

University of Hilla
Department of Refrigeration and Air Conditioning
Technologies

Fundamental of Air Conditioning and Refrigeration

Compiled by

Mr. Ali Hussein Ghalta

Assist. Lecturer in Department of Refrigeration and Air Conditioning
Technologies

Email: Alihussein@hilla-unc.edu.iq

1.1 INTRODUCTION

The earth is heated. Radiantly Almost everyone worldwide identifies with the warmth of the sun. Whether welcoming the sun on a cool spring day or seeking shelter from the sun on a hot summer day, everyone has developed a response to the sun. In fact, many daily activities are planned around the presence or absence of sunshine. Yet, few people think in terms of the sun being the earth's heating system. Perhaps, because the sun is such a dependable, reliable, and predictable heat energy source, people take it for granted.

Few people, whether laypeople or professionals, make the connection between natural radiant heating and cooling in the great outdoors and environmental conditions for human occupancy in the built environment. But they really can be made to function very similarly. The objective is to capture the best of outdoor comfort for creation in the built environment whenever desired or required. To understand what may be involved, it is instructive to examine just how and why nature's heating system actually works.

To the best of our current knowledge, only the earth is dynamically positioned in relation to the sun to enable exploitation of its resources to support life as we know it. The daily rotation of the earth during its annual orbit around the sun results in the simultaneous 24-hour routine we know as day and night and the climatic change we know as seasons that fans out from the equator. In the natural environment, shelter, insulation (fur, clothing, or structural), life-cycle change, and migration are a few of nature's responses to changing daily and seasonal environmental conditions.

The reason for changing the climate in the built environment is to provide for building use, occupant comfort, and safety of the building and its contents. Yet, to do this in a cost-effective manner requires that the entire structure be viewed as a system in support of the relevant preceding objective. Building design and environmental siting impact the performance of heating and cooling design. The focus of efficiently and effectively heating and cooling a building should include a comprehensive analysis of all the relevant interactions, whether natural or manmade, to ensure that the energy balance is optimized in relation to the objective. The same system logic feeds concern about global warming due to a change in any one of the components in the complex web of environmental balance.

The houses in the past were built to keep the rain, snow, and thieves out with hardly any attention given to heat losses and energy conservation. Houses had little or no insulation, and the structures had numerous cracks through which air leaked. We have seen dramatic changes in the construction of residential and commercial buildings in the 20th century as a result of increased awareness of limited energy resources together with the escalating energy prices and the demand for a higher level of thermal comfort.

Today, most local codes specify the minimum level of insulation to be used in the walls and the roof of new houses, and often require the use of double-pane windows. As a result, today's houses are well insulated, weatherproofed, and nearly air tight, and provide better thermal comfort.

Cooling is more difficult than heating. In the past the principal method was to ward off the sun, coupled with the use of heavy stonework for thermal inertia—actually quite effective in climates with cool nights. During the Middle Ages, architects of castles like the Alhambra supplemented that approach with skillful use of running water, providing some evaporative cooling. In certain parts of the world, some buildings take advantage of cool breezes that blow regularly from the same direction.

Effective air conditioning had to await the development of mechanical refrigeration during the first decades of the twentieth century. Routine installation of central air conditioning systems dates from the 1960s. The oil crises of the 1970s stimulated intensive research on ways of reducing energy costs. The efficiency of existing technologies is being improved, and “new” technologies such as solar energy and day lighting are being tried. Some approaches are turning out to be more successful than others. This evolution is continuing at full speed, aided by advances in materials and computers.

The primary function of an HVAC system is either (1) the generation and maintenance of comfort for occupants in a conditioned space; or (2) the supplying of a set of environmental conditions (high temperature and high humidity, low temperature and high humidity, etc.) for a process or product within a space. Human comfort design conditions are quite different from conditions required in textile mills or for grain storage and vary with factors such as time of year and the activity and clothing levels of the occupants.

1.2 Definitions:

The term ‘refrigeration’ may be defined as the process of removing heat from a substance under controlled conditions. It also includes the process of reducing and maintaining the temperature of a body below the general temperature of its surroundings.

The air conditioning is that branch of engineering science which deals with the study of conditioning of air, supplying and maintaining desirable internal atmospheric conditions for human comfort, irrespective of external conditions. This subject, in its broad sense, also deals with the conditioning of air for industrial purposes, food processing, storage of food and other materials.

To establish proper comfort conditions within a building space, the designer must consider the air temperature and the amount of water vapor in the air. The latter is important because too high a level of water vapor can lead to uncomfortable feeling, sweat does not evaporate fast and people feel too wet and “clammy”. If the water vapor is too low, people suffer from dryness in their respiratory systems, skin dries out and is itchy and there can be a buildup of static electricity, which can cause uncomfortable shocks to people and also disrupt electrical equipment. Evaporation of water into the air can serve to lower the air temperature. Cooling towers are used in conjunction with air conditioners to produce a higher COP. The condenser of the air conditioner is in contact with cooling tower water, which is at a lower temperature than the outside ambient air.

We will need to define several new terms to deal with air-water vapor mixtures, sometimes referred to as psychometrics. In addition, mass and energy balances will be established for both the air and the water. A graphical technique will be introduced which aids in visualizing the processes for the air-water vapor mixture.

1.3 Mixtures

When air and water vapor exist alone at moderate or low pressures, always the case for our conditions, they closely approximate ideal gases. In that case the relationship between pressure, temperature and mass can be expressed as,

$$p_a V = n_a \bar{R} T = m_a R_a T \quad (1)$$

for the air and

$$p_v V = n_v \bar{R} T = m_v R_v T \quad (2)$$

for the water vapor.

In these equations, the subscript a refers to the air and v to the water vapor, n is the number of moles, m the mass, T the absolute temperature, in deg Kelvin or Rankine, $\bar{R}$ is the universal gas constant and R_a and R_v are the ratios of $\bar{R}$ to their respective molecular weights. At low pressure, if air and water vapor are mixed together the interaction between the molecules of the two gases is modest so that the mixture can be considered an ideal gas,

$$pV = (n_a + n_v) \bar{R} T \quad (3)$$

where p is the total pressure of the mixture. Adding equations 1 and 2 and comparing to equation 3 it can be seen that the total pressure in the mixture is just the sum of the pressure of each gas existing alone at the same temperature and in the same total volume, Dalton's Law,

$$p = p_a + p_v \quad (4)$$

When the gases are mixed together p_a and p_v are referred to as the partial pressure of air and water vapor, respectively.

The total enthalpy is the sum of the enthalpy of the air and water vapor,

$$H = H_a + H_v = m_a h_a + m_v h_v \quad (5)$$

Dividing the total enthalpy of the mixture by the mass of the air (note, this is not the total mixture mass), we obtain the enthalpy of the mixture per unit mass of dry air which will be written as h without any subscript,

$$m_a \left(\frac{H}{m_a} \right) = m_a(h) = m h_a + m_v h_v \quad (6)$$

Although this seems an odd choice just now, it will become evident why this is a convenient choice. Dividing equation 6 by the mass of the air in the mixture,

$$h = h_a + \left(\frac{m_v}{m_a} \right) h_v = h_a + w h_v \quad (7)$$

Note the new enthalpy of the mixture we just defined, h , is the sum of the specific enthalpy of the air and the product of the specific enthalpy of the water vapor and the ratio of the mass of water vapor to the mass of air in the mixture. The air mass is sometimes called dry air to remind the reader that only the air without the water vapor mass is being considered. Note, be careful when you use h since different tables can use different states for the zero values of h_v and h_a which are added together to get h of the mixture.

1.4 DRY AND ATMOSPHERIC AIR

Atmospheric air : Air in the atmosphere containing some water vapor (or moisture).

Dry air: Air that contains no water vapor. Water vapor in the air plays a major role in human comfort. Therefore, it is an important consideration in air-conditioning applications.

