CHAPTER 6

PSYCHROMETRICS

PSYCHROMETRICS uses thermodynamic properties to analyze conditions and processes involving moist air. This chapter discusses perfect gas relations and their use in common heating, cooling, and humidity control problems. Formulas developed by Hyland and Wexler (1983a) and discussed by Olivieri (1996) may be used where greater precision is required.

Hyland and Wexler (1983a, 1983b) developed formulas for thermodynamic properties of moist air and water. However, perfect gas relations can be substituted in most air-conditioning problems. Kuehn et al. (1998) showed that errors are less than 0.7% in calculating humidity ratio, enthalpy, and specific volume of saturated air at standard atmospheric pressure for a temperature range of −50 to 50°C. Furthermore, these errors decrease with decreasing pressure.

COMPOSITION OF DRY AND MOIST AIR

Atmospheric air contains many gaseous components as well as water vapor and miscellaneous contaminants (e.g., smoke, pollen, and gaseous pollutants not normally present in free air far from pollution sources).

Dry air is atmospheric air with all water vapor and contaminants removed. Its composition is relatively constant, but small variations in the amounts of individual components occur with time, geographic location, and altitude. Harrison (1965) lists the approximate percentage composition of dry air by volume as: nitrogen, 78.084; oxygen, 20.9476; argon, 0.934; carbon dioxide, 0.0314; neon, 0.001 818; helium, 0.000 524; methane, 0.000 15; sulfur dioxide, 0 to 0.0001; hydrogen, 0.000 05; and minor components such as krypton, xenon, and ozone, 0.0002. The relative molecular mass for dry air is 28.9645, based on the carbon-12 scale (Harrison 1965). The gas constant for dry air, based on the carbon-12 scale, is

$$R_{da} = 8314.41/28.9645 = 287.055 \text{ J/(kg}_{da} \cdot \text{K)} \quad (1)$$

Moist air is a binary (two-component) mixture of dry air and water vapor. The amount of water vapor varies from zero (dry air) to a maximum that depends on temperature and pressure. **Saturation** is a state of neutral equilibrium between moist air and the condensed water phase (liquid or solid); unless otherwise stated, it assumes a flat interface surface between moist air and the condensed phase. Saturation conditions change when the interface radius is very small (e.g., with ultrafine water droplets). The relative molecular mass of water is 18.01528 on the carbon-12 scale. The gas constant for water vapor is

$$R_w = 8314.41/18.015 \ 28 = 461.520 \text{ J/(kg}_w \cdot \text{K)} \quad (2)$$

U.S. STANDARD ATMOSPHERE

The temperature and barometric pressure of atmospheric air vary considerably with altitude as well as with local geographic and weather conditions. The standard atmosphere gives a standard of reference for estimating properties at various altitudes. At sea level, standard temperature is 15°C; standard barometric pressure is 101.325 kPa. Temperature is assumed to decrease linearly with increasing altitude throughout the troposphere (lower atmosphere), and to be constant in the lower reaches of the stratosphere. The lower atmosphere is assumed to consist of dry air that behaves as a perfect gas. Gravity is also assumed constant at the standard value, 9.806 65 m/s². Table 1 summarizes property data for altitudes to 10 000 m.

Pressure values in Table 1 may be calculated from

$$p = 101.325(1 - 2.255 \ 77 \times 10^{-5} Z)^{5.2559} \quad (3)$$

The equation for temperature as a function of altitude is

$$t = 15 - 0.0065Z \quad (4)$$

where
    Z = altitude, m
    p = barometric pressure, kPa
    t = temperature, °C

Equations (3) and (4) are accurate from −5000 m to 11 000 m. For higher altitudes, comprehensive tables of barometric pressure

Table 1 Standard Atmospheric Data for Altitudes to 10 000 m

Altitude, m	Temperature, °C	Pressure, kPa
−500	18.2	107.478
0	15.0	101.325
500	11.8	95.461
1 000	8.5	89.875
1 500	5.2	84.556
2 000	2.0	79.495
2 500	−1.2	74.682
3 000	−4.5	70.108
4 000	−11.0	61.640
5 000	−17.5	54.020
6 000	−24.0	47.181
7 000	−30.5	41.061
8 000	−37.0	35.600
9 000	−43.5	30.742
10 000	−50	26.436

and other physical properties of the standard atmosphere can be found in NASA (1976).

THERMODYNAMIC PROPERTIES OF MOIST AIR

Table 2, developed from formulas by Hyland and Wexler (1983a, 1983b), shows values of thermodynamic properties of moist air based on the **thermodynamic temperature scale**. This ideal scale differs slightly from practical temperature scales used for physical measurements. For example, the standard boiling point for water (at 101.325 kPa) occurs at 99.97°C on this scale rather than at the traditional 100°C. Most measurements are currently based on the International Temperature Scale of 1990 (ITS-90) (Preston-Thomas 1990).

The following properties are shown in Table 2:

t = Celsius temperature, based on thermodynamic temperature scale and expressed relative to absolute temperature T in kelvins (K) by the following relation:

$$T = t + 273.15$$

W_s = humidity ratio at saturation; gaseous phase (moist air) exists in equilibrium with condensed phase (liquid or solid) at given temperature and pressure (standard atmospheric pressure). At given values of temperature and pressure, humidity ratio W can have any value from zero to W_s.

v_{da} = specific volume of dry air, m^3/kg_{da}.

v_{as} = $v_s - v_{da}$, difference between specific volume of moist air at saturation and that of dry air, m^3/kg_{da}, at same pressure and temperature.

v_s = specific volume of moist air at saturation, m^3/kg_{da}.

h_{da} = specific enthalpy of dry air, kJ/kg_{da}. In Table 2, h_{da} has been assigned a value of 0 at 0°C and standard atmospheric pressure.

h_{as} = $h_s - h_{da}$, difference between specific enthalpy of moist air at saturation and that of dry air, kJ/kg_{da}, at same pressure and temperature.

h_s = specific enthalpy of moist air at saturation, kJ/kg_{da}.

s_{da} = specific entropy of dry air, $kJ/(kg_{da} \cdot K)$. In Table 2, s_{da} is assigned a value of 0 at 0°C and standard atmospheric pressure.

s_s = specific entropy of moist air at saturation $kJ/(kg_{da} \cdot K)$.

THERMODYNAMIC PROPERTIES OF WATER AT SATURATION

Table 3 shows thermodynamic properties of water at saturation for temperatures from −60 to 160°C, calculated by the formulations described by Hyland and Wexler (1983b). Symbols in the table follow standard steam table nomenclature. These properties are based on the thermodynamic temperature scale. The enthalpy and entropy of saturated liquid water are both assigned the value zero at the triple point, 0.01°C. Between the triple-point and critical-point temperatures of water, two states (**saturated liquid** and **saturated vapor**) may coexist in equilibrium.

The **water vapor saturation pressure** is required to determine a number of moist air properties, principally the saturation humidity ratio. Values may be obtained from Table 3 or calculated from the following formulas (Hyland and Wexler 1983b).

