L_h	Horizontal scalar distance on h-W chart.
L_w	Vertical scalar distance on h-W chart.
m	Mass, lbm or kg.
m_a	Mass of dry air, lbm_a or kg_a.
m_w	Mass of water or water vapor, lbm_w or kg_w.
P	Pressure of moist air, psia, psfa, in. Hg or kPa.
P_a	Partial pressure of dry air, psia, psfa, in. Hg, or kPa.
P_w	Pressure of water vapor, psia, psfa, in. Hg, or kPa.
P_0	Standard atmospheric pressure or reference pressure, psia, psfa, in. Hg, or kPa.
q'	$(h_2 - h_1)/(W_2 - W_1)$, Btu/lbm_w or kJ/kg_w.
R	Gas constant, ft-lbf/lbm · °R, Btu/lbm · °R or kJ/kg · K; R_a for dry air and R_w for water vapor.
S	Psychrometric-chart scale factor.
S	Entropy of moist air, Btu/°R or kJ/K.
$S_{a,0}$	Entropy of dry air at reference conditions, Btu/°R or kJ/K.
$S_{w,0}$	Entropy of water vapor at reference conditions Btu/°R or kJ/K.
$\Delta S_{a,T,P}$	Entropy difference between reference state and state T, P for dry air, Btu/°R or kJ/K.
$\Delta S_{a,\text{mix}}$	Entropy of mixing for dry air, Btu/°R or kJ/K.
$\Delta S_{w,T,P}$	Entropy difference between reference state and state T, P for water vapor, Btu/°R or kJ/K.
$\Delta S_{w,\text{mix}}$	Entropy of mixing for water vapor, Btu/°R or kJ/K.
s	Specific entropy of moist air, Btu/lbm_a · °R or kJ/kg_a · K; s_a for dry air; s_s for saturated moist air at t; $s_{as} = s_s - s_a$.
s_h	Enthalpy scale factor for psychrometric chart.
s_w	Humidity-ratio scale factor for psychrometric chart.
$s_{w,0}$	Specific entropy of water vapor at reference conditions, Btu/lbm_w · °R or kJ/kg_w · K.
T	Absolute dry-bulb temperature, °R or K.
t	Dry-bulb temperature, °F or °C.
t_d	Dew-point temperature, °F or °C.
t^*	Thermodynamic wet-bulb temperature, °F or °C.
$\bar{t}$	Average dry-bulb temperature for a process, °F or °C.
v	Specific volume of moist air, ft³/lbm_a or m³/kg_a; v_s for saturated moist air.
v_a	Specific volume of dry air, ft³/lbm_a or m³/kg_a.
v_{as}	$v_{as} = v_s - v_a$ ft³/lbm_a or m³/kg_a.
W	Humidity ratio, lbm_w/lbm_a or kg_w/kg_a; W_s for air saturated at t; W_s^* for air saturated at t^*.
x_a	Mol-fraction of dry air, moles dry air per mole moist air.
x_w	Mol-fraction of water vapor, moles water vapor per mole moist air; $x_{w,s}$ for case of saturated air at t.
z	Altitude, ft or m.

Greek Letters

α	$180 - (\theta + \beta)$, deg (see Fig. 7.3).
β	Inclination angle of enthalpy lines (see Fig. 7.3).
θ	Inclination angle of any straight line on psychrometric chart (see Fig. 7.3).
μ	Degree of saturation, $\mu = W/W_s$, dimensionless.
ϕ	Relative humidity, dimensionless.

chapter 8

Psychrometric Processes and Applications

8.1 INTRODUCTION

Many of the problems in processing moist air with various apparatus result in rather complex process lines on a psychrometric chart. This chapter will address these problems by first considering a number of fundamental processes that, in various combinations, are used to construct heating, ventilating, air conditioning, and HVAC systems. The remaining sections of the chapter will describe a variety of these HVAC systems.

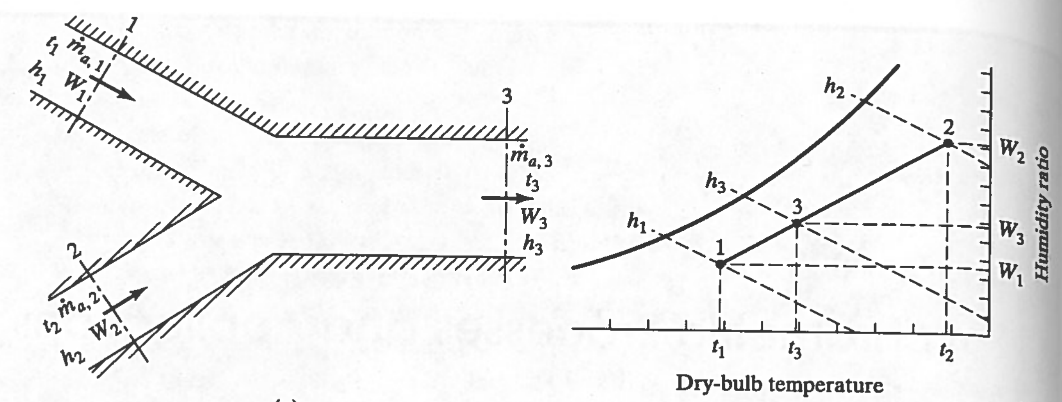
All of the processes considered here will be for steady-flow conditions. The total or barometric pressure will be assumed constant throughout the process. This assumption is valid in almost all psychrometric processing problems, since even in actual apparatus, such as finned coils or spray chambers, the pressure drop is typically less than 1 in. of water (less than 7 kPa).

8.2 ELEMENTARY PSYCHROMETRIC PROCESSES

The analysis of psychrometric processes is performed by considering a control volume and then applying the principles of the conservation of mass and energy to the control volume. All of the processes are steady-flow processes with negligible changes in potential and kinetic energy. In addition, generally no shaft work is involved. Thus the steady-state energy equation reduces to

$$\sum_{\text{in}} \dot{m}h + \dot{Q} = \sum_{\text{out}} \dot{m}h \tag{8.1}$$

The quantity $\dot{Q}$ is the rate of heat added to the control volume. The sign convention used is that $\dot{Q}$ will be taken as positive when heat is added to the control volume. In most cases the control volume selected will be the volume of moist air that is present in an apparatus.



(a) (b)

Figure 8.1 Schematic adiabatic mixing of two streams of moist air.