Water vapor in air behaves as if it existed alone and obeys the ideal-gas relation $P_v = RT$. Then the atmospheric air can be treated as an ideal-gas mixture:

$$P = P_a + P_v \quad (\text{kPa})$$

P_a Partial pressure of dry air

P_v Partial pressure of vapor (vapor pressure)

For water

$$h_g = 2500.9 \text{ kJ/kg at } 0^\circ\text{C}$$

$$c_{p,\text{avg}} = 1.82 \text{ kJ/kg} \cdot ^\circ\text{C at } -10 \text{ to } 50^\circ\text{C range}$$

1.5 Thermodynamic Fundamentals

There are a number of measures of the amount of moisture contained in atmospheric air. Before we begin the discussion of the thermodynamics of moist-air mixtures, we shall define their most important properties and indicate where each is used. However, first, we define the dry-bulb temperature T_d as the temperature of a moist-air mixture measured by a perfectly dry sensor, such as a thermocouple or thermometer. The measures of moist-air– water vapor content are as follows:

1. The partial pressure p_w of water vapor in the homogeneous air–water vapor mixture is used in the definition of relative humidity. Partial pressure is the pressure exerted by one gas component on a mixture of several gases. Dalton’s law of additive pressures states that the sum of the partial pressures of all gases in a mixture is the total pressure exerted by a gas mixture.
2. The relative humidity ϕ is the ratio of the partial pressure of water vapor p_w to the saturation pressure of water vapor p_{sat} at the existing dry-bulb temperature. The saturation pressure is the pressure exerted by saturated water vapor at T_d ; it can be read from steam tables, such as those provided on the CD-ROM. The relative humidity is the most common measure of moisture content—it is the fraction of moisture contained in an air–water vapor mixture relative to what a saturated-air–water vapor mixture at the same temperature would contain.
3. The humidity ratio W is the ratio of the mass of water vapor to the mass of dry air in a moist-air mixture. The humidity ratio is used, e.g., to calculate the enthalpy of moist air. It has units of $\text{lb}_w = \text{lb}_{\text{da}}$ or $\text{kg}_w = \text{kg}_{\text{da}}$, where the subscript “da” denotes dry air.
4. The dew point temperature T_{dew} is the temperature at which a moist-air mixture at a given humidity ratio becomes saturated; i.e., moisture begins to condense, as it is cooled at constant pressure. The dew point is used in design calculations to find the temperature at which condensation must be accounted for in equipment design.

5. The wet-bulb temperature T_{wet} is the temperature of a thermometer with a wetted wick over which air flows at a specified velocity. The wet-bulb temperature is an intermediate quantity used in many psychrometric calculations. Designers use it as a quick indication of the moisture content of air.

Using the ideal gas law, we will now find expressions relating the relative humidity and the humidity ratio to standard thermodynamic properties such as specific volume and pressure. To do this, consider the process 1-2 shown in Fig. 1.1 (the T-s diagram for water) in which superheated water vapor at T_d (point 1) is cooled at constant pressure p_w to a saturated-vapor state at the dew point T_{dew} (point 2). The mixture is assumed to be homogeneous and at thermodynamic equilibrium. In buildings, moist-air mixtures are usually between 0 and 60°C; therefore, the ideal gas laws will give accurate results for building design purposes. (Figure 1.1 also shows the saturation pressure p_{sat} located on the saturation curve at point 3, the temperature corresponding to the dry-bulb temperature of the moist air at condition 1.)

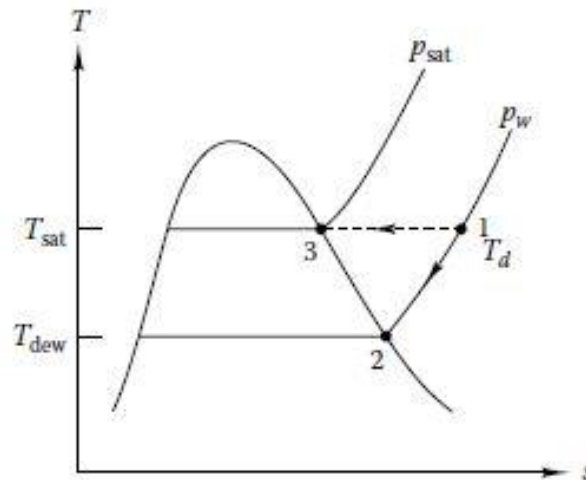


FIGURE 1.1 The T-s diagram for steam showing dry-bulb and dew point temperatures and saturation line

We will begin with the working expression for ϕ :

$$\phi = \frac{p_w}{p_{sat}} \quad (8)$$

The ideal gas law can be used to replace this pressure ratio by a ratio of specific volumes

$$\phi = \frac{RT_d/v_d}{RT_{sat}/v_{sat}} = \frac{v_{sat}}{v_d} \quad (9)$$

If we consider equal volumes V of non-saturated and saturated mixtures, the ratio of the masses of water vapor in the mixtures can be found from the definition of specific volume V/m . Using this substitution in Eq. (9), we have

$$\phi = \frac{m_w}{m_{sat}} \quad (10)$$

Like the relative humidity, the humidity ratio can be expressed in a more useful fashion by using the ideal gas law to replace the mass fraction with a ratio of pressures. We will later show how the saturation pressure of water vapor can be calculated; of course, it can also be looked up in the steam tables in the appendix CD-ROM. The atmospheric pressure depends on the location for which the psychrometric calculations are being done. The ideal gas law written on an extensive basis with the universal gas constant R and molecular weight M is

$$PV = m\left(\frac{R}{M}\right)T \quad (11)$$

Equation (11) can be solved for the mass. For the masses of air and water vapor, we have

$$m_{da} = \frac{p_{da}VM_{da}}{RT} \quad (12)$$

And

$$m_w = \frac{p_wVM_w}{RT} \quad (13)$$

The ratio of m_w to m_{da} is the humidity ratio W by definition; therefore,

$$W = \frac{m_w}{m_{da}} = \frac{p_w M_w}{p_{da} M_{da}} \quad (14)$$

We now substitute the numerical values of the two molecular weights—28.96 for air and 18.02 for water—to get

$$W = \frac{p_w}{p_{da}} \left(\frac{18.02}{28.96} \right) = 0.622 \frac{p_w}{p_{da}} \quad (15)$$

Since the total of the partial pressure of water vapor and dry air is the atmospheric pressure p , by Dalton's law, we can replace p_{da} in Eq. (15) as follows:

$$W = 0.622 \frac{p_w}{p - p_w} \quad (16)$$

Because p_w is much less than p , the humidity ratio and partial pressure are related approximately linearly. We will use this observation later in construction of the psychrometric part.

An alternate form of Eq. (15) using the relative humidity is sometimes useful:

$$W = 0.622 \frac{\phi p_{sat}}{p - \phi p_{sat}} \quad (17)$$

The dry-air basis for the preceding quantities p_{da} and m_{da} is used since the composition of dry air is essentially invariant whereas the composition of moist air obviously varies with the moisture content. The use of the dry-air basis leads to less involved calculations than a moist-air basis, although the latter could be used equally well from a theoretical point of view. For altitude effects, see the tables in the appendices on the CD-ROM.