The saturation pressure over **ice** for the temperature range of −100 to 0°C is given by

$$\ln p_{ws} = C_1/T + C_2 + C_3 T + C_4 T^2 + C_5 T^3$$
$$+ C_6 T^4 + C_7 \ln T \tag{5}$$

where

C_1 = −5.674 535 9 E+03
C_2 = 6.392 524 7 E+00
C_3 = −9.677 843 0 E−03
C_4 = 6.221 570 1 E−07
C_5 = 2.074 782 5 E−09

C_6 = −9.484 024 0 E−13
C_7 = 4.163 501 9 E+00

The saturation pressure over **liquid water** for the temperature range of 0 to 200°C is given by

$$\ln p_{ws} = C_8/T + C_9 + C_{10} T + C_{11} T^2$$
$$+ C_{12} T^3 + C_{13} \ln T \tag{6}$$

where

C_8 = −5.800 220 6 E+03
C_9 = 1.391 499 3 E+00
C_{10} = −4.864 023 9 E−02
C_{11} = 4.176 476 8 E−05
C_{12} = −1.445 209 3 E−08
C_{13} = 6.545 967 3 E+00

In both Equations (5) and (6),

p_{ws} = saturation pressure, Pa
T = absolute temperature, K = °C + 273.15

The coefficients of Equations (5) and (6) were derived from the Hyland-Wexler equations. Because of rounding errors in the derivations and in some computers' calculating precision, results from Equations (5) and (6) may not agree precisely with Table 3 values.

The vapor pressure p_s of water in saturated moist air differs negligibly from the saturation vapor pressure p_{ws} of pure water at the same temperature. Consequently, p_s can be used in equations in place of p_{ws} with very little error:

$$p_s = x_{ws} p$$

where x_{ws} is the mole fraction of water vapor in saturated moist air at temperature t and pressure p, and p is the total barometric pressure of moist air.

HUMIDITY PARAMETERS

Basic Parameters

Humidity ratio W (alternatively, the moisture content or mixing ratio) of a given moist air sample is defined as the ratio of the mass of water vapor to the mass of dry air in the sample:

$$W = M_w/M_{da} \tag{7}$$

W equals the mole fraction ratio x_w/x_{da} multiplied by the ratio of molecular masses (18.015 28/28.9645 = 0.621 98):

$$W = 0.621 98 \, x_w/x_{da} \tag{8}$$

Specific humidity γ is the ratio of the mass of water vapor to total mass of the moist air sample:

$$\gamma = M_w/(M_w + M_{da}) \tag{9a}$$

In terms of the humidity ratio,

$$\gamma = W/(1 + W) \tag{9b}$$

Absolute humidity (alternatively, water vapor density) d_v is the ratio of the mass of water vapor to total volume of the sample:

$$d_v = M_w/V \tag{10}$$

Density ρ of a moist air mixture is the ratio of total mass to total volume:

$$\rho = (M_{da} + M_w)/V = (1/v)(1 + W) \tag{11}$$

where v is the moist air specific volume, m^3/kg_{da}, as defined by Equation (26).

Humidity Parameters Involving Saturation

The following definitions of humidity parameters involve the concept of moist air saturation:

Saturation humidity ratio $W_s(t, p)$ is the humidity ratio of moist air saturated with respect to water (or ice) at the same temperature t and pressure p.

Degree of saturation μ is the ratio of air humidity ratio W to humidity ratio W_s of saturated moist air at the same temperature and pressure:

$$\mu = \frac{W}{W_s}\bigg|_{t,p} \tag{12}$$

Relative humidity ϕ is the ratio of the mole fraction of water vapor x_w in a given moist air sample to the mole fraction x_{ws} in an air sample saturated at the same temperature and pressure:

$$\phi = \frac{x_w}{x_{ws}}\bigg|_{t,p} \tag{13}$$

Combining Equations (8), (12), and (13),

$$\mu = \frac{\phi}{1 + (1 - \phi)W_s/0.62198} \tag{14}$$

Dew-point temperature t_d is the temperature of moist air saturated at pressure p, with the same humidity ratio W as that of the given sample of moist air. It is defined as the solution $t_d(p, W)$ of the following equation:

$$W_s(p, t_d) = W \tag{15}$$

Thermodynamic wet-bulb temperature t^* is the temperature at which water (liquid or solid), by evaporating into moist air at dry-bulb temperature t and humidity ratio W, can bring air to saturation adiabatically at the same temperature t^* while total pressure p is constant. This parameter is considered separately in the section on Thermodynamic Wet-Bulb and Dew-Point Temperature.

PERFECT GAS RELATIONSHIPS FOR DRY AND MOIST AIR

When moist air is considered a mixture of independent perfect gases (i.e., dry air and water vapor), each is assumed to obey the perfect gas equation of state as follows:

Dry air: $\quad p_{da}V = n_{da}RT \tag{16}$

Water vapor: $\quad p_w V = n_w RT \tag{17}$

where

p_{da} = partial pressure of dry air
p_w = partial pressure of water vapor
V = total mixture volume
n_{da} = number of moles of dry air
n_w = number of moles of water vapor
R = universal gas constant, 8314.41 J/(kg mol·K)
T = absolute temperature, K

The mixture also obeys the perfect gas equation:

$$pV = nRT \tag{18}$$

or

$$(p_{da} + p_w)V = (n_{da} + n_w)RT \tag{19}$$

where $p = p_{da} + p_w$ is the total mixture pressure and $n = n_{da} + n_w$ is the total number of moles in the mixture. From Equations (16) to (19), the mole fractions of dry air and water vapor are, respectively,

$$x_{da} = p_{da}/(p_{da} + p_w) = p_{da}/p \tag{20}$$

and

$$x_w = p_w/(p_{da} + p_w) = p_w/p \tag{21}$$

From Equations (8), (20), and (21), the **humidity ratio** W is

$$W = 0.62198\frac{p_w}{p - p_w} \tag{22}$$

The degree of saturation μ is defined in Equation (12), where

$$W_s = 0.62198\frac{p_{ws}}{p - p_{ws}} \tag{23}$$

The term p_{ws} represents the saturation pressure of water vapor in the absence of air at the given temperature t. This pressure p_{ws} is a function only of temperature and differs slightly from the vapor pressure of water in saturated moist air.

The **relative humidity** ϕ is defined in Equation (13). Substituting Equation (21) for x_w and x_{ws},

$$\phi = \frac{p_w}{p_{ws}}\bigg|_{t,p} \tag{24}$$

Substituting Equation (23) for W_s into Equation (14),

$$\phi = \frac{\mu}{1 - (1 - \mu)(p_w/p)} \tag{25}$$

Both ϕ and μ are zero for dry air and unity for saturated moist air. At intermediate states, their values differ, substantially so at higher temperatures.

The **specific volume** v of a moist air mixture is expressed in terms of a unit mass of dry air:

$$v = V/M_{da} = V/(28.9645 n_{da}) \tag{26}$$

where V is the total volume of the mixture, M_{da} is the total mass of dry air, and n_{da} is the number of moles of dry air. By Equations (16) and (26), with the relation $p = p_{da} + p_w$,

$$v = \frac{RT}{28.9645(p - p_w)} = \frac{R_{da}T}{p - p_w} \tag{27}$$

Using Equation (22),

$$v = \frac{RT(1 + 1.6078W)}{28.964p} = \frac{R_{da}T(1 + 1.6078W)}{p} \tag{28}$$

In Equations (27) and (28), v is specific volume, T is absolute temperature, p is total pressure, p_w is partial pressure of water vapor, and W is humidity ratio.