Adiabatic Mixing of Two Streams of Moist Air. In almost every air conditioning system the mixing of two or more air streams may occur. Usually, such mixing processes occur under essentially adiabatic conditions. Figure 8.1(a) schematically shows a mixture of two air streams. The fundamental equations applying to the process are

$$\dot{m}_{a,3} = \dot{m}_{a,1} + \dot{m}_{a,2} \tag{8.2}$$

$$W_3 = \frac{\dot{m}_{a,1}}{\dot{m}_{a,3}} W_1 + \frac{\dot{m}_{a,2}}{\dot{m}_{a,3}} W_2 \tag{8.3}$$

$$h_3 = \frac{\dot{m}_{a,1}}{\dot{m}_{a,3}} h_1 + \frac{\dot{m}_{a,2}}{\dot{m}_{a,3}} h_2 \tag{8.4}$$

By elimination of $\dot{m}_{a,3}$ we obtain

$$\frac{\dot{m}_{a,1}}{\dot{m}_{a,2}} = \frac{h_2 - h_3}{h_3 - h_1} = \frac{W_2 - W_3}{W_3 - W_1} \tag{8.5}$$

or

$$\frac{h_3 - h_2}{W_3 - W_2} = \frac{h_3 - h_1}{W_3 - W_1} \tag{8.6}$$

Equation (8.6) tells us that the process from state 2 to state 3 has the same slope, q', as the process from state 1 to state 3, and, since both processes have state 3 in common, the resulting state 3 must lie on a straight line connecting states 1 and 2 on the psychrometric chart. Furthermore, the lengths of the line segments are proportional to the mass flow rates of dry air mixed. In Fig. 8.1(b) we may write

$$\frac{\dot{m}_{a,1}}{\dot{m}_{a,2}} = \frac{\overline{32}}{\overline{13}} \quad \text{or} \quad \frac{\dot{m}_{a,1}}{\dot{m}_{a,3}} = \frac{\overline{32}}{\overline{12}} \quad \text{or} \quad \frac{\dot{m}_{a,2}}{\dot{m}_{a,3}} = \frac{\overline{13}}{\overline{12}} \tag{8.7}$$

In the United States, air conditioning engineers sometimes use an approximate equation involving the dry-bulb temperatures of the air streams to find the mixed-air condition. By Eqs. (8.4), (7.22), (8.2), and (8.3)

$$\dot{m}_{a,3}(c_{pa} + W_3 c_{pw})t_3 = \dot{m}_{a,1}(c_{pa} + W_1 c_{pw})t_1 + \dot{m}_{a,2}(c_{pa} + W_2 c_{pw})t_2 \tag{8.8}$$

or, in terms of the specific heats of the mixtures,

$$\dot{m}_{a,3} c_{p,3} t_3 = \dot{m}_{a,1} c_{p,1} t_1 + \dot{m}_{a,2} c_{p,2} t_2$$

If we make the approximation that the specific heats of the mixtures are approximately equal, the result is

$$t_3 \cong \frac{\dot{m}_{a,1}}{\dot{m}_{a,3}} t_1 + \frac{\dot{m}_{a,2}}{\dot{m}_{a,3}} t_2 \tag{8.9}$$

For most mixing processes, the accuracy resulting from the use of Eq. (8.9) is well within the accuracy with which one can read the psychrometric chart.

There are several choices in the solution of a mixing problem. We may solve a problem algebraically using the mixing equations, or we may use a graphical solution on the psychrometric chart. The following example illustrates one convenient method of solution.

EXAMPLE 8.1

One stream of moist air (1000 ft³/min, 60 °F dry-bulb temperature, 56 °F thermodynamic wet-bulb temperature) is mixed with a second stream (400 ft³/min, 80 °F dry-bulb temperature, 67 °F thermodynamic wet-bulb temperature). Barometric pressure is 14.696 psia. Determine the dry-bulb and wet-bulb temperatures of the resulting mixture.

Solution: We may locate states 1 and 2 on Fig. C-8E and determine that v_1 = 13.28 ft³/lbm$_a$ and v_2 = 13.85 ft³/lbm$_a$. Thus

$$\dot{m}_{a,1} = \frac{1000}{13.28} = 75.3 \text{ lbm}_a/\text{min}$$

$$\dot{m}_{a,2} = \frac{400}{13.85} = 28.9 \text{ lbm}_a/\text{min}$$

By Eq. (8.2)

$$\dot{m}_{a,3} = 75.3 + 28.9 = 104.2 \text{ lbm}_a/\text{min}$$

In Fig. C-8E, connect states 1 and 2 with a straight line. Then using either Eq. (8.3) or (8.4), it is possible to locate where state 3 lies on the line. For this example problem, use Eq. (8.3). From Fig. C-8E

$$W_1 = 0.00867 \text{ lbm}_w/\text{lbm}_a$$

$$W_2 = 0.01123 \text{ lbm}_w/\text{lbm}_a$$

Thus by Eq. (8.3)

$$W_3 = \frac{75.3}{104.2}(0.00867) + \frac{28.9}{104.2}(0.01123) = 0.00938 \text{ lbm}_w/\text{lbm}_a$$

In Fig. C-8E the intersection of W_3 with the line connecting states 1 and 2 is state 3. We find that t_3 = 65.4 °F and t_3^* = 59.3 °F.

Notice that the approximation for t given by Eq. (8.9) would have resulted in

$$t_3 = \frac{75.3}{104.2}(60) + \frac{28.9}{104.2}(80) = 65.5 \text{ °F}$$

Standard Air. Before we proceed to other fundamental processes, it is useful to describe what ASHRAE defines as *standard air* [1]. Since the specific volume of moist air varies appreciably, all calculations must be made on the basis of mass, not volume, flow rates of air. However, volumetric flow rates are often required for the selection of heating or cooling coils, fans, ducts, etc. In those cases volume values based on measurements at *standard conditions* may be used for accurate results. The value specified by ASHRAE for the specific volume of *standard air*, v_{std}, is

$$v_{std} = 13.33\ \text{ft}^3/\text{lbm}_a (0.830\ \text{m}^3/\text{kg}_a)$$

or the density of standard air, ρ_{std}, is

$$\rho_{std} = 0.075\ \text{lbm}_a/\text{ft}^3 (1.204\ \text{kg}_a/\text{m}^3)$$

For standard atmospheric pressure this density corresponds to that of saturated air at about 60 °F (15 °C) and dry air at about 69 °F (20 °C). Using the definition of standard air, we can express the mass flow rate of dry air through an apparatus as

$$\dot{m}_a = \rho_{std} \dot{V}_{std} \tag{8.10}$$

where $\dot{V}_{std}$ is the volumetric flow rate of standard air.

Because air usually passes through coils, fans, ducts, etc. at a density close to standard, the accuracy desired normally requires no correction. When airflow is to be specified at a particular condition or point, such as at a coil entrance or exit, the true specific volume of the air can be read from the psychrometric chart.

EXAMPLE 8.2

Calculate the volumetric flow rate of standard air of the moist air stream at state 1 of Ex. 8.1.

Solution: In Ex. 8.1 the entering volumetric flow rate at state 1 was 1000 ft³/min at 13.28 ft³/lbm$_a$, or $\dot{m}_a$ = 75.3 lbm$_a$/min. By Eq. (8.10)

$$\dot{V}_{std} = \frac{\dot{m}_a}{\rho_{std}} = \frac{75.3}{0.075} = 1004\ \text{ft}^3/\text{min of standard air}$$

Sensible Heating or Cooling of Moist Air. If heat is added to moist air with no addition of moisture, then we speak of the process as one of *sensible heating*. Such a process may occur if moist air is passed across a heated surface, such as a bundle of finned tubes, where a medium such as hot water or steam circulates inside the tubes.

Sensible cooling is the reverse of sensible heating and may occur if air is passed across a cool surface. To restrict the process to only sensible cooling, the surface temperature must be higher than the air dew-point temperature. Figure 8.2(a) shows a schematic device for heating air, and Fig. 8.2(b) shows the process on a schematic psychrometric chart. The humidity ratio remains constant.

