hoods
- ◯ Optimize control levels
  - ◯ Specify the highest moisture level that will achieve the desired economic benefit
  - ◯ Specify the highest temperature level that will achieve the desired economic benefit
- ◯ Clearly specify tolerances
  - ◯ Specify minimum and maximum humidity in absolute units
  - ◯ Where relative humidity must be specified, ensure temperature range is also specified.
- ◯ Combine desiccant and refrigeration-based dehumidification
  - ◯ Use refrigeration for dehumidifying at high temperatures and high dew points
  - ◯ Use desiccants for control at low temperatures and low dew points

Minimizing Operating cost

- ◯ Modulate energy in response to load changes
  - ◯ Desiccant reactivation heat
  - ◯ Cooling capacity
- ◯ Use multiple systems for intermittant loads
  - ◯ Dehumidfiers
  - ◯ Coolers
- ◯ Use lowest-cost reactivation energy
  - ◯ Energy recovery from reactivation
  - ◯ Heat recovery from refrigeration systems
  - ◯ Off-peak natural gas
  - ◯ Low-cost excess boiler capacity or steam condensate
  - ◯ Use clean, warm exhaust air from mechanical rooms
- ◯ Use lowest-cost cooling source
  - ◯ Indirect evaporative coolers
  - ◯ Cooling-tower water
  - ◯ Refrigeration with high evaporator temperature
- ◯ Optimize system operation
  - ◯ Keep loads to a minimum by minimizing door activity
  - ◯ Reduce maintenance cost through regular filter replacement and drive lubrication

APPENDIX

Table A-1
WEATHER DATA FOR DESIGN

Location	Elevation ft.	Pressure psia	DEHUMIDIFICATION 0.4% (35 hours above...)			DEHUMIDIFICATION 1% (88 hours above...)			Summer Wind			HEATING 99.6% 35 hours below...	COOLING 0.4% 35 hours above...	
			gr/lb	°F (DB)	in. Hg.	gr/lb	°F (DB)	in. Hg.	mph	fpm	PWD	°F (DB)	°F (DB)	°F (WB)
ALABAMA														
Birmingham	615	14.37	139	83	0.9023	133	82	0.8669	8	704	0	21	96	75
Huntsville	624	14.37	139	83	0.9023	133	82	0.8698	7	616	0	19	95	75
Mobile	215	14.58	146	83	0.9673	142	83	0.9390	7	616	10	28	94	77
ALASKA														
Anchorage	138	14.62	70	63	0.4721	66	62	0.4439	7	616	80	-11	73	59
Juneau	16	14.69	70	61	0.4721	67	60	0.4520	8	704	90	5	74	60
ARIZONA														
Phoenix	1107	14.12	120	82	0.7731	113	84	0.7246	9	792	100	39	110	70
Tucson	2549	13.39	118	76	0.7222	114	77	0.6930	11	968	140	32	106	66
ARKANSAS														
Little Rock	311	14.53	152	85	0.9964	144	85	0.9452	8	704	300	18	100	77
CALIFORNIA														
Los Angeles	97	14.64	101	74	0.6742	96	73	0.6468	9	792	90	45	84	63
Merced	152	14.62	89	82	0.5948	82	80	0.5485	9	792	100	29	102	70
Sacramento	23	14.68	90	84	0.6011	85	83	0.5723	7	616	320	30	101	70
San Diego	15	14.69	105	75	0.7026	101	74	0.6765	9	792	340	45	84	65
San Francisco	8	14.69	81	68	0.5485	78	67	0.5238	13	1144	150	40	83	63
COLORADO														
Colorado Springs	6181	11.70	96	66	0.5127	91	66	0.4894	12	1056	150	1	91	59
Denver	5414	12.04	97	68	0.5351	92	68	0.5054	9	792	230	-1	95	60
CONNECTICUT														
Hartford	19	14.69	126	80	0.8383	121	79	0.8079	9	792	0	8	91	73
DELAWARE														
Dover	28	14.68	139	82	0.9266	132	81	0.8815	11	968	320	15	93	76
DISTRICT OF COLUMBIA														
Washington/Reagan	10	14.69	134	83	0.8963	129	82	0.8582	10	880	330	18	95	76
FLORIDA														
Jacksonville	26	14.68	143	83	0.9483	139	83	0.9266	8	704	320	30	95	77
Miami	29	14.69	148	84	0.9801	144	83	0.9546	10	880	340	49	92	78
Tampa	19	14.69	148	85	0.9801	143	84	0.9515	8	704	10	40	93	77
GEORGIA														
Atlanta	1027	14.16	133	81	0.8554	129	80	0.8299	9	792	320	22	94	74
Macon	343	14.51	139	83	0.9114	134	82	0.8845	8	704	300	24	97	75
Savannah	46	14.67	146	84	0.9673	141	83	0.9359	8	704	300	28	96	77
HAWAII														
Honolulu	46	14.67	146	84	0.9673	141	83	0.9359	8	704	300	28	96	77
IDAHO														
Boise	2814	13.26	78	72	0.4772	72	72	0.4375	9	792	120	9	99	64