In specific units, Equation (28) may be expressed as

$$v = 0.2871(t + 273.15)(1 + 1.6078W)/p$$

where

v = specific volume, m^3/kg_{da}
t = dry-bulb temperature, °C
W = humidity ratio, kg_w/kg_{da}
p = total pressure, kPa

The **enthalpy** of a mixture of perfect gases equals the sum of the individual partial enthalpies of the components. Therefore, the specific enthalpy of moist air can be written as follows:

$$h = h_{da} + W h_g \tag{29}$$

where h_{da} is the specific enthalpy for dry air in kJ/kg$_{da}$ and h_g is the specific enthalpy for saturated water vapor in kJ/kg$_w$ at the temperature of the mixture. As an approximation,

$$h_{da} \approx 1.006t \tag{30}$$

$$h_g \approx 2501 + 1.86t \tag{31}$$

where t is the dry-bulb temperature in °C. The moist air specific enthalpy in kJ/kg$_{da}$ then becomes

$$h = 1.006t + W(2501 + 1.86t) \tag{32}$$

THERMODYNAMIC WET-BULB AND DEW-POINT TEMPERATURE

For any state of moist air, a temperature t^* exists at which liquid (or solid) water evaporates into the air to bring it to saturation at exactly this same temperature and total pressure (Harrison 1965). During adiabatic saturation, saturated air is expelled at a temperature equal to that of the injected water. In this constant-pressure process,

- Humidity ratio increases from initial value W to W_s^*, corresponding to saturation at temperature t^*
- Enthalpy increases from initial value h to h_s^*, corresponding to saturation at temperature t^*
- Mass of water added per unit mass of dry air is $(W_s^* - W)$, which adds energy to the moist air of amount $(W_s^* - W)h_w^*$, where h_w^* denotes specific enthalpy in kJ/kg$_w$ of water added at temperature t^*

Therefore, if the process is strictly adiabatic, conservation of enthalpy at constant total pressure requires that

$$h + (W_s^* - W)h_w^* = h_s^* \tag{33}$$

W_s^*, h_w^*, and h_s^* are functions only of temperature t^* for a fixed value of pressure. The value of t^* that satisfies Equation (33) for given values of h, W, and p is the **thermodynamic wet-bulb temperature**.

A **psychrometer** consists of two thermometers; one thermometer's bulb is covered by a wick that has been thoroughly wetted with water. When the wet bulb is placed in an airstream, water evaporates from the wick, eventually reaching an equilibrium temperature called the **wet-bulb temperature**. This process is not one of adiabatic saturation, which defines the thermodynamic wet-bulb temperature, but one of simultaneous heat and mass transfer from the wet bulb. The fundamental mechanism of this process is described by the Lewis relation [Equation (38) in Chapter 5]. Fortunately, only small corrections must be applied to wet-bulb thermometer readings to obtain the thermodynamic wet-bulb temperature.

As defined, thermodynamic wet-bulb temperature is a unique property of a given moist air sample independent of measurement techniques.

Equation (33) is exact because it defines the thermodynamic wet-bulb temperature t^*. Substituting the approximate perfect gas relation [Equation (32)] for h, the corresponding expression for h_s^*, and the approximate relation for saturated liquid water

$$h_w^* \approx 4.186t^* \tag{34}$$

into Equation (33), and solving for the humidity ratio,

$$W = \frac{(2501 - 2.326t^*)W_s^* - 1.006(t - t^*)}{2501 + 1.86t - 4.186t^*} \tag{35}$$

where t and t^* are in °C. Below freezing, the corresponding equations are

$$h_i^* \approx -333.4 + 2.1t^* \tag{36}$$

$$W = \frac{(2830 - 0.24t^*)W_s^* - 1.006(t - t^*)}{2830 + 1.86t - 2.1t^*} \tag{37}$$

A wet/ice-bulb thermometer is imprecise when determining moisture content at 0°C.

The **dew-point temperature** t_d of moist air with humidity ratio W and pressure p was defined as the solution $t_d(p, w)$ of $W_s(p, t_d)$. For perfect gases, this reduces to

$$p_{ws}(t_d) = p_w = (pW)/(0.62198 + W) \tag{38}$$

where p_w is the water vapor partial pressure for the moist air sample and $p_{ws}(t_d)$ is the saturation vapor pressure at temperature t_d. The saturation vapor pressure is obtained from Table 3 or by using Equation (5) or (6). Alternatively, the dew-point temperature can be calculated directly by one of the following equations (Peppers 1988):

Between dew points of 0 and 93 °C,

$$t_d = C_{14} + C_{15}\alpha + C_{16}\alpha^2 + C_{17}\alpha^3 + C_{18}(p_w)^{0.1984} \tag{39}$$

Below 0°C,

$$t_d = 6.09 + 12.608\alpha + 0.4959\alpha^2 \tag{40}$$

where

t_d = dew-point temperature, °C
α = $\ln p_w$
p_w = water vapor partial pressure, kPa
C_{14} = 6.54
C_{15} = 14.526
C_{16} = 0.7389
C_{17} = 0.09486
C_{18} = 0.4569

NUMERICAL CALCULATION OF MOIST AIR PROPERTIES

The following are outlines, citing equations and tables already presented, for calculating moist air properties using perfect gas relations. These relations are accurate enough for most engineering calculations in air-conditioning practice, and are readily adapted to either hand or computer calculating methods. For more details, refer to Tables 15 through 18 in Chapter 1 of Olivieri (1996). Graphical procedures are discussed in the section on Psychrometric Charts.

SITUATION 1.

Given: Dry-bulb temperature t, Wet-bulb temperature t^*, Pressure p

To Obtain	Use	Comments
$p_{ws}(t^*)$	Table 3 or Equation (5) or (6)	Sat. press. for temp. t^*
W_s^*	Equation (23)	Using $p_{ws}(t^*)$
W	Equation (35) or (37)	
$p_{ws}(t)$	Table 3 or Equation (5) or (6)	Sat. press. for temp. t
W_s	Equation (23)	Using $p_{ws}(t)$
μ	Equation (12)	Using W_s
ϕ	Equation (25)	Using $p_{ws}(t)$
v	Equation (28)	
h	Equation (32)	
p_w	Equation (38)	
t_d	Table 3 with Equation (38), (39), or (40)	

SITUATION 2.

Given: Dry-bulb temperature t, Dew-point temperature t_d, Pressure p

To Obtain	Use	Comments
$p_w = p_{ws}(t_d)$	Table 3 or Equation (5) or (6)	Sat. press. for temp. t_d
W	Equation (22)	
$p_{ws}(t)$	Table 3 or Equation (5) or (6)	Sat. press. for temp. t_d
W_s	Equation (23)	Using $p_{ws}(t)$
μ	Equation (12)	Using W_s
ϕ	Equation (25)	Using $p_{ws}(t)$
ν	Equation (28)	
h	Equation (32)	
t^*	Equation (23) and (35) or (37) with Table 3 or with Equation (5) or (6)	Requires trial-and-error or numerical solution method

SITUATION 3.