The steady-flow energy and material-balance equations are

$$\dot{m}_{a,1} h_1 + {}_1\dot{Q}_2 = \dot{m}_{a,2} h_2$$
$$\dot{m}_{a,1} = \dot{m}_{a,2}$$
$$\dot{m}_{a,1} W_1 = \dot{m}_{a,2} W_2$$

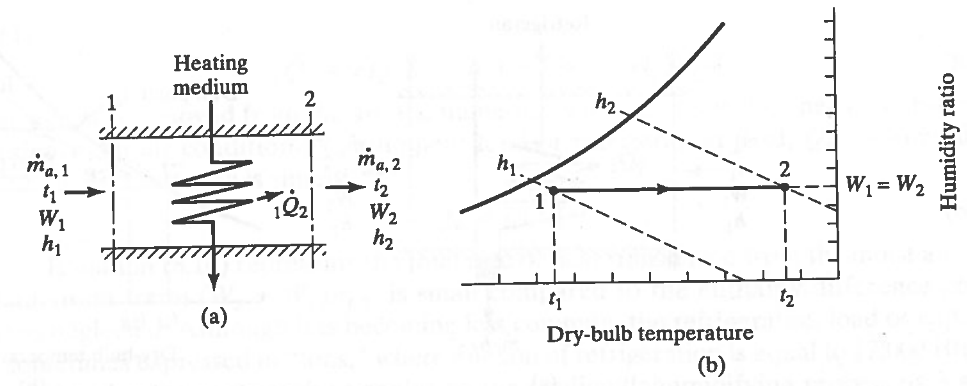


Figure 8.2 Schematic illustration of sensible heating of moist air.

Thus for sensible heating

$${}_1\dot{Q}_2 = \dot{m}_a (h_2 - h_1) \tag{8.11}$$

Since the humidity ratio remains constant, it is closely true by Eqs. (7.22) and (8.11) that

$${}_1\dot{Q}_2 = \dot{m}_a (c_{pa} + c_{pw} W)(t_2 - t_1) \tag{8.12}$$

In this process, as in others to follow, bulk or average properties are used at all of the inlets and outlets of the device being considered. Also, any duct walls which form the boundaries of the control volume are taken to be adiabatic.

EXAMPLE 8.3

Moist air enters a steam-heating coil at 40 °F dry-bulb temperature and 36 °F thermodynamic wet-bulb temperature at a rate of 2000 ft³/min. Barometric pressure is 14.696 psia. The air leaves the coil at a dry-bulb temperature of 140 °F. Determine the lbm/hr of saturated steam at 220 °F required, if the condensate leaves the coil at 200 °F.

Solution: By Fig. C-8E we find that W_1 = 0.00359 lbm$_w$/lbm$_a$, h_1 = 13.47 Btu/lbm$_a$, and v_1 = 12.66 ft³/lbm$_a$. By Fig. C-9E at t_2 = 140 °F and W_2 = 0.00359 lbm$_w$/lbm$_a$, we read h_2 = 37.70 Btu/lbm$_a$. Also,

$$\dot{m}_a = \frac{2000}{12.66}(60) = 9479\ \text{lbm}_a/\text{hr}$$

By Eq. (8.11),

$${}_1\dot{Q}_2 = (9479)(37.70 - 13.47) = 229{,}676\ \text{Btu/hr}$$

or, by Eq. (8.12),

$${}_1\dot{Q}_2 = (9479)[0.240 + (0.444)(0.00359)](140 - 40) = 229{,}000\ \text{Btu/hr}$$

For the steam, by Table A.1E, h_g (at 220 °F) = 1153.28 Btu/lbm$_w$. For the condensate, by Table A.1E, h_f (at 200 °F) = 168.34 Btu/lbm$_w$. Thus, the rate of steam flow is

$$\dot{m}_s = \frac{229{,}000}{1153.28 - 168.34} = 232.5\ \text{lbm}_w/\text{hr}$$

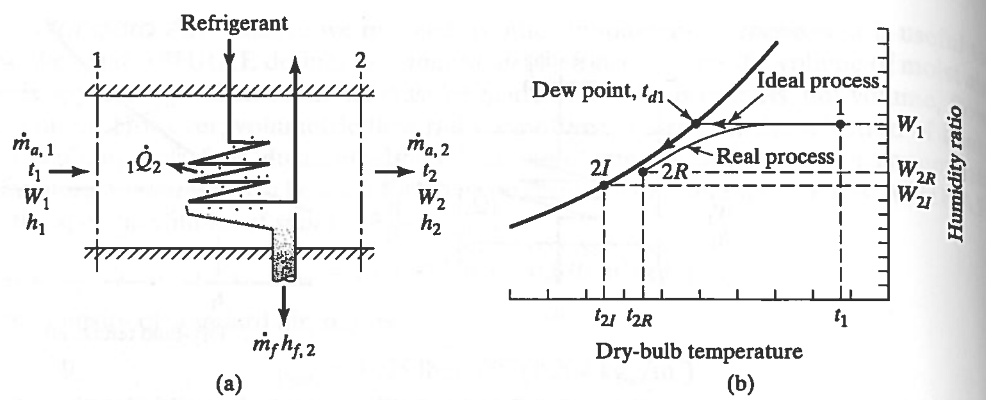


Figure 8.3 Schematic illustration of dehumidification by cooling.

Dehumidification of Moist Air by Cooling. If moist air is cooled below its dew point, condensation of moisture will occur. Figure 8.3(a) shows a schematic cooling device, and Fig. 8.3(b) shows schematically the psychrometric-chart solution when moist air is cooled below its initial dew-point temperature. Two cases are depicted in Fig. 8.3(b), an ideal process and a real process. The ideal case corresponds to one in which the air is uniformly and perfectly contacted by the cooling coil. In this case no condensation occurs until the average or bulk temperature of the air reaches the dew-point temperature. As the temperature is further reduced, the process follows the saturation line to the final state.

In a real process the air does not come into perfect or uniform contact with the heat-exchanger surfaces. A typical heat-exchanger design consists of a series of flat, parallel, cooled metal surfaces which form passages through which the moist air flows. (A detailed description of the heat exchanger is presented in Chapter 11.) The temperature of the air flowing through the passage is nonuniform. In the entrance region of the passage the air temperature near the surfaces will drop below the dew-point temperature, while that near the center line of the passage remains above. This results in dehumidification, even though the average air temperature in the passage is above the dew point. Since process lines on the psychrometric chart represent average or bulk conditions for the air, the real process appears as the curved line shown in Fig. 8.3(b). How closely a real process approaches the ideal and whether the final state is saturated depends on the design of the cooling coil.

The condensate can leave the apparatus at various temperatures ranging from the initial dew point to the final temperature, t_2. It is customary to assume that the condensate leaves at the final temperature, t_2. Thus the steady-flow energy equation is

$$ {}_1\dot{Q}_2 + \dot{m}_{a,1}h_1 = \dot{m}_{a,2}h_2 + \dot{m}_f h_{f,2} $$

The conservation of mass for the dry air and water results in

$$ \dot{m}_{a,1} = \dot{m}_{a,2} = \dot{m}_a $$

and

$$ \dot{m}_{a,1}W_1 = \dot{m}_{a,2}W_2 + \dot{m}_f $$

Thus

$$ \dot{m}_f = \dot{m}_a(W_1 - W_2) \tag{8.13} $$

and

$$ {}_1\dot{Q}_2 = \dot{m}_a[(h_2 - h_1) - (W_2 - W_1)h_{f,2}] \tag{8.14} $$

Since heat is removed from the air, the numerical value of ${}_1\dot{Q}_2$ will be negative. For convenience, an air conditioning equipment load or refrigeration load, $\dot{Q}_R$, is sometimes defined. This quantity is simply

$$ \dot{Q}_R = |{}_1\dot{Q}_2| \tag{8.15} $$

Equation (8.14) represents the total rate of heat transferred from the moist air. The condensate term, $(W_2 - W_1)h_{f,2}$, is small compared to the enthalpy difference and is often neglected. Although it is becoming less common, the refrigeration load or capacity is sometimes expressed in "tons," where one ton of refrigeration is equal to 12,000 Btu/hr.