Weather Data Derived From:
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Vapor pressure by Linric Company (www.LINRIC.com) using PsyFunc© add-in functions for MicroSoft EXCEL©

Table A-1
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Location	Elevation ft.	Pressure psia	DEHUMIDIFICATION 0.4% (35 hours above...) gr/lb	°F (DB)	in. Hg.	1% (88 hours above...) gr/lb	°F (DB)	in. Hg.	Summer Wind mph	fpm	PWD	HEATING 99.6% 35 hours below... °F (DB)	COOLING 0.4% 35 hours above... °F (DB)	°F (WB)
ILLINOIS														
Belleville/Scott AFB	459	14.45	149	86	0.9705	141	85	0.9266	8	704	330	7	95	78
Chicago/Miegs	612	14.37	134	84	0.8727	125	82	0.8161	13	1144	270	1	92	75
Springfield	594	14.38	141	86	0.9205	134	84	0.8756	9	792	280	1	93	77
INDIANA														
Fort Wayne	791	14.28	134	83	0.8669	127	81	0.8216	11	968	260	-1	91	75
Terre Haute	575	14.39	141	85	0.9205	134	83	0.8727	9	792	310	2	93	76
IOWA														
Des Moines	957	14.19	140	85	0.9023	133	84	0.8554	12	1056	320	-4	93	76
Souix City	1095	14.12	142	85	0.9114	133	84	0.8525	14	1232	320	-7	93	75
KANSAS														
Whichita	1321	14.01	134	84	0.8525	128	82	0.8161	15	1320	0	8	101	73
KENTUCKY														
Fort Knox	755	14.30	142	84	0.9205	135	83	0.8756	7	616	280	9	93	76
Louisville	488	14.44	137	85	0.8993	131	83	0.8582	9	792	300	11	94	75
LOUISIANA														
New Orleans	4	14.69	148	85	0.9834	145	84	0.9610	7	616	10	33	94	78
Shreveport	254	14.56	139	83	0.9144	135	83	0.8933	7	616	10	26	99	76
MAINE														
Bangor	148	14.62	110	78	0.7321	103	75	0.6906	10	880	310	-7	88	71
MASSACHUSSETTS														
Boston	12	14.69	121	80	0.8079	115	79	0.7731	14	1232	290	9	91	73
Worcester	1000	14.17	120	77	0.7757	115	76	0.7396	11	968	280	2	86	71
MICHIGAN														
Detroit	626	14.37	126	82	0.8216	119	80	0.7810	11	968	240	5	91	73
MINNESOTA														
Minneapolis	872	14.24	129	84	0.8383	121	81	0.7836	13	1144	300	-11	91	73
MISSIPPI														
Jackson	330	14.52	142	84	0.9359	138	83	0.9083	7	616	340	24	96	76
MISSOURI														
Kansas City	1005	14.17	146	87	0.9328	138	85	0.8845	12	1056	310	2	96	77
MONTANA														
Great Falls	3664	12.85	82	67	0.4824	76	67	0.4455	10	880	220	-16	93	61
NEBRASKA														
Lincoln	1189	14.08	144	86	0.9205	135	84	0.8582	4	352	0	-3	95	74
Omaha	982	14.18	143	86	0.9205	135	84	0.8669	14	1232	340	-3	95	76
NEVADA														
Las Vegas	2203	13.56	105	84	0.6468	97	87	0.5990	10	880	320	32	108	68