Given: Dry-bulb temperature t, Relative humidity ϕ, Pressure p

To Obtain	Use	Comments
$p_{ws}(t)$	Table 3 or Equation (5) or (6)	Sat. press. for temp. t
p_w	Equation (24)	
W	Equation (22)	
W_s	Equation (23)	Using $p_{ws}(t)$
μ	Equation (12)	Using W_s
ν	Equation (28)	
h	Equation (32)	
t_d	Table 3 with Equation (38), (39), or (40)	
t^*	Equation (23) and (35) or (37) with Table 3 or with Equation (5) or (6)	Requires trial-and-error or numerical solution method

Moist Air Property Tables for Standard Pressure

Table 2 shows thermodynamic properties for standard atmospheric pressure at temperatures from −60 to 90°C. Properties of intermediate moist air states can be calculated using the degree of saturation μ:

$$\text{Volume} \qquad v = v_{da} + \mu v_{as} \qquad (41)$$

$$\text{Enthalpy} \qquad h = h_{da} + \mu h_{as} \qquad (42)$$

These equations are accurate to about 70°C. At higher temperatures, errors can be significant. Hyland and Wexler (1983a) include charts that can be used to estimate errors for v and h for standard barometric pressure. Nelson and Sauer (2002) provide psychrometric tables and charts up to 320°C and 1.0 kg_w/kg_{da}.

PSYCHROMETRIC CHARTS

A psychrometric chart graphically represents the thermodynamic properties of moist air.

The choice of coordinates for a psychrometric chart is arbitrary. A chart with coordinates of enthalpy and humidity ratio provides convenient graphical solutions of many moist air problems with a minimum of thermodynamic approximations. ASHRAE developed seven such psychrometric charts. Chart No. 1 is shown as Figure 1; the others may be obtained through ASHRAE.

Charts 1, 2, and 3 are for sea-level pressure (101.325 kPa). Chart 5 is for 750 m altitude (92.66 kPa), Chart 6 is for 1500 m altitude (84.54 kPa), and Chart 7 is for 2250 m altitude (77.04 kPa). All charts use oblique-angle coordinates of enthalpy and humidity ratio, and are consistent with the data of Table 2 and the properties computation methods of Goff (1949) and Goff and Gratch (1945), as well as Hyland and Wexler (1983a). Palmatier (1963) describes the geometry of chart construction applying specifically to Charts 1 and 4.

The dry-bulb temperature ranges covered by the charts are

Charts 1, 5, 6, 7	Normal temperature	0 to 50°C
Chart 2	Low temperature	−40 to 10°C
Chart 3	High temperature	10 to 120°C
Chart 4	Very high temperature	100 to 200°C

Psychrometric properties or charts for other barometric pressures can be derived by interpolation. Sufficiently exact values for most purposes can be derived by methods described in the section on Perfect Gas Relationships for Dry and Moist Air. Constructing charts for altitude conditions has been discussed by Haines (1961), Karig (1946), and Rohsenow (1946).

Comparison of Charts 1 and 6 by overlay reveals the following:

- The dry-bulb lines coincide.
- Wet-bulb lines for a given temperature originate at the intersections of the corresponding dry-bulb line and the two saturation curves, and they have the same slope.
- Humidity ratio and enthalpy for a given dry- and wet-bulb temperature increase with altitude, but there is little change in relative humidity.
- Volume changes rapidly; for a given dry-bulb and humidity ratio, it is practically inversely proportional to barometric pressure.

The following table compares properties at sea level (Chart 1) and 1500 m (Chart 6):

Chart No.	db	wb	h	W	rh	v
1	40	30	99.5	23.0	49	0.920
6	40	30	114.1	28.6	50	1.111

Figure 1 shows humidity ratio lines (horizontal) for the range from 0 (dry air) to 30 grams moisture per kilogram dry air. Enthalpy lines are oblique lines across the chart precisely parallel to each other.

Dry-bulb temperature lines are straight, not precisely parallel to each other, and inclined slightly from the vertical position. Thermodynamic wet-bulb temperature lines are oblique and in a slightly different direction from enthalpy lines. They are straight but are not precisely parallel to each other.

Relative humidity lines are shown in intervals of 10%. The saturation curve is the line of 100% rh, whereas the horizontal line for $W = 0$ (dry air) is the line for 0% rh.

Specific volume lines are straight but are not precisely parallel to each other.

A narrow region above the saturation curve has been developed for fog conditions of moist air. This two-phase region represents a mechanical mixture of saturated moist air and liquid water, with the two components in thermal equilibrium. Isothermal lines in the fog region coincide with extensions of thermodynamic wet-bulb temperature lines. If required, the fog region can be further expanded by extending humidity ratio, enthalpy, and thermodynamic wet-bulb temperature lines.

The protractor to the left of the chart shows two scales: one for sensible/total heat ratio, and one for the ratio of enthalpy difference to humidity ratio difference. The protractor is used to establish the direction of a condition line on the psychrometric chart.

Example 1 illustrates use of the ASHRAE Psychrometric Chart to determine moist air properties.

Example 1. Moist air exists at 40°C dry-bulb temperature, 20°C thermodynamic wet-bulb temperature, and 101.325 kPa pressure. Determine the humidity ratio, enthalpy, dew-point temperature, relative humidity, and specific volume.
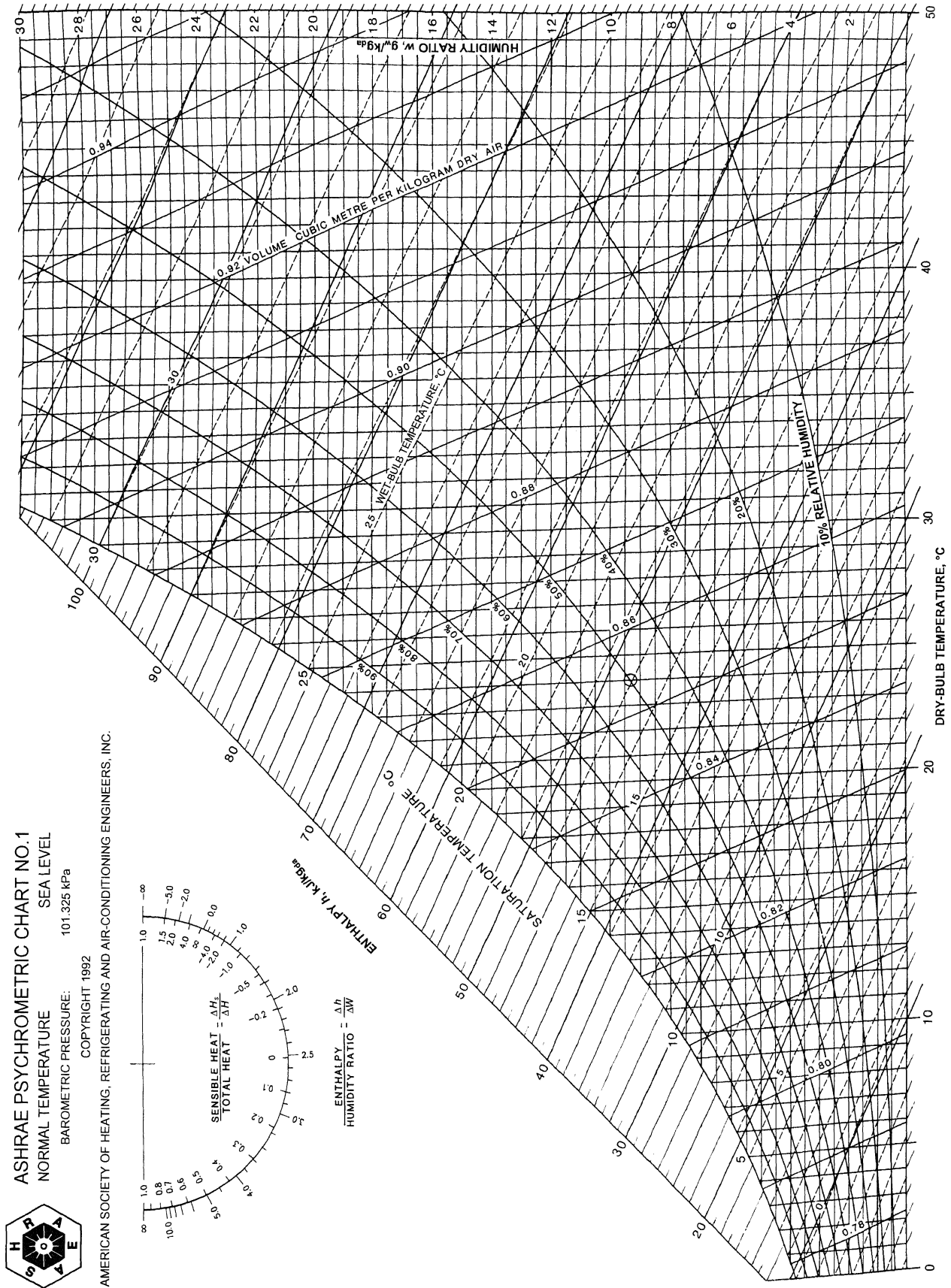
ASHRAE PSYCHROMETRIC CHART NO.1
NORMAL TEMPERATURE   SEA LEVEL
BAROMETRIC PRESSURE: 101.325 kPa
COPYRIGHT 1992
AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, INC.