1.6 Adiabatic Saturation and the Wet-Bulb Temperature

In principle, the relative humidity or the humidity ratio could be used to specify the moisture content of atmospheric air. The problem is that neither can be measured directly with inexpensive and durable instruments. Hence, an indirect method using a straightforward temperature measurement is chosen. Consider the apparatus shown in Fig. 1.2 in

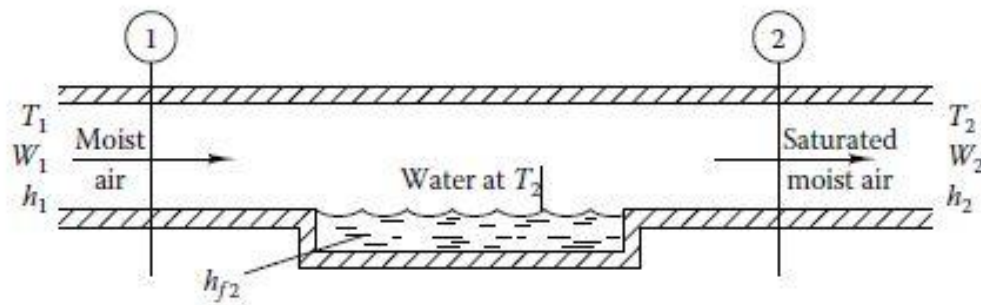


FIGURE 1.2 Schematic diagram showing adiabatic saturation process equipment.

which air at temperature T_1 enters an adiabatic (i.e., no heat transfer occurs across the surface bounding the apparatus) enclosure. The entering moist air becomes saturated as it passes over the liquid due to the evaporation that occurs. Heat is removed from the airstream to accomplish the evaporation, and the outlet temperature T_2 is lower than the inlet temperature as a result.

The adiabatic saturation outlet temperature T_2 is called the thermodynamic wet-bulb temperature. By applying the conservation of mass law to both the dry air and the water systems and by applying the open-system first law to the thermodynamic system consisting of both the water and the air, it is possible to find the unknown inlet humidity ratio W_1 . This device is impractical for field use, so the psychrometer (a wet- and dry-bulb thermometer arrangement described below) is used instead. The wet-bulb temperature so measured is very close to the thermodynamic wet-bulb temperature and is substituted for it in common psychrometric calculations.

Figure 1.3 shows a device for finding the wet-bulb temperature approximately. The sling psychrometer is standard equipment in many laboratories and HVAC field engineering services. As air flows over the moistened wick surrounding the “wet bulb,” water evaporates, extracting heat from the thermometer and lowering its temperature. At equilibrium, the thermometer bulb will be at a temperature at which the rate of evaporation times the heat of vaporization just balances the heat transferred to the bulb. The wet bulb’s temperature is below the dry bulb’s temperature because heat must be transferred to the wet bulb in order to evaporate moisture. Kuehn et al. (1998) offer a detailed heat and mass transfer analysis of the sling psychrometer.

The suggested velocity of air over the wet bulb is 7.5 to 15 ft/s (2.5 to 5 m/s). The wick over the bulb is made of fine cotton, which must be kept clean. Periodic replacement is needed depending on the water mineral content and air cleanliness. The service interval can be increased if distilled water is used. An alternative to the sling psychrometer is a

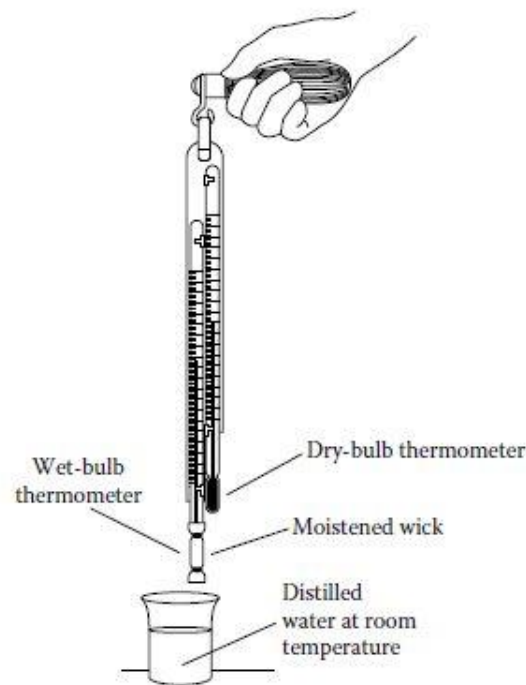


FIGURE 1.3 Sling psychrometer device for conveniently measuring wet- and dry-bulb temperatures

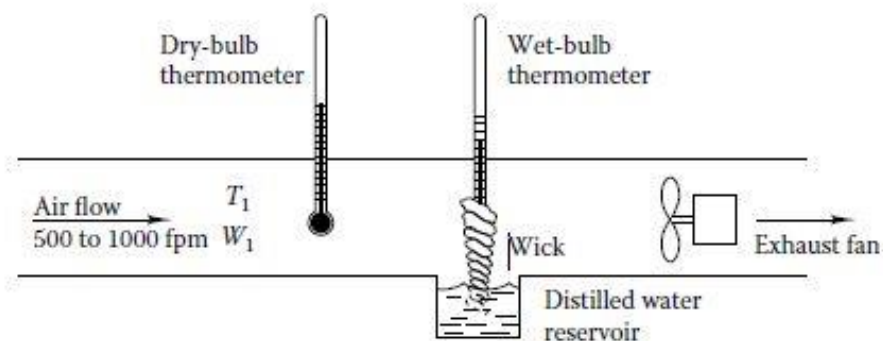


FIGURE 1.4 Psychrometry apparatus for measuring wet- and dry-bulb temperatures.

small duct through which air flows over a stationary wet-bulb thermometer. Figure 1.4 shows this device schematically.

The devices described measure a close approximation to the thermodynamic wet-bulb temperature. To find the relative humidity directly from the wet-bulb temperature (and the concurrently measured dry-bulb temperature), the following approach (Pallady, 1989) is recommended since it is more accurate than the expression given in ASHRAE (1989a,2001).² The algorithm below is reported to be accurate to 0.3 percent on average (not accounting for errors in temperature measurement) and is based on the van der Waals equation. It uses the critical pressure p_c and temperature T_c of water to find the vapor pressure and humidity, as shown below. The relative humidity is calculated as follows from this alternative, working form of Eq. (8) in which the numerator p_w has been replaced by $p_{sat}(T_{wet}) - p_m$ defined as

$$\phi = \left(\frac{p_{sat}(T_{wet}) - p_m}{p_{sat}(T_d)} \right) \quad (18)$$

Where

$p_{sat}(T_{wet})$ = saturation pressure at wet-bulb temperature T_{wet} ; temperature units uses with equations in this section are absolute—°R or K

$p_{sat}(T_d)$ = saturation pressure at dry-bulb temperature T_d , °R (K)

p_m = partial pressure of water vapor due to depression of wet-bulb temperature below dry-bulb temperature; it does not express a physical process result but in just an intermediate quantity useful for relative humidity calculations. And p_m is given by

$$p_m = p \left(\frac{T_d - T_{wet}}{2725} \right) \left(1 + \frac{T_{wet} - 492}{1571} \right) \quad (19US)$$

Or, in SI units

$$p_m = p \left(\frac{T_d - T_{wet}}{1514} \right) \left(1 + \frac{T_{wet} - 273.2}{873} \right) \quad (19SI)$$

The two saturation partial pressures in Eq. (18) are found at dry- and wet-bulb temperatures from

$$p_{sat}(T) = p_c(10^{k(1-\frac{T_c}{T})}) \quad (20)$$

Where $p_{sat}(T)$ is the saturation pressure of water at temperature T . This expression for saturation pressure is to be applied at both the dry- and wet-bulb absolute temperatures. The critical pressure of water p_c is 3226 psia (22.1MPa), and the critical temperature T_c is 1165.67°R (647.30 K).

The parameter K depends on the temperature and is given by

$$K = 4.39553 - 3.469\left(\frac{T}{1000}\right) + 3.072\left(\frac{T}{1000}\right)^2 - 0.8833\left(\frac{T}{1000}\right)^3 \quad (21US)$$

in which T is the dry- or wet-bulb temperature in degrees Rankine. In SI units, this dimensional expression is

$$K = 4.39553 - 6.2442\left(\frac{T}{1000}\right) + 9.953\left(\frac{T}{1000}\right)^2 - 5.151\left(\frac{T}{1000}\right)^3 \quad (21SI)$$

in which T is the dry- or wet-bulb temperature in kelvins.