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NEW HAMPSHIRE														
Concord	346	14.51	118	78	0.7810	112	77	0.7396	10	880	320	-3	90	71
Portsmouth	100	14.64	120	80	0.7997	114	78	0.7601	10	880	290	3	90	73
NEW JERSEY														
Trenton	184	14.60	128	81	0.8468	123	80	0.8161	9	792	320	12	93	74
NEW MEXICO														
Albuquerque	5310	12.09	99	68	0.5485	95	69	0.5238	10	880	0	18	96	60
NEW YORK														
Binghamton	1595	13.87	117	77	0.7396	111	75	0.7026	10	880	320	0	85	70
Buffalo	716	14.32	123	79	0.7997	115	78	0.7524	12	1056	260	4	86	71
New York/LGA	11	14.69	127	81	0.8468	122	80	0.8133	12	1056	300	14	93	74
Syracuse	413	14.48	120	81	0.7916	113	79	0.7472	10	880	90	-1	89	73
NORTH CAROLINA														
Asheville	2151	13.59	130	77	0.7997	124	76	0.7653	2	176	30	13	86	70
Charlotte	728	14.31	131	81	0.8525	127	80	0.8244	7	616	310	22	94	75
Greensboro	890	14.23	129	81	0.8327	125	80	0.8079	8	704	300	19	93	74
Fayetteville	186	14.60	136	82	0.8963	132	81	0.8756	9	792	20	23	96	76
NORTH DAKOTA														
Bismark	1651	13.84	121	82	0.7627	111	79	0.7002	12	1056	320	-18	93	70
Fargo	900	14.22	124	82	0.7997	114	80	0.7371	14	1232	310	-19	90	72
OHIO														
Akron	1044	14.15	124	80	0.7970	118	78	0.7627	10	880	220	4	89	72
Dayton	962	14.19	129	81	0.8271	125	80	0.8051	9	792	280	5	90	74
Toledo	669	14.34	130	82	0.8440	123	80	0.7997	11	968	240	2	91	74
OKLAHOMA														
Oklahoma City	1285	14.03	134	84	0.8554	129	83	0.8244	12	1056	0	15	101	74
Tulsa	650	14.35	137	86	0.8904	132	85	0.8582	10	880	350	14	100	75
OREGON														
Portland	29	14.68	88	73	0.5948	85	73	0.5703	7	616	80	25	91	67
Salem	205	14.59	83	73	0.5544	78	72	0.5238	9	792	0	24	92	67
North Bend	17	14.69	81	64	0.5427	75	63	0.5054	13	1144	150	31	72	61
Pendleton	1486	13.92	77	71	0.4947	71	71	0.4553	10	880	150	8	97	65
PENNSYLVANIA														
Harrisburg	312	14.53	133	83	0.8756	126	81	0.8299	9	792	310	12	92	75
Philadelphia	10	14.69	133	83	0.8845	127	81	0.8468	12	1056	290	15	94	75
Pittsburgh	1203	14.07	124	80	0.7943	119	78	0.7601	9	792	250	6	89	72
SOUTH CAROLINA														
Charleston	40	14.67	149	84	0.9866	144	84	0.9546	10	880	30	28	94	78
Columbia	225	14.58	136	82	0.8993	132	81	0.8727	9	792	260	23	97	75

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SOUTH DAKOTA														
Rapid City	3160	13.09	111	78	0.6603	103	76	0.6160	13	1144	350	-9	97	66
Sioux City	1428	13.95	137	84	0.8669	128	82	0.8106	15	1320	310	-11	92	74
TENNESSEE														
Knoxville	962	14.19	132	81	0.8468	127	80	0.8188	7	616	20	17	93	74
Memphis	254	14.56	141	86	0.9266	135	85	0.8933	8	704	10	20	97	77
Nashville	600	14.38	135	83	0.8786	129	82	0.8440	7	616	330	15	95	75
TEXAS														
Amarillo	3604	12.88	114	75	0.6719	110	74	0.6445	15	1320	0	10	99	65
Dallas/Fort Worth	560	14.40	136	84	0.8874	131	83	0.8554	10	880	340	23	101	74
Houston	32	14.68	162	84	1.0707	153	83	1.0129	8	704	340	34	97	78
Lubbock	3254	13.05	119	77	0.7050	114	76	0.6789	14	1232	20	16	99	66
San Antonio	789	14.28	140	80	0.9053	136	80	0.8815	10	880	10	30	99	74
UTAH														
Cedar City	5586	11.96	89	67	0.4876	82	68	0.4520	11	968	60	2	94	59
Salt Lake City	4225	12.59	92	72	0.5275	84	73	0.4841	11	968	140	10	98	63
VERMONT														
Burlington	330	14.52	117	79	0.7731	110	77	0.7296	10	880	80	-7	88	71
VIRGINIA														
Richmond	164	14.61	135	83	0.8963	130	82	0.8611	9	792	0	19	95	76
Roanoke	1175	14.08	124	79	0.7970	120	79	0.7679	9	792	300	17	92	73
WASHINGTON														
Seattle	370	14.50	82	70	0.5427	77	69	0.5164	9	792	20	25	85	65
Spokane	2353	13.49	78	69	0.4859	73	68	0.4504	8	704	30	5	93	63
WISCONSIN														
Madison	866	14.24	130	83	0.8440	123	81	0.7943	10	880	290	-6	89	74
Milwaukee	670	14.34	128	82	0.8327	121	80	0.7863	14	1232	300	-1	90	74
WYOMING														
Casper	5313	12.09	86	67	0.4789	80	66	0.4423	12	1056	40	-9	94	60
Cheyenne	6130	11.72	92	66	0.4947	87	65	0.4670	12	1056	0	-4	90	58
PUERTO RICO														
Ceiba/RR	33	14.68	152	84	1.0063	150	84	0.9964	12	1056	330	68	90	78