Fig. 1   ASHRAE Psychrometric Chart No. 1

Solution: Locate state point on Chart 1 (Figure 1) at the intersection of 40°C dry-bulb temperature and 20°C thermodynamic wet-bulb temperature lines. Read **humidity ratio** $W = 6.5$ g_w/kg_{da}.

The **enthalpy** can be found by using two triangles to draw a line parallel to the nearest enthalpy line (60 kJ/kg$_{da}$) through the state point to the nearest edge scale. Read $h = 56.7$ kJ/kg$_{da}$.

Dew-point temperature can be read at the intersection of $W = 6.5$ g_w/kg_{da} with the saturation curve. Thus, $t_d = 7$°C.

Relative humidity ϕ can be estimated directly. Thus, $\phi = 14\%$.

Specific volume can be found by linear interpolation between the volume lines for 0.88 and 0.90 m³/kg$_{da}$. Thus, $v = 0.896$ m³/kg$_{da}$.

TYPICAL AIR-CONDITIONING PROCESSES

The ASHRAE psychrometric chart can be used to solve numerous process problems with moist air. Its use is best explained through illustrative examples. In each of the following examples, the process takes place at a constant total pressure of 101.325 kPa.

Moist Air Sensible Heating or Cooling

Adding heat alone to or removing heat alone from moist air is represented by a horizontal line on the ASHRAE chart, because the humidity ratio remains unchanged.

Figure 2 shows a device that adds heat to a stream of moist air. For steady-flow conditions, the required rate of heat addition is

$$_1q_2 = \dot{m}_{da}(h_2 - h_1) \qquad (43)$$

Example 2. Moist air, saturated at 2°C, enters a heating coil at a rate of 10 m³/s. Air leaves the coil at 40°C. Find the required rate of heat addition.

Solution: Figure 3 schematically shows the solution. State 1 is located on the saturation curve at 2°C. Thus, $h_1 = 13.0$ kJ/kg$_{da}$, $W_1 = 4.38$ g_w/kg_{da}, and $v_1 = 0.785$ m³/kg$_{da}$. State 2 is located at the intersection of $t = 40$°C
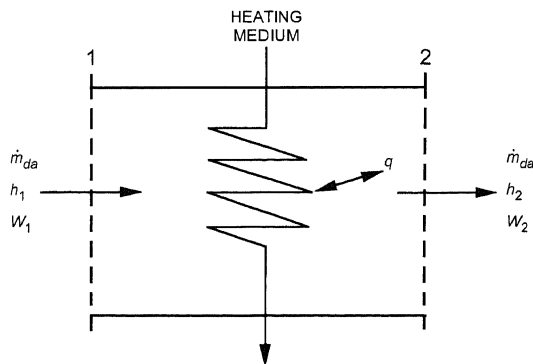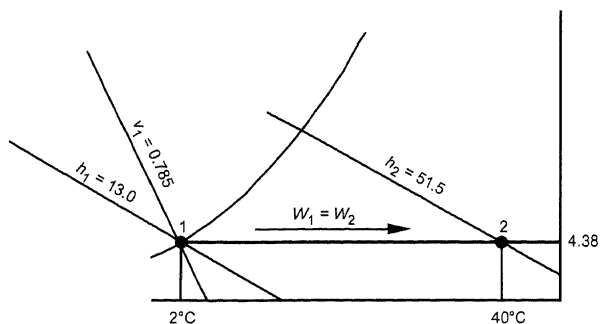

Fig. 2 Schematic of Device for Heating Moist Air

Fig. 3 Schematic Solution for Example 2

and $W_2 = W_1 = 4.38$ g_w/kg_{da}. Thus, $h_2 = 51.5$ kJ/kg$_{da}$. The mass flow of dry air is

$$\dot{m}_{da} = 10/0.785 = 12.74 \text{ kg}_{da}/s$$

From Equation (43),

$$_1q_2 = 12.74(51.5 - 13.0) = 490 \text{ kW}$$

Moist Air Cooling and Dehumidification

Moisture condensation occurs when moist air is cooled to a temperature below its initial dew point. Figure 4 shows a schematic cooling coil where moist air is assumed to be uniformly processed. Although water can be removed at various temperatures ranging from the initial dew point to the final saturation temperature, it is assumed that condensed water is cooled to the final air temperature t_2 before it drains from the system.

For the system in Figure 4, the steady-flow energy and material balance equations are

$$\dot{m}_{da}h_1 = \dot{m}_{da}h_2 + {}_1q_2 + \dot{m}_w h_{w2}$$
$$\dot{m}_{da}W_1 = \dot{m}_{da}W_2 + \dot{m}_w$$

Thus,

$$\dot{m}_w = \dot{m}_{da}(W_1 - W_2) \qquad (44)$$

$$_1q_2 = \dot{m}_{da}[(h_1 - h_2) - (W_1 - W_2)h_{w2}] \qquad (45)$$

Example 3. Moist air at 30°C dry-bulb temperature and 50% rh enters a cooling coil at 5 m³/s and is processed to a final saturation condition at 10°C. Find the kW of refrigeration required.