In this method, the partial pressure of water vapor p_w needed to find ϕ from Eq. (18) is found indirectly from the difference between the saturation partial pressure at the wet-bulb temperature $p_{sat}(T_{wet})$ (note that this is different from p_{sat} used in the previous section) and the partial pressure of water vapor p_m due to the depression of the wet-bulb temperature below the dry-bulb temperature. These two partial pressures are not used directly in design but are convenient for finding the relative humidity (RH).

To summarize the calculation:

- Find both $p_{sat}(T_d)$ and $p_{sat}(T_{wet})$ from Eqs. (20) and (21), using the known wet-bulb and dry-bulb temperatures.
- Next, find p_m from Eq. (19).
- Finally, calculate the relative humidity, using Eq. (18).

The following example illustrates the method.

Example 1.1: Calculation of the Relative Humidity from the Dry- and Wet-Bulb Temperatures

Find the relative humidity and the humidity ratio of moist air at 204°F if the wet-bulb temperature is measured at 190°F by a sling psychrometer. The atmospheric pressure is that of a standard atmosphere, 14.696 psia.

Given: $T_d = 204^\circ\text{F} = 664^\circ\text{R}$ (369 K)

$T_{\text{wet}} = 190^\circ\text{F} = 650^\circ\text{R}$ (361 K)

$P = 14.696$ psia (101 kPa)

Find: ϕ, W

SOLUTION:

The approach described in the previous paragraph will be used. From Eq. (21) the K values at the dry- and wet-bulb temperatures are found to be

$$K(T_d) = 3.18814 \quad K(T_{\text{wet}}) = 3.19611$$

The values of $p_{\text{sat}}(T_d)$ and $p_{\text{sat}}(T_{\text{wet}})$ are then found from Eq. (20):

$$p_{\text{sat}}(T_d) = 3226 \times 10^{3.18814(1 - 1165.67/664)} = 12.58 \text{ psia}$$

$$p_{\text{sat}}(T_{\text{wet}}) = 3226 \times 10^{3.19622(1 - 1165.67/650)} = 9.397 \text{ psia}$$

Finally, Eq. (19) is used to find p_m :

$$P_m = (14.696 \text{ psia}) \left(\frac{664 - 650}{2725} \right) \left(1 + \frac{650 - 492}{1571} \right) = 0.0831 \text{ psia}$$

The relative humidity is found from Eq. (18):

$$\phi = \frac{9.397 - 0.0831}{12.58} = 0.74$$

The humidity ratio is readily found from Eq. (17), where p_{sat} was evaluated at T_d :

$$W = 0.622 \frac{\phi p_{\text{sat}}}{p - \phi p_{\text{sat}}} = 0.622 \left(\frac{0.74 \times 12.58}{14.696 - 0.74 \times 12.58} \right) = 1.08 \text{ lb}_w / \text{lb}_{\text{da}}$$

1.7 Dew Point Temperature

The dew point temperature is shown on the saturation curve in Fig. 1.1. It is the temperature at and below which moist air is saturated, since air at the dew point temperature contains the maximum possible amount of water vapor. The humidity ratio of air remains constant as it is cooled until the dew point is reached. Below the dew point, condensation occurs and the humidity ratio decreases.

The dew point is important in building design since condensation must be avoided within building elements such as walls or on HVAC system components such as chilled water pipes. Of course, condensation is desirable in air conditioning systems since it is the mechanism by which humidity is removed from humid outdoor air before introducing such air into the interior of buildings for ventilation purposes.

The dew point temperature can be found from p_w by using the following expression (for temperatures above the freezing point) (ASHRAE, 1989a):

$$T_{\text{dew}} = 100.45 + 33.193\alpha + 2.19\alpha^2 + 0.1707\alpha^3 + 1.2063p_w^{0.1984} \text{ } ^\circ\text{F} \quad (22\text{US})$$

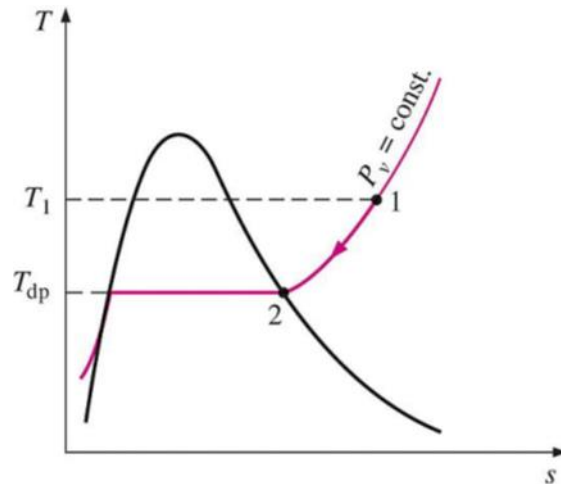
in which $\alpha = \ln p_w$ and p_w has units of pounds per square inch absolute. In SI units, the dew point is

$$T_{\text{dew}} = 6.54 + 14.256\alpha + 0.7389\alpha^2 + 0.0948\alpha^3 + 0.4569p_w^{0.1984} \text{ } ^\circ\text{C} \quad (22\text{SI})$$

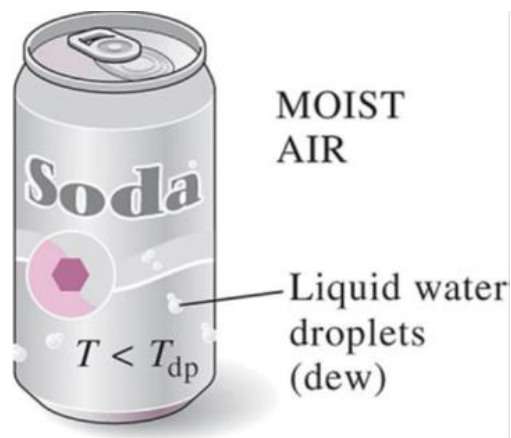
in which $\alpha = \ln p_w$ and p_w has units of kilopascals.

A convenient rule of thumb states that at 70 percent relative humidity a difference of 10°F (5.5°C) between the dew point and dry-bulb temperature exists. A difference of 20°F (11°C) between dew point and dry-bulb temperature corresponds roughly to a relative humidity of 50 percent. These and similar rules for other dew point–dry-bulb differences apply to within a few percent accuracy in the restricted range of dry-bulb temperatures involved in building design.

Constant-pressure cooling of moist air and the dew-point temperature on the T-s diagram of water.



When the temperature of a cold drink is below the dew point temperature of the surrounding air, it “sweats.”



1.8 Moist-Air Enthalpy

The enthalpy of moist air is the sum of the enthalpy of the dry air and of the water vapor comprising the mixture. On an intensive basis (i.e., per unit mass of dry air), the enthalpy h of the mixture is

$$h = h_{da} + Wh_g$$

where h_g is the enthalpy of water vapor.

The enthalpy changes, that is, Δh , are all that ever arise in steady first-law calculations in buildings. As a result, the enthalpy at the temperature on which we base the enthalpy calculation cancels out. For convenience, therefore, the

standard base temperature for moist-air properties is 0°F or 0°C, depending upon which units are being used. In terms of temperature, we express the enthalpy of moist air as

$$h = c_{pa}T_d + W(h_{g, \text{ref}} + c_{pw}T_d) \quad (23)$$

where

h = enthalpy of moist air, Btu/lb (kJ/kg)

c_{pa} = specific heat of dry air, Btu/(lb . °F) [kJ/(kg . °C)]

c_{pw} = specific heat of superheated water vapor, Btu/(lb . °F) [kJ/(kg . °C)]

T_d = dry-bulb temperature, °F (°C)

$h_{g, \text{ref}}$ = enthalpy of water vapor at appropriate reference temperature, Btu/lb (kJ/kg)

W = humidity ratio, lb=lb_{da} (kg=kg_{da})

Substituting numerical values for the two specific heats and for the extrapolated value of enthalpy of saturated water vapor (at 0°F or 0°C), we have the following pair of dimensional equations ($h_{g, \text{ref}}$ =1061.2 Btu/lb is the enthalpy of saturated water vapor at 0°F)

$$h = 0.240T_d + W(1061.2 + 0.444T_d) \text{ Btu/lb}_{da} \quad (24\text{US})$$

in which T_d is in degrees Fahrenheit and

$$h = 1.0T_d + W(2501.3 + 1.86T_d) \text{ kJ/kg}_{da} \quad (24\text{SI})$$

in which T_d is in degrees Celsius; in SI units, $h_{g, \text{ref}}$ =2501.3 kJ/kg, the enthalpy of saturated water vapor at 0°C.