An alternate approach to analyzing the cooling/dehumidifying process uses a so-called *bypass factor*. Consider a real process shown on the psychrometric chart in Fig. 8.4. In addition to the actual process line, Fig. 8.4 contains a straight dashed line which connects states 1 and 2 and which is extended to the saturation curve at state *d*. This point represents the apparatus dew-point temperature of the cooling coil. As discussed above, in a real cooling process such as the one shown in Fig. 8.4 all of the air passing between the surfaces of the heat exchanger or cooling coil is not cooled to the surface temperature. Thus the cooling coil performs as if a portion of the air were brought to saturation at the apparatus dew point or cooling-coil temperature, t_d, while the remaining air bypassed the cooling coil. The final state, (state 2), is then thought of as the adiabatic mixture of the bypassed air, which is at state 1, and the saturated air at state *d*. Figure 8.5 schematically depicts this representation of the process.

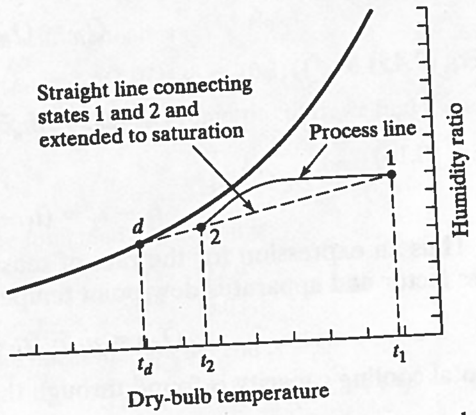


Figure 8.4 Schematic psychrometric chart of a real cooling/dehumidifying process.

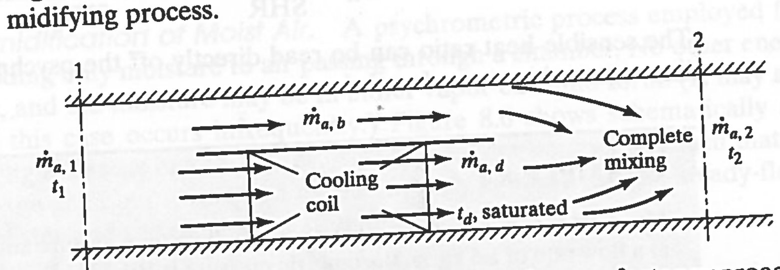


Figure 8.5 Schematic representation of the bypass-factor approach for a cooling/dehumidifying process.

The mass flow rate, $\dot{m}_{a,b}$, is thought of as bypassing the coil while $\dot{m}_{a,d}$ passes through the coil. The continuity equation for dry air gives

$$\dot{m}_{a,1} = \dot{m}_{a,2} = \dot{m}_a$$

$$\dot{m}_{a,b} + \dot{m}_{a,d} = \dot{m}_a$$

A *bypass factor*, b, is defined as

$$b = \frac{\dot{m}_{a,b}}{\dot{m}_a} \tag{8.16}$$

Applying Eq. (8.9) for the mixing process provides

$$t_2 = \frac{\dot{m}_{a,d}}{\dot{m}_a} t_d + \frac{\dot{m}_{a,b}}{\dot{m}_a} t_1$$

or

$$\frac{\dot{m}_{a,b}}{\dot{m}_a} = \frac{t_2 - t_d}{t_1 - t_d} \tag{8.17}$$

Thus from Eqs. (8.16) and (8.17)

$$b = \frac{t_2 - t_d}{t_1 - t_d} \tag{8.18}$$

This analysis approach also divides the total process into sensible and latent components $\dot{Q}_{RS}$ and $\dot{Q}_{RL}$, respectively.

$$\dot{Q}_R = \dot{Q}_{RS} + \dot{Q}_{RL} \tag{8.19}$$

By Eq. (7.45)

$$\dot{Q}_{RS} = \dot{m}_a \bar{c}_p (t_1 - t_2) \tag{8.20}$$

By Eq. (8.18)

$$t_1 - t_2 = (t_1 - t_d)(1 - b)$$

Thus an expression for the rate of sensible cooling can be written in terms of the bypass factor and apparatus dew-point temperature

$$\dot{Q}_{RS} = \dot{m}_a \bar{c}_p (t_1 - t_d)(1 - b) \tag{8.21}$$

The total cooling capacity is found through the use of the sensible-heat ratio:

$$\dot{Q}_R = \frac{\dot{Q}_{RS}}{\text{SHR}} = \frac{\dot{m}_a \bar{c}_p (t_1 - t_d)(1 - b)}{\text{SHR}} \tag{8.22}$$

The sensible-heat ratio can be read directly off the psychrometric chart for the straight line connecting states 1 and d.

EXAMPLE 8.4

Moist air enters a cooling coil at 28 °C dry-bulb temperature and 50 percent relative humidity at a flow rate of 1.5 kg_a/s. Barometric pressure is 101.325 kPa. The air leaves at 13 °C dry-bulb temperature and 90 percent relative humidity. Calculate the air conditioning load on the coil.

Solution: Using Fig. C-8SI and Table A.1SI, we find $W_1 = 0.0118\ kg_w/kg_a$, $h_1 = 58.2$ kJ/kg_a, $W_2 = 0.0084\ kg_w/kg_a$, $h_2 = 34.2$ kJ/kg_a, and $h_{f,2} = 53.61$ kJ/kg_w.

By Eq. (8.14)

$${}_1\dot{Q}_2 = 1.5[(34.2 - 58.2) - (0.0084 - 0.0118)(53.61)]\ \text{kJ/s} = -35.7\ \text{kJ/s}$$

or

$$\dot{Q}_R = 35.7\ \text{kW}$$

(Note that if we had neglected the condensate term, the answer would have been $\dot{Q}_R = 36.0$ kW.)

EXAMPLE 8.5

Work Ex. 8.4 using the bypass-factor approach. Find the sensible, latent, and total loads on the coil, the apparatus dew-point temperature, and the bypass factor.

Solution: Drawing a straight line through states 1 and 2 and extending it to saturation, we read the apparatus dew-point temperature as

$$t_d = 10.3\ °\text{C}$$

By Eq. (8.18), the bypass factor is

$$b = \frac{13 - 10.3}{28 - 10.3} = 0.153$$

By Eq. (8.21), the sensible load is

$$\dot{Q}_{RS} = 1.5(1.02)(28 - 10.3)(1 - 0.153) = 22.9$$

Using the protractor on the psychrometric chart, we find the sensible heat ratio for the process to be

$$\text{SHR} = 0.63$$

By Eq. (8.22), the total load on the coil is

$$\dot{Q}_R = \frac{22.9}{0.63} = 36.3\ \text{kW}$$

By Eq. (8.19), the latent load on the coil is

$$\dot{Q}_{RL} = 36.3 - 22.9 = 13.4\ \text{kW}$$

Humidification of Moist Air. A psychrometric process employed frequently is that of adding only moisture to air passing through a chamber. No other energy is added to the air, and the moisture may be in either vapor or liquid form. (It may also be solid, although this case occurs infrequently.) Figure 8.6 shows schematically a device for humidifying moist air in steady flow. This type of device operates such that all moisture added in the chamber is retained by the air passing through. The steady-flow mass balance for water and the energy equation are

$$\dot{m}_w = \dot{m}_a(W_2 - W_1)$$

$$\dot{m}_w h_w = \dot{m}_a(h_2 - h_1)$$

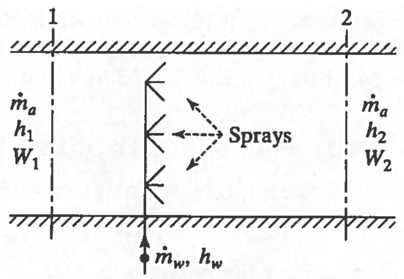


Figure 8.6 Schematic of humidification by the injection of water (liquid or vapor) into moist air.