Solution: Figure 5 shows the schematic solution. State 1 is located at the intersection of $t = 30$°C and $\phi = 50\%$. Thus, $h_1 = 64.3$ kJ/kg$_{da}$, $W_1 = 13.3$ g_w/kg_{da}, and $v_1 = 0.877$ m³/kg$_{da}$. State 2 is located on the saturation curve at 10°C. Thus, $h_2 = 29.5$ kJ/kg$_{da}$ and $W_2 = 7.66$ g_w/kg_{da}. From Table 2, $h_{w2} = 42.11$ kJ/kg$_w$. The mass flow of dry air is

$$\dot{m}_{da} = 5/0.877 = 5.70 \text{ kg}_{da}/s$$

From Equation (45),

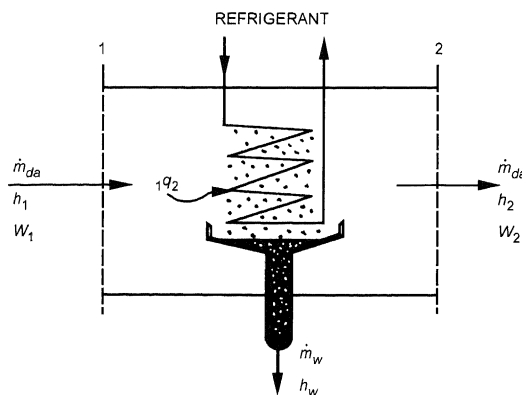$$_1q_2 = 5.70[(64.3 - 29.5) - (0.0133 - 0.00766)42.11]$$
$$= 197 \text{ kW}$$

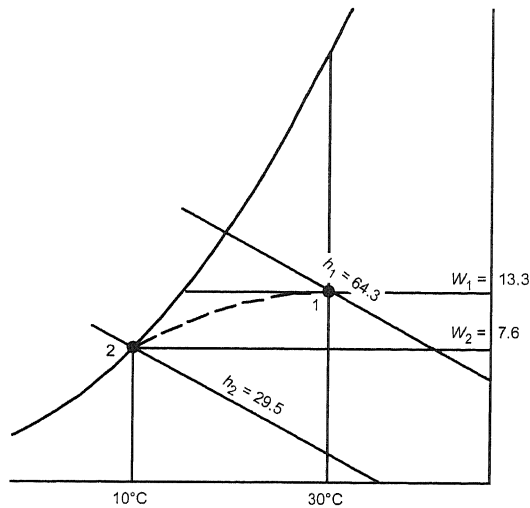**Fig. 4 Schematic of Device for Cooling Moist Air**

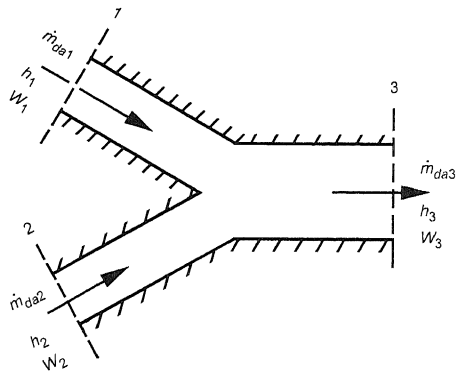**Fig. 5   Schematic Solution for Example 3**

Fig. 6   Adiabatic Mixing of Two Moist Airstreams

Adiabatic Mixing of Two Moist Airstreams

A common process in air-conditioning systems is the adiabatic mixing of two moist airstreams. Figure 6 schematically shows the problem. Adiabatic mixing is governed by three equations:

$$\dot{m}_{da1}h_1 + \dot{m}_{da2}h_2 = \dot{m}_{da3}h_3$$

$$\dot{m}_{da1} + \dot{m}_{da2} = \dot{m}_{da3}$$

$$\dot{m}_{da1}W_1 + \dot{m}_{da2}W_2 = \dot{m}_{da3}W_3$$

Eliminating $\dot{m}_{da3}$ gives

$$\frac{h_2 - h_3}{h_3 - h_1} = \frac{W_2 - W_3}{W_3 - W_1} = \frac{\dot{m}_{da1}}{\dot{m}_{da2}} \qquad (46)$$

according to which, on the ASHRAE chart, the state point of the resulting mixture lies on the straight line connecting the state points of the two streams being mixed, and divides the line into two segments, in the same ratio as the masses of dry air in the two streams.

Example 4. A stream of 2 m³/s of outdoor air at 4°C dry-bulb temperature and 2°C thermodynamic wet-bulb temperature is adiabatically mixed with 6.25 m³/s of recirculated air at 25°C dry-bulb temperature and 50% rh. Find the dry-bulb temperature and thermodynamic wet-bulb temperature of the resulting mixture.
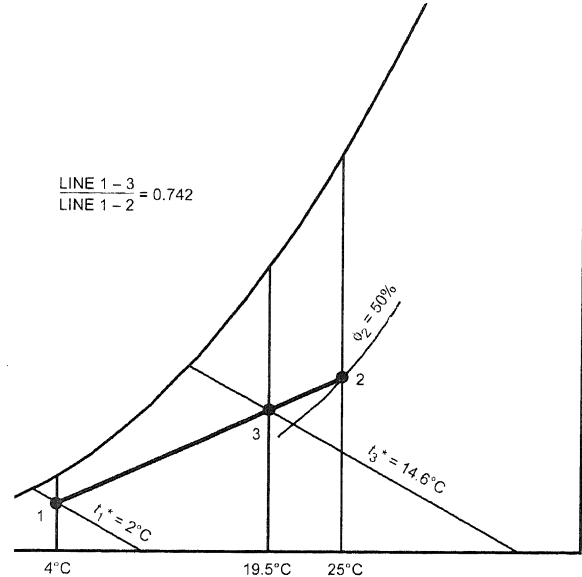
Fig. 7   Schematic Solution for Example 4

Solution: Figure 7 shows the schematic solution. States 1 and 2 are located on the ASHRAE chart: $v_1 = 0.789$ m³/kg$_{da}$, and $v_2 = 0.858$ m³/kg$_{da}$. Therefore,

$$\dot{m}_{da1} = 2/0.789 = 2.535 \text{ kg}_{da}/\text{s}$$

$$\dot{m}_{da2} = 6.25/0.858 = 7.284 \text{ kg}_{da}/\text{s}$$

According to Equation (46),

$$\frac{\text{Line 3-2}}{\text{Line 1-3}} = \frac{\dot{m}_{da1}}{\dot{m}_{da2}} \text{ or } \frac{\text{Line 1-3}}{\text{Line 1-2}} = \frac{\dot{m}_{da2}}{\dot{m}_{da3}} = \frac{7.284}{9.819} = 0.742$$

Consequently, the length of line segment 1–3 is 0.742 times the length of entire line 1–2. Using a ruler, State 3 is located, and the values $t_3 = 19.5°C$ and $t_3^* = 14.6°C$ found.

Adiabatic Mixing of Water Injected into Moist Air

Steam or liquid water can be injected into a moist airstream to raise its humidity, as shown in Figure 8. If mixing is adiabatic, the following equations apply:

$$\dot{m}_{da}h_1 + \dot{m}_w h_w = \dot{m}_{da}h_2$$

$$\dot{m}_{da}W_1 + \dot{m}_w = \dot{m}_{da}W_2$$

Therefore,

$$\frac{h_2 - h_1}{W_2 - W_1} = \frac{\Delta h}{\Delta W} = h_w \qquad (47)$$

according to which, on the ASHRAE chart, the final state point of the moist air lies on a straight line in the direction fixed by the specific enthalpy of the injected water, drawn through the initial state point of the moist air.