1.9 The Psychrometric Chart and Tables of Air Properties

1.9.1 Construction of the Chart

Although the equations that we have developed for the many properties of moist air are useful for computer calculations, it is convenient to have them plotted in chart form for easy reference during preliminary design of HVAC systems. The standard psychrometric chart serves this purpose. It is also of value in plotting psychrometric processes such as heating, cooling, mixing, drying, or humidification of airstreams.

The psychrometric chart is an xy plot with dry-bulb temperature T_d as the abscissa and the humidity ratio W as the ordinate.⁴ Since these are two independent thermodynamic variables, all other properties of moist air can be expressed as functions of them at a given atmospheric pressure. On the standard psychrometric chart, the following moist-air properties are plotted:

- Relative humidity
- Wet-bulb temperature
- Vapor pressure
- Specific volume
- Enthalpy

Figure 4.5a to e shows how lines of constant properties listed appear on the psychrometric chart. Figure 4.6 is the psychrometric chart in USCS units, and Fig. 4.7 is presented in SI units. The charts shown are for the “normal” temperature range.

The expressions developed above depend on the local atmospheric pressure, which is assumed to be 1 atm if the elevation is not much different from sea level. However, at high altitude, the values of thermodynamic variables can be affected considerably by the lower local barometric pressure. As a result, high-altitude psychrometric charts must be used. In the psychrometric chart, the relative humidity lines curve upward from the lower left to the upper right of the chart. The saturation curve (100 percent relative humidity) can be plotted from data in the steam tables at the desired atmospheric pressure. Since saturated steam properties are given in steam tables as a function of pressure, one uses Eq. (16) to find the psychrometric chart ordinate, the humidity ratio, corresponding to the selected vapor pressure p_w . Lines of constant relative humidity less than 100 percent are plotted by

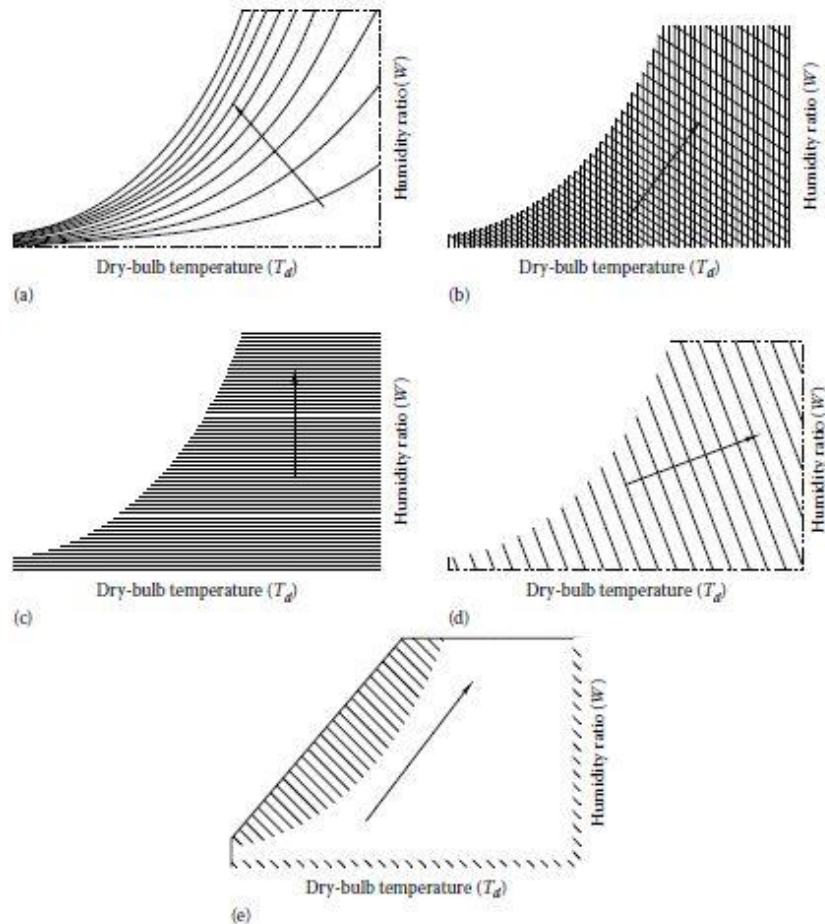


FIGURE 1.5 Skeleton psychrometric chart showing lines of (a) constant relative humidity, (b) constant wet- and dry-bulb temperatures, (c) constant humidity ratio, (d) constant specific volume, and (e) constant enthalpy.

linearly scaling vertical distances between the abscissa and the saturation curve by their respective values of relative humidity. The relative humidity values are written along the curves. The increment between lines is 10 percent in the two charts. On the 100 percent humidity line, the wet-bulb, dry-bulb, and dew point temperatures are identical by their definitions.

Wet-bulb temperature lines slope downward from left to right in the chart. By solving Eqs. (18) to (21) for dry-bulb temperatures as a function of the relative humidity for selected values of the wet-bulb temperature, one can plot lines of constant wet-bulb temperature, as shown in the figure; values are written along the lines. The increment