Weather Data Derived From:
ASHRAE Handbook—Fundamentals 2017, used by permission of the American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc. www.ASHRAE.org
Vapor pressure by Linric Company (www.LINRIC.com) using PsyFunc© add-in functions for MicroSoft EXCEL©

Table A-2
FILTER SELECTION GUIDE

	FILTER DESCRIPTION	MAXIMUM FACE VELOCITY fpm	MAXIMUM PRESSURE DROP in H_2O CLEAN	DIRTY	EFFICIENCY[1] %	ARRESTANCE[2] %	COMMENTS Application and Limitations
A	Metal Mesh, 2″ Deep, Oiled (Cleanable)	520	.35	.5	15 – 20	50%	Reasonably effective on large particles and fibers such as lint. Generally, ineffective on pollen, smoke and settling dust.
B	Microfine Glass Fiber Media with Wire Support Grid, 2″ Deep (Disposable)	500	.30	.90	30 – 35	90%	Central domestic heating and cooling systems. Prefilter for higher efficiency filters. Limited effect on smoke and soiling particles.
C		500	.35	1.5	50 – 55	95%	Commercial fresh and recirculated systems. Effective on pollen and fine dust particles. Limited effect on fume and smoke. Ineffective on tobacco smoke.
D	High Density Microfine Glass Fibers with Wire Support Grid & Contour Stabilizers, 12″ Deep (Disposable)	500	.5	1.5	80 – 85	99%	Commercial fresh and recirculated systems. Effective on all pollen and dust, most soiling particles, fume coal and oil smoke.
E		500	.65	15	90 – 95	100%	Pharmaceutical and clean hospital areas, effective on soiling particles, coal and oil smoke, and bacteria. Effective against tobacco smoke.
F	Continuous Sheet of Glass Micro Fiber Paper with Corrugated Separators. Wood Board Frame with Gaskets. 12″ Deep (Disposable)	275	1.0	3 – 4.0	99.97[3]	100%	Excellent protection against all smokes and fumes. Bacteria and all forms of dusts. Surgery rooms, intensive care wards, clean rooms and pharmaceutical packaging.

NOTES:

1. Efficiency based on ASHRAE Dust Spot Efficiency Test Method using atmospheric dust.
2. Arrestance based on ASHRAE Weight Arrestance Test Method using synthetic dust.
3. Based on DOP Test (0.3 micron smoke).

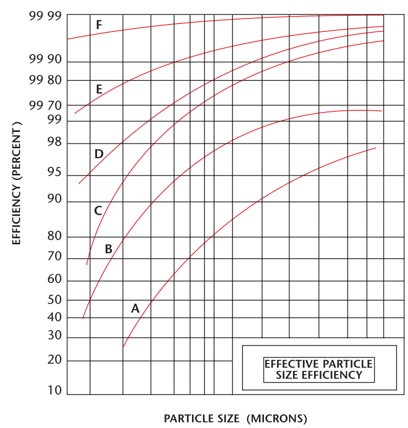


Table A-3
USEFUL STEAM DATA

PRESSURE psig	TEMPERATURE °F	LATENT HEAT Btu/lb
2	219	965
5	227	960
10	239	952
15	250	945
20	259	939
25	267	933
30	274	928
40	287	919
50	298	911
60	307	904
70	316	897
80	324	891
90	331	885
100	338	880
125	353	868
150	366	857
175	377	847
200	388	838

NOTES:

$$\text{STEAM CONSUMPTION} = \frac{1.08 \times \text{scfm} \times \text{Air Temp Rise}}{\text{Latent Heat of Steam}}$$