Example 5. Moist air at 20°C dry-bulb and 8°C thermodynamic wet-bulb temperature is to be processed to a final dew-point temperature of 13°C by adiabatic injection of saturated steam at 110°C. The rate of dry airflow is 2 kg$_{da}$/s. Find the final dry-bulb temperature of the moist air and the rate of steam flow.

Solution: Figure 9 shows the schematic solution. By Table 3, the enthalpy of the steam $h_g = 2691$ kJ/kg$_w$. Therefore, according to Equation

Fig. 8 Schematic Showing Injection of Water into Moist Air

Fig. 10 Schematic of Air Conditioned Space

Fig. 9 Schematic Solution for Example 5

(47), the condition line on the ASHRAE chart connecting States 1 and 2 must have a direction:

$$\Delta h / \Delta W = 2.691 \text{ kJ/g}_w$$

The condition line can be drawn with the $\Delta h/\Delta W$ protractor. First, establish the reference line on the protractor by connecting the origin with the value $\Delta h/\Delta W$ = 2.691 kJ/g$_w$. Draw a second line parallel to the reference line and through the initial state point of the moist air. This second line is the condition line. State 2 is established at the intersection of the condition line with the horizontal line extended from the saturation curve at 13°C (t_{d2} = 13°C). Thus, t_2 = 21°C.

Values of W_2 and W_1 can be read from the chart. The required steam flow is,

$$\dot{m}_w = \dot{m}_{da}(W_2 - W_1) = 2 \times 1000(0.0093 - 0.0018)$$
$$= 15.0 \text{ kg}_w/\text{s}$$

Space Heat Absorption and Moist Air Moisture Gains

Air conditioning required for a space is usually determined by (1) the quantity of moist air to be supplied, and (2) the supply air condition necessary to remove given amounts of energy and water from the space at the exhaust condition specified.

Figure 10 shows a space with incident rates of energy and moisture gains. The quantity q_s denotes the net sum of all rates of heat gain in the space, arising from transfers through boundaries and from sources within the space. This heat gain involves energy addition alone and does not include energy contributions from water (or

water vapor) addition. It is usually called the **sensible heat gain**. The quantity $\Sigma \dot{m}_w$ denotes the net sum of all rates of moisture gain on the space arising from transfers through boundaries and from sources within the space. Each kilogram of water vapor added to the space adds an amount of energy equal to its specific enthalpy.

Assuming steady-state conditions, governing equations are

$$\dot{m}_{da}h_1 + q_s + \sum(\dot{m}_w h_w) = \dot{m}_{da}h_2$$

$$\dot{m}_{da}W_1 + \sum \dot{m}_w = \dot{m}_{da}W_2$$

or

$$q_s + \sum(\dot{m}_w h_w) = \dot{m}_{da}(h_2 - h_1) \tag{48}$$

$$\sum \dot{m}_w = \dot{m}_{da}(W_2 - W_1) \tag{49}$$

The left side of Equation (48) represents the total rate of energy addition to the space from all sources. By Equations (48) and (49),

$$\frac{h_2 - h_1}{W_2 - W_1} = \frac{\Delta h}{\Delta W} = \frac{q_s + \sum(\dot{m}_w h_w)}{\sum \dot{m}_w} \tag{50}$$

according to which, on the ASHRAE chart and for a given state of withdrawn air, all possible states (conditions) for supply air must lie on a straight line drawn through the state point of withdrawn air, with its a direction specified by the numerical value of $[q_s + \Sigma(\dot{m}_w h_w)]/\Sigma\dot{m}_w$. This line is the condition line for the given problem.

Example 6. Moist air is withdrawn from a room at 25°C dry-bulb temperature and 19°C thermodynamic wet-bulb temperature. The sensible rate of heat gain for the space is 9 kW. A rate of moisture gain of 0.0015 kg$_w$/s occurs from the space occupants. This moisture is assumed as saturated water vapor at 30°C. Moist air is introduced into the room at a dry-bulb temperature of 15°C. Find the required thermodynamic wet-bulb temperature and volume flow rate of the supply air.
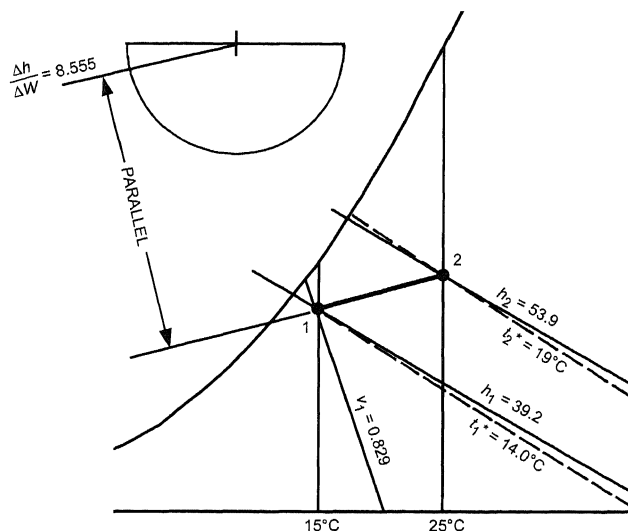
Solution: Figure 11 shows the schematic solution. State 2 is located on the ASHRAE chart. From Table 3, the specific enthalpy of the added water vapor is h_g = 2555.52 kJ/kg$_w$. From Equation (50),

$$\frac{\Delta h}{\Delta W} = \frac{9 + 0.0015 \times 2555.52}{0.0015} = 8555 \text{ kJ/kg}_w$$

With the $\Delta h/\Delta W$ protractor, establish a reference line of direction $\Delta h/\Delta W$ = 8.555 kJ/g$_w$. Parallel to this reference line, draw a straight line on the chart through State 2. The intersection of this line with the 15°C dry-bulb temperature line is State 1. Thus, t_1^* = 14.0°C.

Fig. 11 Schematic Solution for Example 6

Table 4 Calculated Diffusion Coefficients for Water/Air at 101.325 kPa

Temp., °C	mm²/s	Temp., °C	mm²/s	Temp., °C	mm²/s
−70	13.2	0	22.2	50	29.5
−50	15.6	5	22.9	55	30.3
−40	16.9	10	23.6	60	31.1
−35	17.5	15	24.3	70	32.7
−30	18.2	20	25.1	100	37.6
−25	18.8	25	25.8	130	42.8
−20	19.5	30	26.5	160	48.3
−15	20.2	35	27.3	190	54.0
−10	20.8	40	28.0	220	60.0
−5	21.5	45	28.8	250	66.3

An alternative (and approximately correct) procedure in establishing the condition line is to use the protractor's sensible/total heat ratio scale instead of the $\Delta h/\Delta W$ scale. The quantity $\Delta H_s/\Delta H_t$ is the ratio of rate of sensible heat gain for the space to rate of total energy gain for the space. Therefore,

$$\frac{\Delta H_s}{\Delta H_t} = \frac{q_s}{q_s + \Sigma(\dot{m}_w h_w)} = \frac{9}{9 + (0.0015 \times 2555.52)} = 0.701$$

Note that $\Delta H_s/\Delta H_t = 0.701$ on the protractor coincides closely with $\Delta h/\Delta W = 8.555$ kJ/g$_w$.