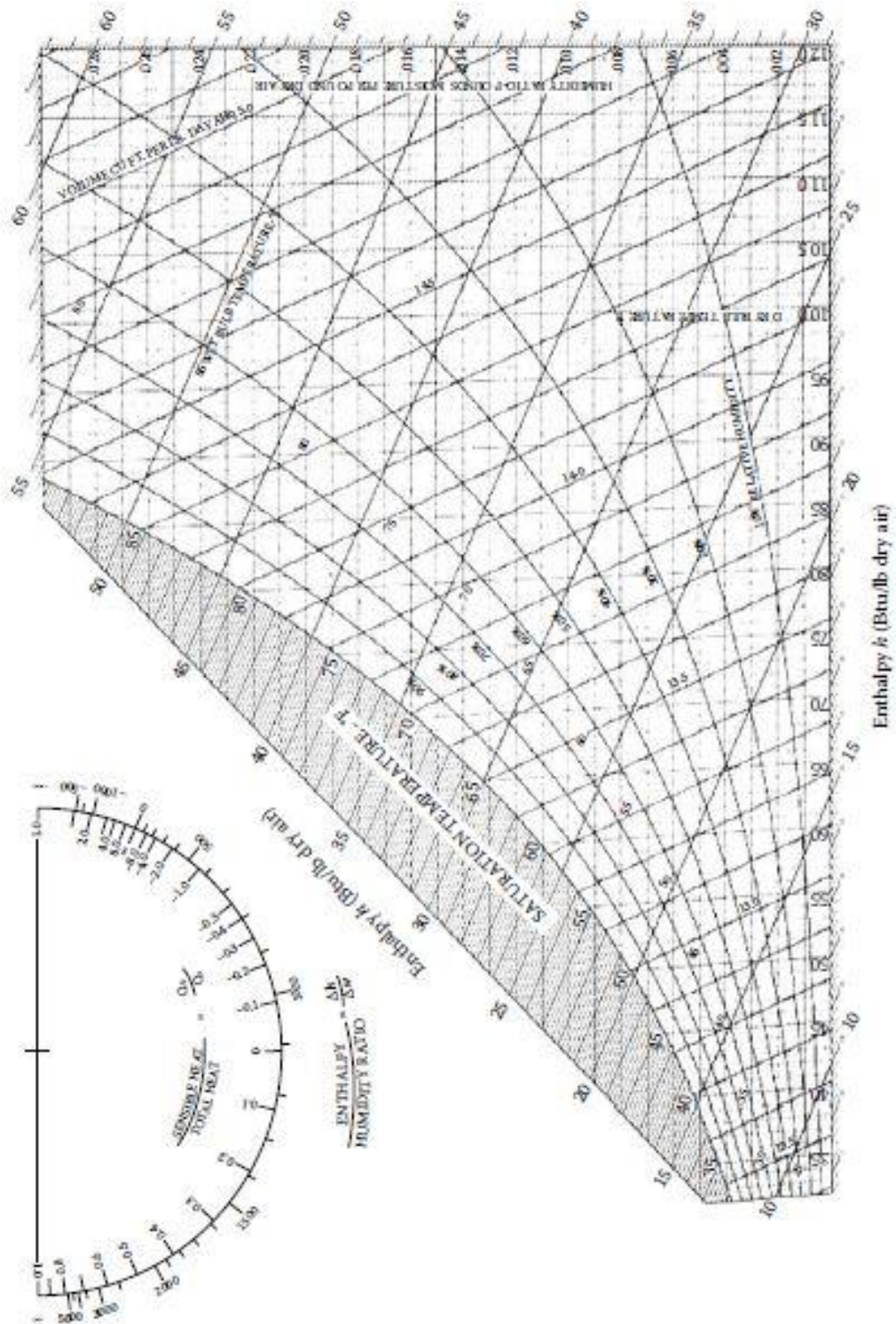


FIGURE 1.6 Psychrometric chart for sea level in USCS units. (Courtesy of ASHRAE, Handbook of Fundamentals, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Atlanta, GA, 1989a. With permission.)

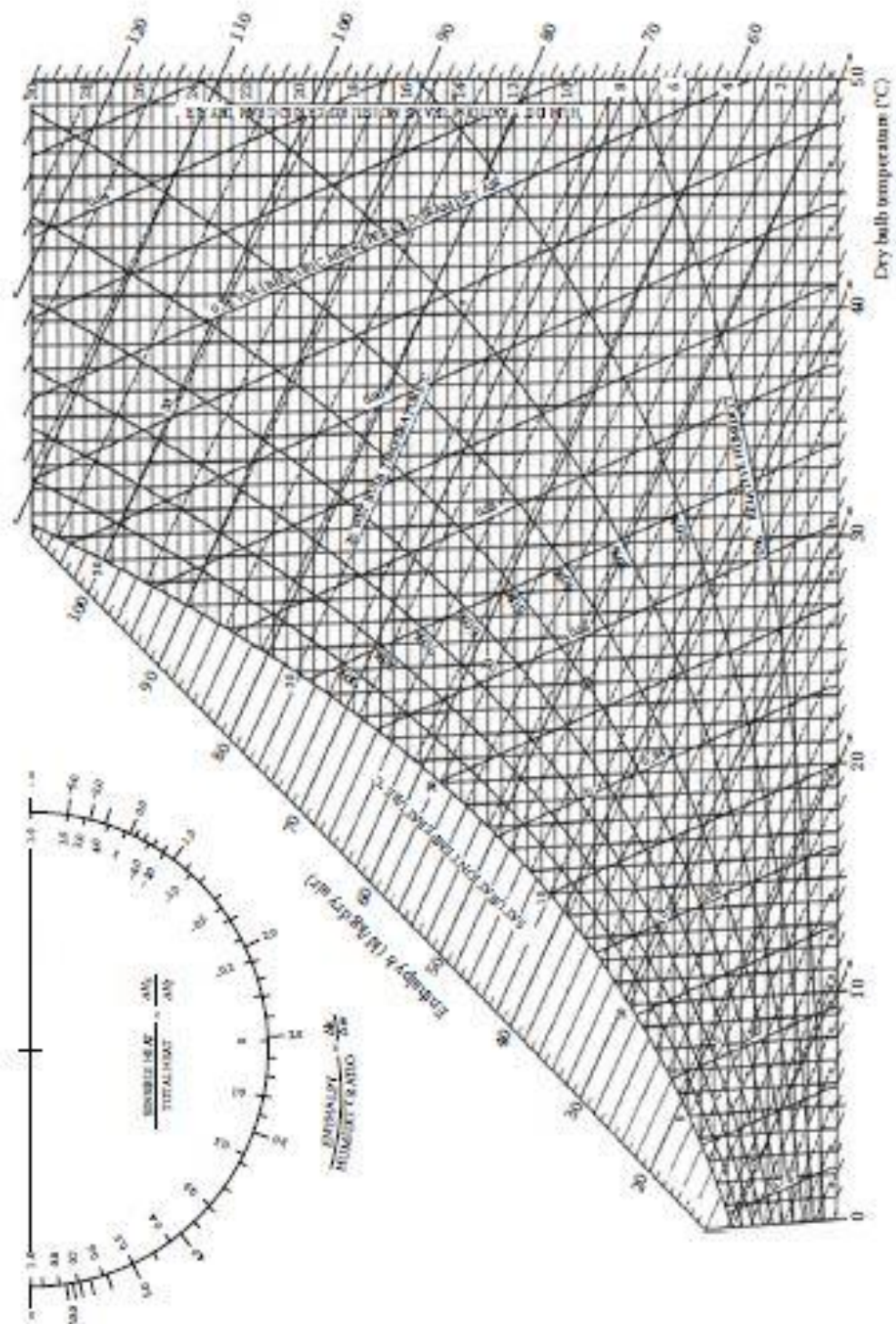


FIGURE 1.7 Psychrometric chart for sea level in SI units. (Courtesy of ASHRAE, Handbook of Fundamentals, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Atlanta, GA, 1989a. With permission.)

between labeled lines in the two charts is 5°F (5°C). The T_{wet} lines are straight due to a fortuitous combination of heat and mass transfer properties for air and water vapor (Stoecker and Jones, 1982).

Enthalpy lines also slope downward from left to right in the chart. Values are noted at the left and right ends of the lines. The increment between marked lines in the two figures is $5 \text{ Btu}=\text{lb}_{\text{da}}$ ($10 \text{ kJ}=\text{kg}_{\text{da}}$). Enthalpy lines and wet-bulb temperature lines are very nearly parallel, but lines of constant wet-bulb temperature are slightly steeper than lines of constant enthalpy. Therefore, it is best to use the auxiliary enthalpy scales (1) along the left diagonal border of the chart and (2) along the bottom and right side of the chart along with a straightedge for accurate readings. Enthalpy lines are plotted by selecting a value of enthalpy, inserting it into Eq. (24), and finding the temperature T_d for several values of the humidity ratio. The results, when plotted, provide the enthalpy lines on the psychrometric chart. For more accurate values, one can use the table of moist-air properties as described in the next section.

Specific-volume lines slope steeply downward from left to right of the chart at about 20° from the vertical. Lines of constant specific volume are found by solving Eq. (12) for the pressure $p_{\text{da}}=p - p_w$ as a function of the temperature and specific volume. Then, for a given specific volume, one finds T - p_w pairs that are plotted. The specific-volume values are written along the lines. The increment between marked lines is $0.5 \text{ ft}^3=\text{lb}_{\text{da}}$ ($0.02 \text{ m}^3=\text{kg}_{\text{da}}$) in the two charts.

Vapor pressure p_w lines are horizontal and parallel to lines of constant humidity ratio. Values are found by solving Eq. (16) for the vapor pressure as a function of the humidity ratio. In some psychrometric charts (not Figs. 1.6 and 1.7), these values are written along an auxiliary ordinate which is nearly linear. Likewise, dew point temperature lines are horizontal and parallel to lines of constant humidity ratio. Values are found from Eq. (22) and are optionally written along an auxiliary, nonlinear ordinate. However, the dew point is just as easily read along the saturation curve itself. Details of the geometric construction of the psychrometric chart are contained in Kuehn et al. (1998).