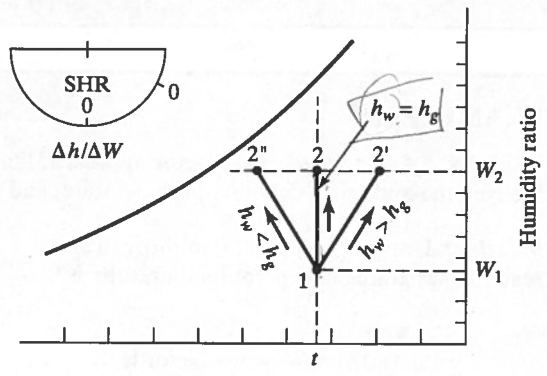


Figure 8.7 Schematic illustration of humidification processes.

Thus

$$q' = \frac{h_2 - h_1}{W_2 - W_1} = h_w \qquad (8.23)$$

The direction of the condition line connecting states 1 and 2 depends on the enthalpy of the moisture added. Two unique cases will be mentioned. Each has been considered previously.

Equation (7.53) states that when air is humidified at constant dry-bulb temperature, the steam added must have a specific enthalpy equal to that of saturated steam at the air dry-bulb temperature. If water at the air thermodynamic wet-bulb temperature is added, the entering and leaving air wet-bulb temperatures must be identical.

Figure 8.7 schematically shows several humidification condition lines. The constant dry-bulb temperature line divides the processes into two categories. If $h_w > h_g$, the air will be sensibly heated as well as humidified with a moisture spray. If $h_w < h_g$, the air will be cooled during the process of humidification.

EXAMPLE 8.6

Moist air enters a humidifier at 40 °C dry-bulb temperature and 20 °C wet-bulb temperature. Wet steam at 101.3 kPa and 75 percent quality (i.e., 25 percent liquid) is injected into the moist air flow. The moist air exits the humidifier with a relative humidity of 60 percent. Determine the dry-bulb temperature of the air as it leaves the humidifier.

Solution: The solution to this example can be carried out using the psychrometric chart. Equation (8.23) shows that the slope, $\Delta h/\Delta W$ or q', is equal to the enthalpy of the steam which is injected. Therefore, through the use of two straightedges draw a line through state 1 parallel to the line on the protractor with slope $q' = h_w$. The intersection of this line with the 60 percent relative-humidity curve locates state 2.

From Table A.1SI at 101.3 kPa, h_f = 419.51 kJ/kg$_w$ and h_g = 2675.60 kJ/kg$_w$.

At a quality of 75 percent

$$h_w = 419.51 + (0.75)(2675.60 - 419.51) = 2111.58 \text{ kJ/kg}_w$$

Therefore the slope on the chart is q' = 2.1 kJ/gram of water.

Figure 8.8 shows the construction on the psychrometric chart. After locating state 2, we read the dry-bulb temperature as t_2 = 34 °C.

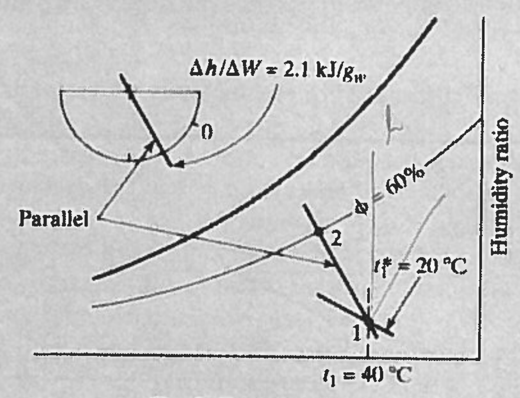


Figure 8.8 Psychrometric chart for Ex. 8.6.

EXAMPLE 8.7

Moist air is heated and humidified by passing it first over a heating coil and then adding moisture, as shown schematically in Fig. 8.9(a). The moist air enters the system at 40 °F dry-bulb and 36 °F thermodynamic wet-bulb temperature at a rate of 235 lbm$_a$/min. The humidifier injects saturated steam at 230 °F. The moist air exits the system at 90 °F dry-bulb temperature and 40 percent relative humidity. Locate state 2 on a psychrometric chart and determine the rate of heat addition by the heating coil and the rate of mass addition by the humidifier.

Solution: This example illustrates the operation of a typical heating/humidification unit, where the desired initial and final moist air states are known or specified and the problem is to size the components. State 2 can be found by the intersection of the sensible heating-process line (i.e., constant W) from state 1 and the humidifying-process line $q' = h_w$ passing through state 3. From Table A.1E at 230 °F, h_w = 1156.93 Btu/lbm$_w$. The psychrometric chart for this example is shown in Fig. 8.9(b). The following values are read from the chart:

h_1 = 13.4 Btu/lbm$_a$, W_1 = 0.0036 lbm$_w$/lbm$_a$

h_3 = 35.0 Btu/lbm$_a$, W_3 = 0.0122 lbm$_w$/lbm$_a$

$W_2 = W_1$ = 0.0036 lbm$_w$/lbm$_a$, t_2 = 87.6 °F, h_2 = 25.1 Btu/lbm$_a$

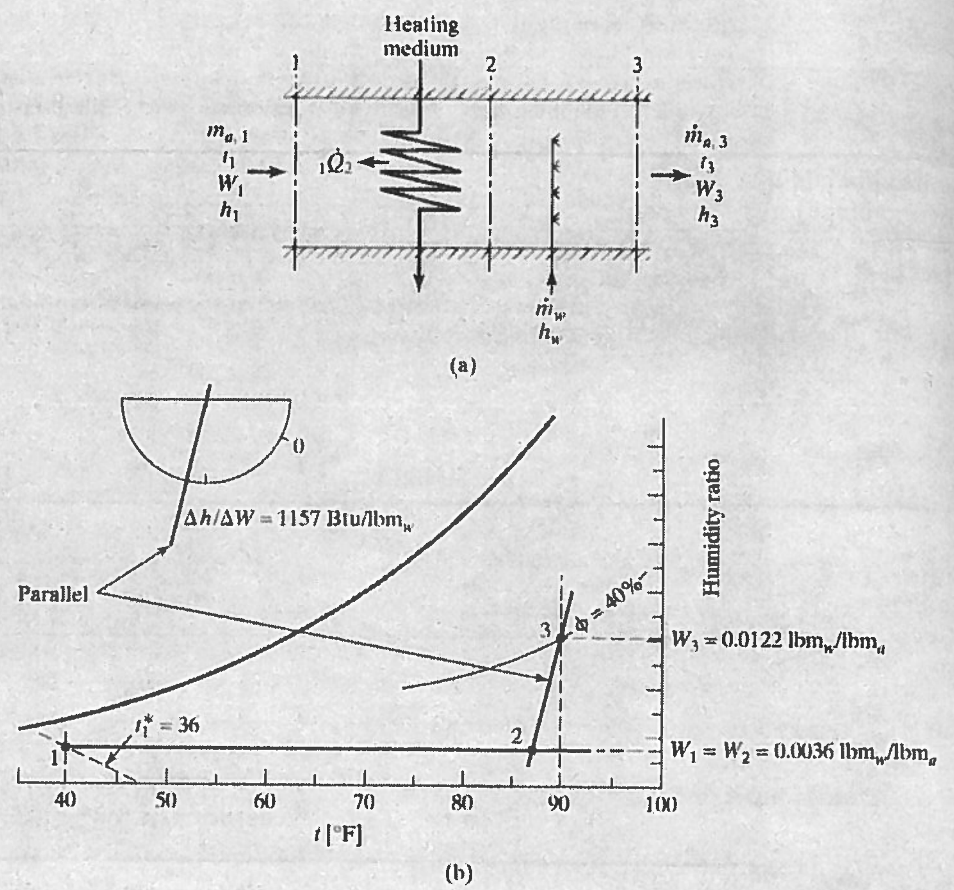


Figure 8.9 Schematic illustration of addition of heat and moisture to moist air.

The rate of heat addition is given by Eq. (8.11):

$$_1\dot{Q}_2 = 235(25.1 - 13.4) = 2750 \text{ Btu/hr}$$

The rate of water added is

$$\dot{m}_w = \dot{m}_a(W_3 - W_2) = 235(0.0122 - 0.0036) = 2.02 \text{ lbm}_w\text{/hr}$$

An alternate approach to solving the example is to solve for h_2 using Eq. (8.20) and observing from the sensible-heating process that $W_2 = W_1$.

Evaporative Cooling of Moist Air. In hot and dry climates evaporative cooling can be an effective means of reducing air temperature. Rather than pass the air through a cooling coil, we can take advantage of the low humidity to achieve cooling. This is accomplished by passing the air stream through a spray device using directly recirculated water. Such a device is called an *evaporative cooler* or *air washer*. Figure 8.10 shows a schematic diagram of such a device. Owing to the low relative humidity, part of the liquid-water stream evaporates. The energy for the evaporation process comes from the air

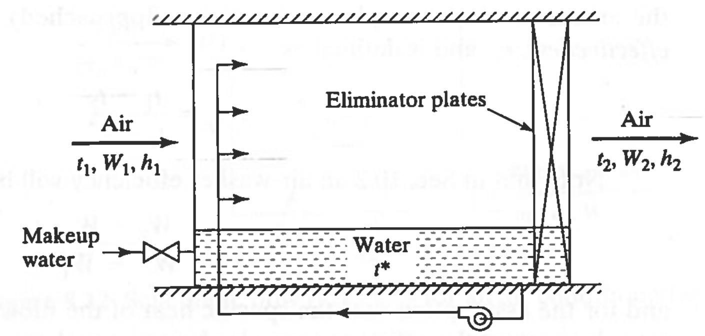


Figure 8.10 Schematic diagram of an evaporative cooler (air washer) using directly recirculated spray water.

stream. Thus, the overall effect is to cool and humidify the air. The equivalent effect may be carried out by passing the air through wetted media.

In this type of device liquid water is lifted from a sump to a distributing system from which it runs down through pads (wetted media), and the portion of the liquid not evaporated then returns to the sump to be recirculated.

A detailed analysis of the evaporative-cooling process is presented in Sec. 10.2. The analysis is carried out for the conditions that (a) the evaporation rate is much smaller than the water-recirculation rate, thus making the energy introduced by the makeup water negligible, (b) the walls of the device are adiabatic, and (c) the power of the recirculation pump is negligible. For our present purposes the key results of the analysis are:

1. The air passing through the evaporative cooler undergoes a constant wet-bulb temperature process.
2. All the liquid in the apparatus is at the wet-bulb temperature of the air stream.

Figure 8.11 shows the evaporative-cooling process on a psychrometric chart. The minimum temperature to which the air can be cooled is the wet-bulb temperature, t^*. The extent to which the leaving air temperature approaches this minimum temperature (or

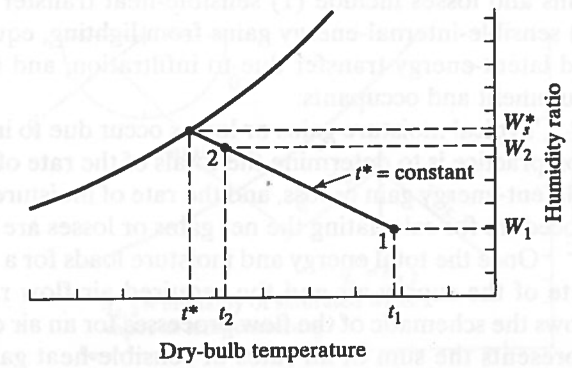


Figure 8.11 Psychrometric chart showing an evaporative-cooling process.

the extent to which complete saturation is approached) is referred to as the *saturation effectiveness*, e_c, and is defined as

$$e_c = \frac{t_1 - t_2}{t_1 - t^*} \tag{8.24}$$

Note that in Sec. 10.2 an air-washer efficiency will be defined as

$$\eta_w = \frac{W_2 - W_1}{W^* - W_1}$$

and for the assumption that the specific heat of the moist air passing through the apparatus is constant the efficiency can also be expressed as

$$\eta_w = \frac{t_1 - t_2}{t_1 - t^*}$$

However, for an evaporative cooler ASHRAE specifically points out the preferred use of the terminology *saturation effectiveness* and the definition given in Eq. (8.24) [2].

EXAMPLE 8.8

Moist air enters an evaporative cooler at 95 °F dry-bulb temperature and 10 percent relative humidity. The evaporative cooler has a saturation effectiveness of 85 percent. Determine the dry-bulb temperature and relative humidity of the air exiting the evaporative cooler.

Solution: Using Fig. C-8E at t_1 = 95 °F dry-bulb and ϕ_1 = 10 percent relative humidity, we find t^* = 60.6 °F. From Eq. (8.24)

$$t_2 = t_1 - e_c(t_1 - t^*) = 95 - (0.85)(95 - 60.6) = 65.8\ °\mathrm{F}$$

Thus state 2 is at t_2 = 65.8 °F and t^* = 60.6 °F, which, from Fig. C-8E, results in

$$\phi_2 = 75 \text{ percent}$$

Condition Line for a Space. A space which is to be conditioned can consist of a single room, a group of rooms, or an entire building. The space is subject to energy and moisture gains or losses (which are usually called the *loads* on the space). Typical energy gains and losses include (1) sensible-heat transfer through walls, roofs, windows, etc., (2) sensible-internal-energy gains from lighting, equipment, and occupants, (3) sensible- and latent-energy transfer due to infiltration, and (4) latent-internal-energy gains from equipment and occupants.

Typical moisture gains or losses occur due to infiltration and internal sources. Common practice is to determine the totals of the rate of sensible-energy gain or loss, the rate of latent-energy gain or loss, and the rate of moisture addition or removal for a space. The procedure for calculating the net gains or losses are presented in Chapters 14, 15, and 16.