EXAMPLE:	Air Flow	500scfm
	Temp Rise	185°F
	Steam Pressure	30 psig
	Latent Heat	928 Btu/lb

$$\frac{1.08 \times 500 \text{ scfm} \times 185°\text{F Temp Rise}}{928 \text{ Btu/lb Latent Heat}} = 107.7 \text{ lb/hr steam consumption}$$

Table A-4
USEFUL CONVERSION FACTORS

TO CONVERT	INTO	MULTIPLY BY
atmospheres	kilpascal (kP_a)	101.1
Btu	joules	1,055
Btu	kilowatt-hrs	2.928×10^{-4}
Btu/hr	horsepower	3.929×10^{-4}
Btu/hr	watts	0.293
Btu/hr-sq ft	watts/sq meter	3.15
Btu/lb	kilojoules/kilogram	2.33
Centigrade	Fahrenheit	(C° x 9/5) + 32
cost, $/lb	$/kilogram	2.205
cost, $/ton (refrig)	$/kilowatt	0.284
cubic feet	cu meters	0.02832
cubic feet	liters	28.32
cubic feet/min, cfm	liters/sec	0.4720
cubic inches	milliliters	16.4
cubic meters	cu feet	35.31
cubic meters	liters	1,000
Fahrenheit	Centigrade	(F°–32) x 5/9
feet	meters	0.3048
feet/min, fpm	meters/sec	5.08×10^{-3}
feet/min, fpm	miles/hr	0.01136
gallons	cu feet	0.1337
gallons	liters	3.785
gallons/hr	milliliters/sec	1.05
gallons/min	liters/sec	0.06308
grains (1/7000lb)	grams	0.06480
grains/lb	grams/kilogram	0.143
grams	grains	15.43
grams	pounds	2.205×10^{-3}
horsepower	Btu/hr	2,547.
horsepower	kilowatts	0.7457
inches	millimeters	25.40
inches of mercury	inches of water	13.596
inches of mercury	kilpascal (kP_a)	3.38
inches of water (at 4°C)	inches of mercury	0.07355
inches of water (at 4°C)	pounds/sq in.	0.03613
inches of water (at 4°C)	pascal (P_a)	249.
joules	Btu	9.48×10^{-4}
joules	watt-hrs	2.778×10^{-4}

Table A-4
USEFUL CONVERSION FACTORS (continued)

TO CONVERT	INTO	MULTIPLY BY
kilograms	pounds	2.205
kilograms/cu meter	pounds/cu ft	0.06243
kilometers	feet	3,281.
kilometers	miles	0.6214
kilometers/hr	miles/hr	0.6214
kilowatts	Btu/hr	3,414.
liters	cu feet	0.03531
liters	gallons (U.S. liq.)	0.2642
liters/sec	gallons/min	15.852
meters	feet	3.281
meters/sec	feet/min	196.8
meters/sec	kilometers/hr	3.6
meters/sec	miles/hr	2.237
miles (statute)	meters	1,609.
miles/hr	feet/min	88.
miles/hr	kilometers/hr	1.609
millimeters	inches	0.03937
newton	pounds (force)	0.225
newton/sq meter	pascal (P_a)	1.0
pascal (P_a)	inches of mercury	2.959×10^{-4}
pascal (P_a)	inches of water	4.016×10^{-3}
pascal (P_a)	pounds/sq in.	1.451×10^{-4}
pounds (force)	newtons	4.448
pounds (mass)	grains	7,000.
pounds (mass)	kilograms	0.4536
pounds of water	cu feet	0.01603
pounds of water	gallons	0.1199
pounds/cu ft	kgs/cu meter	16.02
pounds/sq in.	inches of mercury	2.036
pounds/sq in.	kilpascal (kP_a)	6.89
pounds/sq in.	inches of water	27.68
tons, refrigeration	Btu/hr	12,000.
tons, regrigeration	kilowatts	3.52
watts	Btu/hr	3.414
watts	horsepower (elec.)	1.341×10^{-3}
watt-hours	Btu	3.414
watt-hours	joules	3.60×10^{3}