1.9.2 Use of the Chart

The psychrometric chart can be used in analyzing many processes involving moist atmospheric air. This section contains three examples of such calculations, and the following sections consider a number of processes which we will find to be important in the design of heating and cooling systems in buildings.

Example 1.2: Use of the Psychrometric Chart

Find the five listed properties for atmospheric air in a building maintained at 70°F (21°C). The wet-bulb temperature is measured to be 60°F (15.5°C), a typical interior condition in winter in heating climates.

Given: $T_d = 70^\circ\text{F}$

$T_{\text{wet}} = 60^\circ\text{F}$

Figure: See Fig. 1.6.

Assumptions: The atmospheric pressure is the standard atmosphere, 14.7 psia.

Find: ϕ , h , v , T_{dew} , p_w

Solution:

This problem can be solved easily by using the USCS psychrometric chart. Note that only one of the two independent variables (T_d) used to plot the chart is known; but two different independent properties are known. Hence, the problem can be solved.

First, find the intersection of the 70°F dry-bulb vertical line and the 60°F wet-bulb line. This intersection is approximately at the 57 percent relative humidity point.

At the same intersection point, the following values can be read by interpolation. The humidity ratio is 0.0088 lb=lb_{da}. The enthalpy is read from the scales at the upper left and bottom of the chart. The enthalpy is 26.5 Btu=lb_{da}. By interpolation the specific volume is 13.55 ft³=lb_{da}. A better approach than this rough interpolation is to calculate the specific volume from the partial pressure p_w and the dry-bulb temperature by using the ideal gas law:

$$v = \frac{R_{air} T_d}{p - p_w}$$

The partial pressure of water vapor p_w cannot be found from the chart but is easy to find from Eq. (16) since the humidity ratio W is already known:

$$p_w = \frac{p}{1 + 0.622/W} = \frac{14.696 \text{ psia}}{1 + 0.622/0.0088} = 0.21 \text{ psia}$$

An alternate method of finding p_w is to use the basic equation for the relative humidity, Eq. (8).

$$P_w = \phi p_{\text{sat}} = 0.57(0.3633 \text{ psia}) = 0.21 \text{ psia}$$

in which the saturation pressure has been read from the last column of Table 4.1 with appropriate units conversion. (Table 4.1 is discussed in detail shortly.)

The value of p_w can be used in the ideal gas law above to find a more accurate value of the specific volume.

The dew point is found at the intersection of the humidity ratio line (0.0088) and the saturation curve. The dew point is 54°F.

Example 1.3: Sensible Cooling Process on Psychrometric Chart

Moist air is cooled from 40°C (104°F) and 30 percent relative humidity to 30°C (86°F). Does moisture condense? What are the values of the relative humidity and the humidity ratio at the process endpoint?

Given: $T_{d1} = 40^\circ\text{C}$

$\phi_1 = 0.30$

$T_{d2} = 30^\circ\text{C}$

Figure: See Fig. 1.8, process 1-2.

Find: W_2, ϕ_2

SOLUTION:

First, we must determine whether moisture condenses. In terms of psychrometric variables, we need to determine if the final temperature is above or below the dew point.

By referring to the SI psychrometric chart (Fig. 1.7), we observe that the dew point temperature $T_{\text{dew}}=19^{\circ}\text{C}$. Therefore, moisture does not condense, and the problem can be solved using the properties of moist air as plotted in Fig. 1.7 directly. This is a sensible cooling process, shown by points 1-2 in Fig. 1.8a; no latent heat removal is involved.

Second, we observe that the humidity ratio will not change since no moisture condensation has occurred. Therefore, the humidity ratio at the end of the process will be identical to that at the start $0.014 \text{ kg}=\text{kg}_{\text{da}}$.

Third, the relative humidity will change since the relative ability of the cooler air to hold moisture is less than that of the warm air at the start of the process. Since this process is a constant-humidity-ratio process, it is plotted as a horizontal line on the psychrometric chart. One moves to the left from the initial point until the intersection with the $T_d = 30^{\circ}\text{C}$ vertical line. At this point, we read a relative humidity value of 53 percent.

Example 1.4: Latent and Sensible Cooling Process on Psychrometric Chart
Moist air is cooled from 40°C (104°F) and 30 percent relative humidity to 15°C (50°F). Does moisture condense? What are the values of the relative humidity and the humidity ratio at the process endpoint?

Given: $T_{d1}=40^{\circ}\text{C}$

$\phi_1=0.30$

$T_{d2}=15^{\circ}\text{C}$

Figure: See Fig. 1.8a, process 1-3.

Find: W_3, ϕ_3

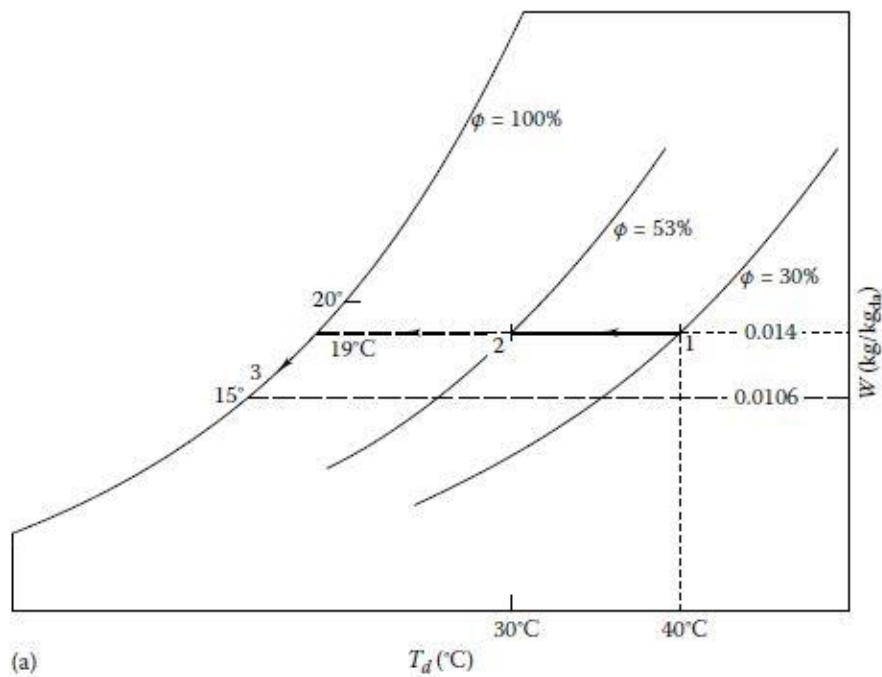


FIGURE 1.8 a. Cooling processes plotted on psychrometric chart: (a) Sensible cooling only (1–2); sensible and latent cooling (1–3).

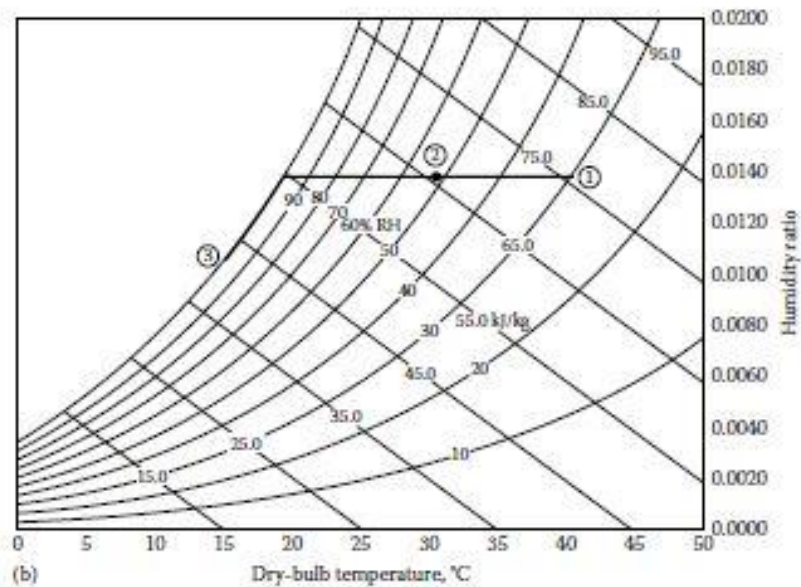


FIGURE 1.8 (b) Sensible and latent process as calculated with HCB software.