Once the total energy and moisture loads for a space are known, the thermodynamic state of the supply air and the required air-flow rate may be determined. Figure 8.12 shows the schematic of the flow processes for an air conditioned space. The quantity $\Sigma\,\dot{Q}_S$ represents the sum of all rates of sensible-heat gain (i.e., sensible load). The quantity $\Sigma\,\dot{Q}_L = \Sigma\,\dot{m}_w h_w$ represents the sum of all rates of energy gain from added moisture (i.e., latent load). The quantity $\Sigma\,\dot{m}_w$ represents the net sum of all rates of moisture gain.

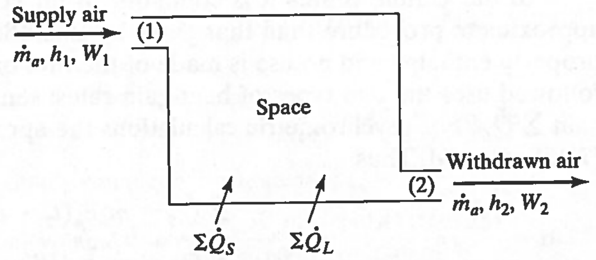


Figure 8.12 Schematic flow processes for an air conditioned space.

Assuming steady-flow conditions, we have

$$\Sigma\,\dot{Q}_S + \Sigma\,\dot{Q}_L = \dot{m}_a(h_2 - h_1) \tag{8.25}$$

$$\Sigma\,\dot{m}_w = \dot{m}_a(W_2 - W_1) \tag{8.26}$$

Thus the enthalpy-moisture ratio, q', is given by

$$q' = \frac{\Sigma\,\dot{Q}_S + \Sigma\,\dot{Q}_L}{\Sigma\,\dot{m}_w} = \frac{h_2 - h_1}{W_2 - W_1} \tag{8.27}$$

Equation (8.27) reveals that for a given state of the withdrawn air, all possible psychrometric states (conditions) for the supply air must lie on the same straight line, which has a direction q' and which is drawn through state 2 on the psychrometric chart. This straight line is called the *condition line*. Figure 8.13 shows three schematic condition lines. If $q' = h_{g,2}$, the condition line coincides with the dry-bulb temperature line t_2. Once q' is known, the condition line may be drawn with the aid of the chart protractor.

According to Eq. (8.27), state 1 may lie at any location on the condition line. However, changing of state 1 requires a change in the air-flow rate $\dot{m}_a$. Practical considerations influence the location of state 1. In summer-comfort air conditioning systems, $(t_2 - t_1)$ may vary from approximately 15 to 25 °F (10 to 15 °C), depending upon the method of air distribution.

The procedure given by Eqs. (8.25)–(8.27) is consistent with basic thermodynamics. It is a realistic procedure, since all quantities involved have definite physical meaning. Further, the procedure is precisely compatible with the *h-W* type of psychrometric chart.

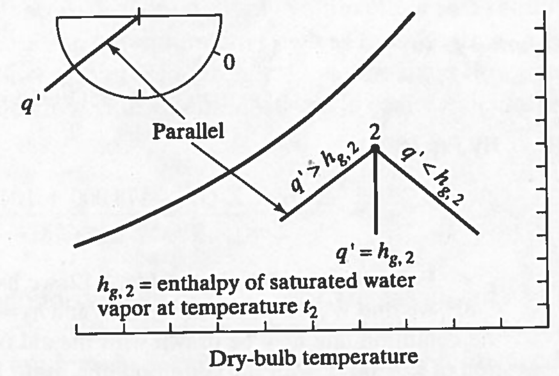


Figure 8.13 The condition line.

In the United States it is common for air conditioning engineers to use a more approximate procedure than that given by Eqs. (8.25)–(8.27). Little use is made of the property enthalpy and no use is made of the rate of moisture gain $\sum \dot{m}_w$. The approach followed uses the two types of heat-gain rates: sensible-heat gain $\sum \dot{Q}_S$ and latent-heat gain $\sum \dot{Q}_L$. For psychrometric calculations the approximations given by Eqs. (7.45) and (7.46) are used. Thus

$$\sum \dot{Q}_S = \dot{m}_a \bar{c}_p (t_2 - t_1) \tag{8.28}$$

$$\sum \dot{Q}_L = \dot{m}_a \bar{h}_g (W_2 - W_1) \tag{8.29}$$

As previously discussed, constant values are generally used for $\bar{c}_p$ and $\bar{h}_g$. The specific heat is commonly taken as 0.245 Btu/lbm$_a$ · °F (1.02 kJ/kg$_a$ · °C), and, depending upon the reference, values selected for $\bar{h}_g$ vary from about 1050 to 1150 Btu/lbm$_w$ (2500 to 2700 kJ/kg$_w$). In Sec. 7.7 it was shown that, in constructing the protractors included in the normal-temperature, sea-level psychrometric charts (Figs. C-8E and C-8SI), values of approximately $\bar{h}_g$ = 1100 Btu/lbm$_w$ and $\bar{h}_g$ = 2500 kJ/kg$_w$ were used. Therefore, it will be convenient to use these values for our calculations.

When using Eqs. (8.28) and (8.29) it is also customary to use the sensible-heat ratio instead of the enthalpy-moisture ratio. By definition

$$\text{SHR} = \frac{\sum \dot{Q}_S}{\sum \dot{Q}_S + \sum \dot{Q}_L} \tag{8.30}$$

Thus, using this approach the space-condition line is the straight line which has a direction SHR and which is drawn through state 2 on the psychrometric chart.

EXAMPLE 8.9

The air in a restaurant is to be maintained at 75 °F dry-bulb temperature and 50 percent relative humidity. The load calculations for the restaurant estimate the sensible rate of heat gain to be $\sum \dot{Q}_S$ = 178,000 Btu/hr. The rate of moisture gain is 95 lbm$_w$/hr with an average enthalpy of the moisture of h_w = 1095 Btu/lbm$_w$. Determine (a) the required dew-point temperature of the supply air, and (b) the required volume flow rate of supply air in ft³/min. The supply air is to be at a dry-bulb temperature of 60 °F. Assume standard atmospheric pressure. The solution to the example will be given first using the enthalpy-humidity ratio and then using the procedure given by Eqs. (8.28) and (8.29).

Solution 1: **(a)**

$$\sum \dot{Q}_L = \sum \dot{m}_w h_w = 95(1095) = 104{,}000 \text{ Btu/hr}$$

By Eq. (8.27)

$$q' = \frac{\sum \dot{Q}_S + \sum \dot{Q}_L}{\sum \dot{m}_w} = \frac{178{,}000 + 104{,}000}{95} = 2968 \text{ Btu/lbm}_w$$

Following the notation of Fig. 8.12, we have t_2 = 75 °F, ϕ_2 = 50 percent. By Fig. C-8E, we find W_2 = 0.00928 lbm$_w$/lbm$_a$ and h_2 = 28.15 Btu/lbm$_a$. As shown by Fig. 8.13, the condition line may be drawn with the aid of the chart protractor. At the intersection of t_1 = 60 °F with the condition line, state 1 is established. We read $t_{d,1}$ = 49 °F.