Table A-5

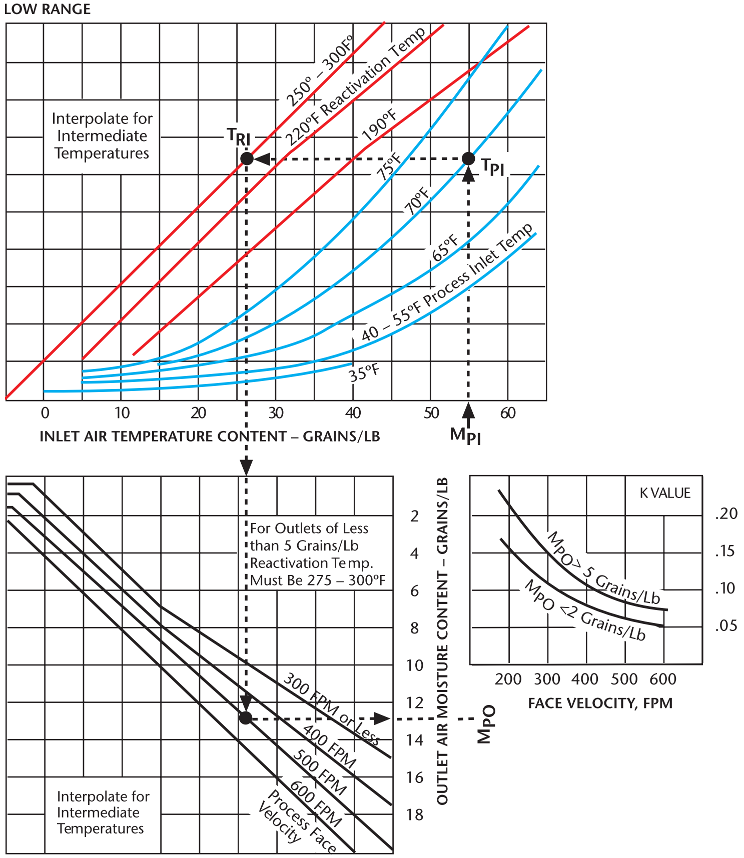
DEW POINTs vs. MOISTURE CONTENT at ALTITUDES or HIGH PRESSURE

		MOISTURE CONTENT (grains/lb) @ ABSOLUTE PRESSURES				
DEW POINT TEMP (°F)	**VAPOR PRESSURE (in Hg)**	**SEA LEVEL 29.92 in.Hg.**	**3000 ft. 26.82 in.Hg.**	**5000 ft. 24.90 in.Hg.**	**7000 ft. 23.09 in.Hg.**	**100 psig 233.4 in.Hg.**
– 60	.00101	.148	.164	.177	.190	.0188
–55	.00143	.209	.232	.250	.270	.0267
–50	.00199	.291	.323	.348	.375	.0371
–45	.00276	.404	.448	.483	.520	.0515
–40	.00379	.555	.615	.662	.715	.0707
–35	.00518	.758	.841	.906	.977	.0966
–30	.00700	1.02	1.14	1.22	.132	.131
–25	.00944	1.38	1.53	1.65	1.78	.176
–20	.0126	1.84	2.05	2.20	2.38	.235
–15	.0167	2.44	2.71	2.92	3.15	.312
–10	.0220	3.22	3.57	3.85	4.15	.410
–5	.0289	4.23	4.70	5.06	5.46	.539
0	.0376	5.51	6.11	6.59	7.10	.701
5	.0488	7.14	7.94	8.55	9.22	.910
10	.0629	9.20	10.2	11.0	11.90	1.17
15	.0806	11.8	13.1	14.1	15.3	1.50
20	.103	15.1	16.8	18.1	19.5	1.92
25	.130	19.1	21.2	22.9	24.7	2.43
30	.165	24.2	26.9	29.0	31.3	3.08
35	.203	29.9	33.2	35.8	38.6	3.79
40	.247	36.5	40.5	43.6	47.1	4.61
45	.300	44.3	49.3	53.1	57.3	5.60
50	.362	53.6	59.6	64.2	69.3	6.76
55	.436	64.6	72.0	77.6	83.8	8.15
60	.522	77.6	86.4	93.2	101.	9.76
65	.622	92.8	103.	112.	121.	11.6
70	.739	111.	123.	133.	144.	13.8
75	.875	132.	147.	159.	171.	16.4
80	1.03	156.	174.	188.	203.	19.3
85	1.21	185.	206.	222.	241.	22.7
90	1.42	218.	243.	263.	285.	26.6
95	1.66	257.	287.	310.	337.	31.2
100	1.93	302.	325.	366.	397.	36.3