SOLUTION:

From the previous example, the dew point is known to be 19°C for the moist air at the beginning of the process. Therefore, moisture condenses since the outlet temperature (15°C) for this case is below this dew point. This is both a sensible and a latent cooling process.

Inspection of process 1-3 in Fig. 1.8 shows that the humidity ratio changes since moisture condensation has occurred. The saturated air at the exit temperature of 15°C has a humidity ratio, according to Fig. 1.7, of 0.0106 kg/kg_{da}.

Finally, the relative humidity changes to the saturated value, 100 percent, as shown in Fig. 1.8a. This process is a two-part process: (1) constant humidity-ratio cooling until saturation and (2) constant-relative-humidity cooling with condensation of moisture. Figure 1.8b shows the process as it is displayed within the HCB software.

1.10 Tabulations of Moist Air Properties

Tables of thermodynamic properties of air (or computer software) are needed for calculations in which the accuracy of the psychrometric chart is inadequate. Table 1.1 is an excerpt of the complete moist-air table. This table contains air properties at 1 atm of pressure. It includes the following data from left to right: The second column is the saturated humidity ratio W_{sat} followed by columns of specific volume v , enthalpy h , and entropy s , each at three conditions: (1) dry air (denoted by “da”), (2) difference between dry air and saturated air (denoted by “d, s”), and (3) saturated air (denoted by “sat”). The reference state for enthalpy and entropy is 0°F and 1 atm pressure in Table 1.1. For moist air that is neither saturated nor perfectly dry, the following equations are used to find the enthalpy and specific volume. Other properties are found similarly

$$h = h_{da} + \frac{W}{W_{sat}} h_{d,s} \quad (25a)$$

$$v = v_{da} + \frac{W}{W_{sat}} v_{d,s} \quad (25b)$$

Temp., °F	Saturated Humidity Ratio W_{sat} lb _m /lb _m	Vol. ratio, ft ³ /lb Dry Air			Enthalpy, Btu/lb Dry Air			Entropy, Btu/(lb Dry Air · °R)			Enthalpy* h_f Btu/lb	Entropy* s_f Btu/(lb · °R)	Vapor Pressure [†] p_{sat} in Hg
		v_{da}	v_{da}	v_{da}	h_{da}	h_{da}	h_{da}	s_{da}	s_{da}	s_{da}			
10	0.0013158	11.832	0.025	11.857	2.402	1.402	3.804	0.00517	0.00315	0.00832	-154.13	-0.3141	0.062901
12	0.0014544	11.883	0.028	11.910	2.882	1.590	4.433	0.00619	0.00347	0.00966	-153.17	-0.3120	0.069511
14	0.0016062	11.933	0.031	11.964	3.363	1.714	5.077	0.00721	0.00381	0.01102	-152.20	-0.3100	0.076751
16	0.0017724	11.984	0.034	12.018	3.843	1.892	5.736	0.00822	0.00419	0.01241	-151.22	-0.3079	0.084673
18	0.0019543	12.035	0.038	12.072	4.324	2.068	6.412	0.00923	0.00460	0.01383	-150.25	-0.3059	0.093334
20	0.0021531	12.085	0.042	12.127	4.804	2.303	7.107	0.01023	0.00505	0.01528	-149.27	-0.3038	0.102798
22	0.0023703	12.136	0.046	12.182	5.285	2.537	7.822	0.01123	0.00554	0.01677	-148.28	-0.3018	0.113130
24	0.0026073	12.186	0.051	12.237	5.765	2.793	8.568	0.01223	0.00607	0.01830	-147.30	-0.2997	0.124396
26	0.0028660	12.237	0.056	12.293	6.246	3.073	9.318	0.01322	0.00665	0.01987	-146.30	-0.2977	0.136684
28	0.0031480	12.287	0.062	12.349	6.726	3.378	10.104	0.01420	0.00728	0.02148	-145.31	-0.2956	0.150066
30	0.0034552	12.338	0.068	12.406	7.206	3.711	10.917	0.01519	0.00796	0.02315	-144.31	-0.2936	0.164631
...													
55	0.009233	12.970	0.192	13.162	13.213	10.016	23.229	0.02715	0.02048	0.04763	23.11	0.0459	0.43992
56	0.009580	12.995	0.200	13.195	13.453	10.397	23.890	0.02762	0.02122	0.04884	24.11	0.0478	0.45205
57	0.009938	13.021	0.207	13.228	13.694	10.790	24.484	0.02808	0.02198	0.05006	25.11	0.0497	0.46370

Table 1. Thermodynamic Properties of Moist Air at Standard Barometric Pressure (14.696 psia)

58	0.010309	13.046	0.216	13.362	13.934	11.197	25.131	0.02855	0.02277	0.05132	26.11	0.0517	0.48589
59	0.010692	13.071	0.224	13.395	14.174	11.618	25.792	0.02901	0.02358	0.05259	27.11	0.0536	0.50363
60	0.011087	13.096	0.233	13.329	14.415	12.052	26.467	0.02947	0.02442	0.05389	28.11	0.0555	0.52193
61	0.011496	13.122	0.242	13.364	14.655	12.502	27.157	0.02994	0.02528	0.05522	29.12	0.0575	0.54082
62	0.011919	13.147	0.251	13.398	14.895	12.966	27.862	0.03040	0.02617	0.05657	30.11	0.0594	0.56032
63	0.012355	13.172	0.261	13.433	15.135	13.446	28.582	0.03086	0.02709	0.05795	31.11	0.0613	0.58041
64	0.012805	13.198	0.271	13.468	15.376	13.942	29.318	0.03132	0.02804	0.05936	32.11	0.0632	0.60113
65	0.013270	13.223	0.281	13.504	15.616	14.454	30.071	0.03178	0.02902	0.06080	33.11	0.0651	0.62252
66	0.013750	13.248	0.292	13.540	15.856	14.983	30.840	0.03223	0.03003	0.06226	34.11	0.0670	0.64454
67	0.014246	13.273	0.303	13.577	16.097	15.530	31.626	0.03269	0.03107	0.06376	35.11	0.0689	0.66725
68	0.014758	13.299	0.315	13.613	16.337	16.094	32.431	0.03315	0.03214	0.06529	36.11	0.0708	0.69065
69	0.015286	13.324	0.326	13.650	16.577	16.677	33.254	0.03360	0.03325	0.06685	37.11	0.0727	0.71479
70	0.015832	13.349	0.339	13.688	16.818	17.279	34.097	0.03406	0.03438	0.06844	38.11	0.0746	0.73966
71	0.016395	13.375	0.351	13.726	17.058	17.901	34.959	0.03451	0.03556	0.07007	39.11	0.0765	0.76528
72	0.016976	13.400	0.365	13.764	17.299	18.543	35.841	0.03496	0.03677	0.07173	40.11	0.0783	0.79167
73	0.017575	13.425	0.378	13.803	17.539	19.204	36.743	0.03541	0.03801	0.07343	41.11	0.0802	0.81882
74	0.018194	13.450	0.392	13.843	17.779	19.889	37.668	0.03586	0.03930	0.07516	42.11	0.0821	0.84684
75	0.018833	13.476	0.407	13.882	18.020	20.595	38.615	0.03631	0.04062	0.07694	43.11	0.0840	0.87567

Source: Courtesy of ASHRAE, Handbook of Fundamentals, American Society of Heating Refrigerating and Air-Conditioning Engineers, Atlanta, GA, 1988a.
With permission.)

* Saturated conditions.

† Multiply $\ln h_g$ by 0.49115 to convert to ρ_{sat} .