The flow of dry air can be calculated from either Equation (48) or (49). From Equation (48),

$$\dot{m}_{da} = \frac{q_s + \Sigma(\dot{m}_w h_w)}{h_2 - h_1} = \frac{9 + (0.0015 \times 2555.52)}{53.9 - 39.2}$$

$$= 0.873 \text{ kg}_{da}/\text{s}$$

At State 1, $v_1 = 0.829$ m³/kg$_{da}$.

Therefore, supply volume $= \dot{m}_{da} v_1 = 0.873 \times 0.829 = 0.724$ m³/s

TRANSPORT PROPERTIES OF MOIST AIR

For certain scientific and experimental work, particularly in the heat transfer field, many other moist air properties are important. Generally classified as transport properties, these include diffusion coefficient, viscosity, thermal conductivity, and thermal diffusion factor. Mason and Monchick (1965) derive these properties
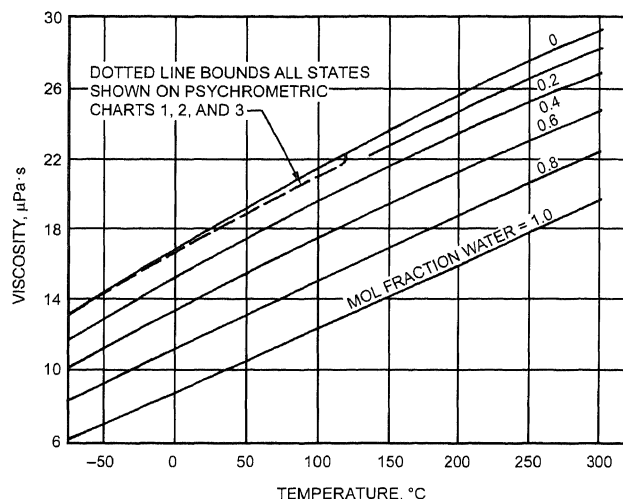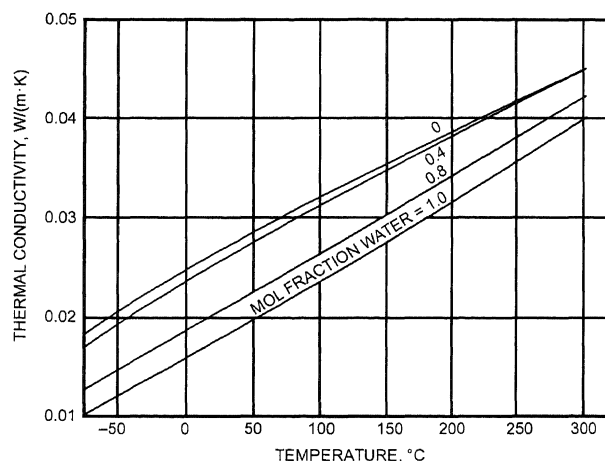

Fig. 12 Viscosity of Moist Air

Fig. 13 Viscosity of Moist Air

by calculation. Table 4 and Figures 12 and 13 summarize the authors' results on the first three properties listed. Note that, within the boundaries of ASHRAE Psychrometric Charts 1, 2, and 3, viscosity varies little from that of dry air at normal atmospheric pressure, and thermal conductivity is essentially independent of moisture content.

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Compressibility factor of moist air at pressures from 0 to 10 MPa, at values of degree of saturation from 0 to 100, and for temperatures from 0 to 60°C (Smithsonian Institution). [Note: At the time the Smithsonian Meteorological Tables were published, the value $\mu = W/W_s$ was known as relative humidity, in terms of a percentage. Since that time, there has been general agreement to designate the value μ as degree of saturation, usually expressed as a decimal and sometimes as a percentage. See Goff (1949) for more recent data and formulations.]

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Virial coefficients and other information for use with Goff and Gratch formulation (Goff 1949).

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Water properties. Includes properties of ordinary water substance for the gaseous, liquid, and solid phases (Dorsey 1940).

SYMBOLS

C_1 to C_{18} = constants in Equations (5), (6), and (39)

d_v = absolute humidity of moist air, mass of water per unit volume of mixture, kg_w/m^3

h = specific enthalpy of moist air, kJ/kg_{da}

H_s = rate of sensible heat gain for space, kW

h_s^* = specific enthalpy of saturated moist air at thermodynamic wet-bulb temperature, kJ/kg_{da}

H_t = rate of total energy gain for space, kW

h_w^* = specific enthalpy of condensed water (liquid or solid) at thermodynamic wet-bulb temperature and a pressure of 101.325 kPa, kJ/kg_w

M_{da} = mass of dry air in moist air sample, kg_{da}

$\dot{m}_{da}$ = mass flow of dry air, per unit time, kg_{da}/s

M_w = mass of water vapor in moist air sample, kg_w

$\dot{m}_w$ = mass flow of water (any phase), per unit time, kg_w/s

n = $n_{da} + n_w$, total number of moles in moist air sample

n_{da} = moles of dry air

n_w = moles of water vapor

p = total pressure of moist air, kPa

p_{da} = partial pressure of dry air, kPa

p_s = vapor pressure of water in moist air at saturation, kPa. Differs slightly from saturation pressure of pure water because of presence of air.

p_w = partial pressure of water vapor in moist air, kPa

p_{ws} = pressure of saturated pure water, kPa

q_s = rate of addition (or withdrawal) of sensible heat, kW

R = universal gas constant, 8314.41 J/(kg mole·K)

R_{da} = gas constant for dry air, $kJ/(kg_{da}·K)$

R_w = gas constant for water vapor, $kJ/(kg_w·K)$

s = specific entropy, $kJ/(kg_{da}·K)$ or $kJ/(kg_w·K)$

T = absolute temperature, K

t = dry-bulb temperature of moist air, °C

t_d = dew-point temperature of moist air, °C

t^* = thermodynamic wet-bulb temperature of moist air, °C

V = total volume of moist air sample, m^3

v = specific volume, m^3/kg_{da} or m^3/kg_w

v_T = total gas volume, m^3

W = humidity ratio of moist air, kg_w/kg_{da} or g_w/kg_{da}

W_s^* = humidity ratio of moist air at saturation at thermodynamic wet-bulb temperature, kg_w/kg_{da} or g_w/kg_{da}

x_{da} = mole fraction of dry air, moles of dry air per mole of mixture

x_w = mole fraction of water, moles of water per mole of mixture

x_{ws} = mole fraction of water vapor under saturated conditions, moles of vapor per mole of saturated mixture

Z = altitude, m

Greek

α = $\ln(p_w)$, parameter used in Equations (39) and (40)

γ = specific humidity of moist air, mass of water per unit mass of mixture

μ = degree of saturation W/W_s

ρ = moist air density

ϕ = relative humidity, dimensionless

Subscripts

as = difference between saturated moist air and dry air

da = dry air

f = saturated liquid water

fg = difference between saturated liquid water and saturated water vapor

g = saturated water vapor

i = saturated ice

ig = difference between saturated ice and saturated water vapor

s = saturated moist air

t = total

w = water in any phase

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