Table A-6
ROTARY HONEYCOMBE DEHUMIDIFIER PERFORMANCE

MODEL	NORMAL DRYING CAPACITY lb/hr	MAXIMUM DRY PROCESS AIR scfm	NOMINAL PROCESS FACE AREA sq ft	MINIMUM REACTIVATION AIR REQUIRED scfm	MAXIMUM REACTIVATION AIR CAPACITY scfm
HC-1125	3 – 40	1,125	1.88	100	400
HC-2250	10 – 60	2,250	3.75	250	600
HC-4500	20 – 120	4,500	7.50	500	1,300
HC-9000	40 – 300	9,000	15.0	1,000	2,800
HCE-15,000	100 – 750	15,000	25.0	2,000	7,500
HCE-20,000	130 – 1000	20,000	33.3	2,700	10,000
HCE-30,000	200 – 1500	30,000	50.0	4,000	15,000
HCE-40,000	260 – 2000	40,000	67.7	5,400	20,000

MOISTURE REMOVAL PERFORMANCE



NOMENCLATURE:

T_{RI} = Heated reactivation air inlet temperature. °F
T_{RU} = Unheated reactivation air inlet temperature. °F
T_{PI} = Process air inlet temperature. °F
T_{PO} = Process air outlet temperature. °F
M_{PI} = Process air inlet moisture. grains/lb.
M_{PO} = Process air outlet moisture. grains/lb.
M_{RI} = Reactivation air inlet moisture. grains/lb.
K = Variable factor from Figure 3.

EQUATIONS:

1. Process Air Volume, scfm
   V_P = Nominal Face Area sq ft x Face Velocity (fpm)
2. Moisture Removal Rate, grains/hr
   $W = 4.5 \times V_P \times (M_{PI} - M_{PO})$
3. Process Air Temperature Rise
   $(T_{PO} - T_{PI}) = .625\,(M_{PI} - M_{PO}) + K(T_{RI} - T_{RU})$
4. Reactivation Air Volume, scfm
   $$V_R = \frac{V_P}{(T_{RT} - 120)} \times (T_{PO} - T_{PI})$$
5. Reactivation Energy, Btu/hr
   $Q_R = 1.08 \times V_R \times (T_{RI} - T_{RU})$
6. $T_{PO} = T_{PI} + (T_{PO} - T_{PI})$

EXAMPLE:

GIVEN: T_{PI} = 70°F, M_{PI} = 55 grains/lb, V_P = 1875 scfm
T_{RI} = 260°F, T_{RU} = 95°F, Face Area = 3.75 sq ft

FIND: M_{PO}, W, Q_R, T_{PO}

1. Face Velocity = $\frac{1875}{3.75}$ = 500 fpm
2. From Curves, M_{PO} = 13 grains/lb
3. W = 4.5 x 1875 x (55 – 13) = 354,375 grains/hr
4. $T_{PO} - T_{PI}$ = .625 (55 – 13) + .08 (260 – 70) = 41.45°F
5. $V_R = \frac{1875}{(260 - 120)}$ x 41.45 = 555 scfm
6. Q_R = 1.08 x 555 x (260 – 95) = 98,901 Btu/hr
7. T_{PO} = 70 + 41.45 = 111.4°F

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Moisture Load Calculation Sheet

Project Data

Project Name

Location

Application

Calculations made by (name) Date

Approved by (name) Date

Purpose of the project:

Design Conditions

	Weather Extremes Summer	Weather Extremes Winter	Ground water	Internal Conditions Room	Internal Conditions Building
Dry Bulb Temperature					
Dew point					
Humidity Ratio					
Vapor pressure					

Elevation ________ft. above sea level — standard air density = ________lb/cu.ft.

Permeation

	Surface Area (sq.ft.)		Permeance Factor (gr/hr/sq.ft.)		Larger Vapor Pressure (in.hg)		Smaller Vapor Pressure (in.hg)		Permeation Load (gr/hr)	
Wall 1		x		x (		-		) =		gr/hr
Wall 2		x		x (		-		) =		gr/hr
Wall 3		x		x (		-		) =		gr/hr
Wall 4		x		x (		-		) =		gr/hr
Floor		x		x (		-		) =		gr/hr
Ceiling		x		x (		-		) =		gr/hr
								Total		gr/hr

Products, Packaging & Clothing

	Item Entry Rate (lb/hr)		Original Moisture Content (lb/lb)		Final Moisture Content (lb/lb)		Grains Per Pound		Moisture Load (gr/hr)	
Item 1		x (		-		) x	7000	=		gr/hr
Item 2		x (		-		) x	7000	=		gr/hr
Item 3		x (		-		) x	7000	=		gr/hr
Item 4		x (		-		) x	7000	=		gr/hr
							Total			gr/hr

Personnel

	Number Of People		Moisture Load (gr/hr/person)		Moisture Load (gr/hr)	
Seated		x		=		gr/hr
Standing		x		=		gr/hr
Light work		x		=		gr/hr
Moderate work		x		=		gr/hr
Room visitors		x		=		gr/hr
			Total			gr/hr

Open Gas Flame

Gas Burning Rate (cu.ft./hr)		Water Vapor Generation (gr/cu.ft.)		Moisture Load (gr/hr)	
	x		=		gr/hr
		Typical Value 650 gr/cu.ft.			

Wet Surfaces

Wetted Surface Area (sq.ft.) · Latent Heat Tranfer Rate (Btu/sq.ft./in.hg.) · Water Surface Vapor Pressure (in.hg.) · Air Vapor Pressure (in.hg.) · Grains Per Pound · Moisture Load (gr/hr)

$$\left[\frac{\_\_\_\_ \times \_\_\_\_ \times (\_\_\_\_ - \_\_\_\_)}{\_\_\_\_} \right] \times 7000 = \_\_\_\_ \text{ gr/hr}$$

Latent Heat Of Vaporization At The Water Temperature (Btu/lb)

Exterior Walls

	Surface Area (sq.ft.)		Air Infiltration Rate (cu.ft./hr/sq.ft.)		Moisture Outside (gr/lb)		Moisture Inside (gr/lb)		Air Density (lb/cu.ft.)		Moisture Load (gr/hr)	
Wall 1	______	x	______	x (	______	-	______	) x	______	=	______	gr/hr
Wall 2	______	x	______	x (	______	-	______	) x	______	=	______	gr/hr
									Total		______	gr/hr

Cracks

	Crack Length or Component Area (ft or sq.ft.)		Air Infiltration Rate (cu.ft./hr/ft.)		Moisture Outside (gr/lb)		Moisture Inside (gr/lb)		Air Density (lb/cu.ft.)		Moisture Load (gr/hr)	
Door Frames	______	x	______	x (	______	-	______	) x	______	=	______	gr/hr
Windows	______	x	______	x (	______	-	______	) x	______	=	______	gr/hr
Ductwork	______	x	______	x (	______	-	______	) x	______	=	______	gr/hr
									Total		______	gr/hr

Door Openings

	Airflow Velocity (fpm)		Open Area (sq.ft.)		Air Density (lb/cu.ft.)		Time Open (min/hr)		Air Moisture Outside (gr/lb)		Air Moisture Inside (gr/lb)		Moisture Load (gr/hr)	
Door 1	______	x	______	x	______	x	______	x (	______	-	______	) =	______	gr/hr
Door 2	______	x	______	x	______	x	______	x (	______	-	______	) =	______	gr/hr

Airlock Dimensions (ft): Height, Length, Width; Air Density (lb/cu.ft.); Opening Frequency (openings/hr); Air Moisture Outside (gr/lb); Air Moisture Inside (gr/lb)

Airlock Vestibule (____ x ____ x ____) x ______ x ______ x $\left[\dfrac{(\_\_\_\_ - \_\_\_\_)}{2}\right]$ = ______ gr/hr

Total ______ gr/hr

Wall Openings

	Open Area (sq.ft.)		Air Entry Velocity (fpm)		Moisture Outside (gr/lb)		Moisture Inside (gr/lb)		Air Density (lb/cu.ft.)	Minutes Per Hour		Moisture Load (gr/hr)	
Conveyor Openings	______	x	______	x (	______	-	______	) x	______	x 60	=	______	gr/hr
Open Doorways and Holes	______	x	______	x (	______	-	______	) x	______	x 60	=	______	gr/hr
									Total			______	gr/hr

Fresh Air

	Fresh Air Flow Rate (cfm)		Moisture Outside (gr/lb)		Moisture Control Level (gr/lb)		Air Density (lb/cu.ft.)	Minutes Per Hour		Moisture Load (gr/hr)	
Net Fresh Air for Personnel, Exhaust Air Makeup And Room Pressurization	______	x (	______	-	______	) x	______	x 60	=	______	gr/hr

Summary

Permeation	______
Products	______
Personnel	______
Gas Flame	______
Wet Surfaces	______
Exterior Walls	______
Cracks	______
Door Openings	______
Wall Openings	______
Total Internal Moisture Load	______
Fresh Air	______

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AMESBURY, MA 01913-0640 USA

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dhinfo@munters.com

Psychrometric Chart

BAROMETRIC PRESSURE - 29.921 INCHES OF MERCURY

This chart was generated by algorithms created by LINRIC Company, which produce values that agree with those presented in Chapter 6 of the ASHRAE Handbook—Fundamentals 2001.

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