(b) At state 1, h_1 = 22.45 Btu/lbm$_a$, W_1 = 0.00740 lbm$_w$/lbm$_a$, and v_1 = 13.25 ft³/lbm$_a$. By Eq. (8.25)

$$\dot{m}_a = \frac{282{,}000}{28.15 - 22.45} = 49{,}474 \text{ lbm}_a/\text{hr}$$

Thus the supply volumetric flow rate is

$$\text{supply volume flow rate} = \frac{\dot{m}_a v_1}{60} = \frac{(49{,}474)(13.25)}{60} = 10{,}930 \text{ ft}^3/\text{min}$$

Solution 2: **(a)**

$$\sum \dot{Q}_L = \sum \dot{m}_w h_w = 95(1095) = 104{,}000 \text{ Btu/hr}$$

Thus the total load, $\sum \dot{Q}_T$, is

$$\sum \dot{Q}_T = \sum \dot{Q}_S + \sum \dot{Q}_L = 282{,}000 \text{ Btu/hr}$$

By Eq. (8.30)

$$\text{SHR} = \frac{178{,}000}{282{,}000} = 0.631$$

Similar to the procedure of Solution 1, but utilizing the SHR instead of q', the space-condition line may be drawn on Fig. C-8E with the aid of the chart protractor. At the intersection of t_1 = 60 °F with the condition line, state 1 is established. We read $t_{d,1}$ = 49 °F.

(b) At state 1, v_1 = 13.25 ft³/lbm$_a$. By Eq. (8.28) and with $\bar{c}_p$ = 0.245 Btu/lbm$_a$ · °F, we have

$$\dot{m}_a = \frac{\sum \dot{Q}_S}{\bar{c}_p (t_2 - t_1)} = \frac{178{,}000}{0.245(15)} = 48{,}435 \text{ lbm}_a/\text{hr}$$

Thus

$$\text{supply volume flow rate} = \frac{\dot{m}_a v_1}{60} = \frac{48{,}435(13.25)}{60} = 10{,}700 \text{ ft}^3/\text{min}$$

We may observe that the answers obtained in Ex. 8.9 using the two solution approaches are nearly equal. In later work in this chapter, the procedure given by Eqs. (8.28) and (8.29) will be emphasized.

Example 8.9 shows the actual supply volume flow rate required. As previously noted, commercial air conditioning equipment (such as fans, heating coils, etc.) is usually rated in terms of volume flow rate of standard air. Once the mass flow rate of dry air, $\dot{m}_a$, is determined, the volume flow rate of standard air is easily calculated using Eq. (8.10). (See Ex. 8.2.)

8.3 PSYCHROMETRIC SYSTEMS—SINGLE ZONE

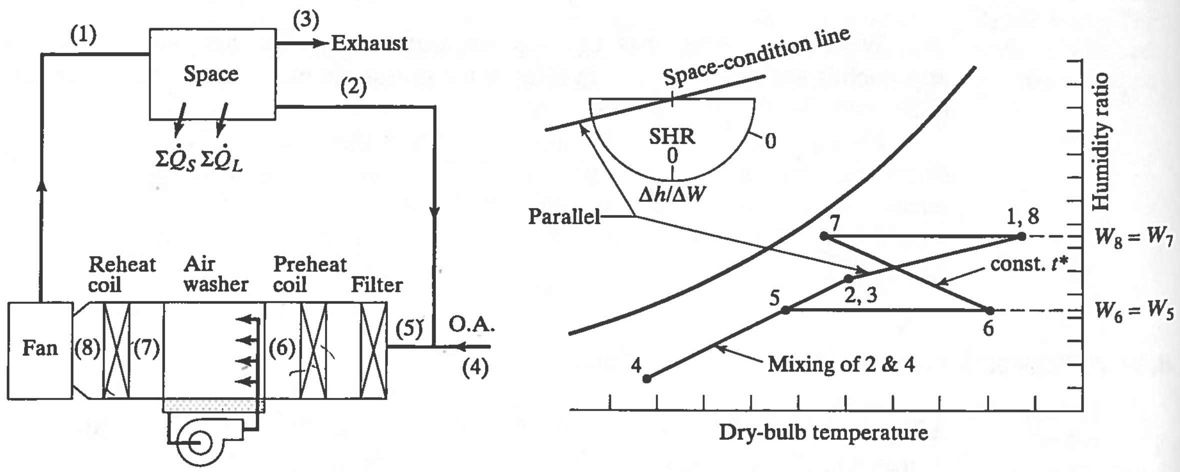
After the heat and moisture loads of a space are calculated and the space-condition line is drawn on the psychrometric chart, a psychrometric system may be chosen to process the moist air. The equipment requirements are greatly influenced by the space conditions desired and by the magnitude of the enthalpy-moisture ratio q'. The system should be capable of maintaining acceptable space conditions at all times. Since the thermal loads of the space may be highly variable, automatic control of the processing equipment becomes an important consideration. In most large buildings the loads will vary throughout the different locations in the building. In order to maintain comfort in the entire building, it will be subdivided into zones. The term *zone* is used to identify an area of the building which has its own thermostatic control. Thus, a zone could be as small as one room or as large as the entire building. In this section we will discuss several types of psychrometric systems for a single zone or space. Our purpose will be to illustrate general principles rather than to attempt to cover all combinations. In the final section of this chapter, examples of multizone systems will be presented.

Figure 8.14(a) shows a schematic system which may produce controlled temperature and humidity conditions within a space which has both heat and moisture losses; Fig. 8.14(b) schematically shows the psychrometric processes. We will first make a few general comments about Fig. 8.14, which will apply also to all other systems discussed in this section.

Energy added to air by a fan causes some rise in temperature without change in humidity ratio. This energy input may be calculated only after the ductwork is designed and processing devices are selected. Usually the energy input is small. In our discussions we will assume that an allowance for energy input by the fan has been included in the space load calculation. Thus, in Fig. 8.14, states 1 and 8 are assumed to be the same.

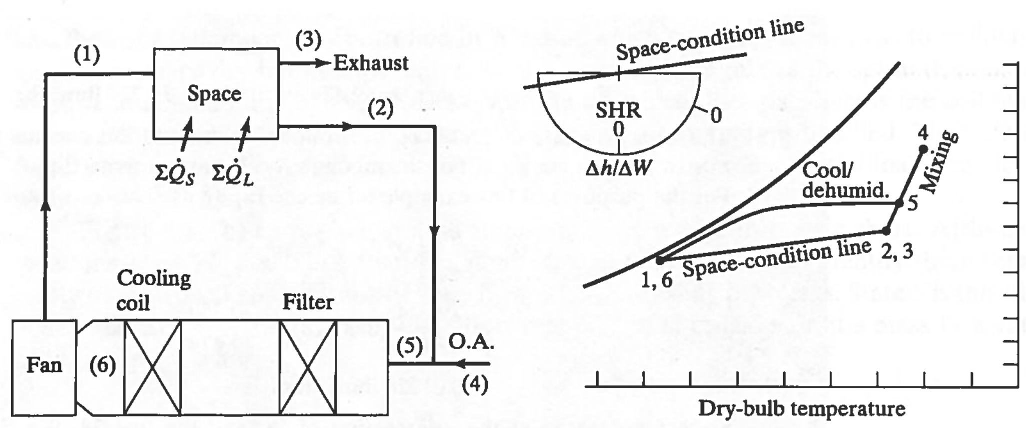
For each of our systems we will schematically include filters. Although filters (pads of viscous coated fibers, electrostatic type, etc.) do not change the moist air state, they are required for control of particulate matter carried by the air.

In each system we will assume that some outdoor air is positively introduced. Such air may be required for ventilation purposes or as makeup air for exhaust fans. Introduction of outdoor air requires that an equal mass for dry air be removed from the system. We will show this removal as occurring directly from the space [location 3 in Fig. 8.14(a)]. Figure 8.14(a) shows only one fan. In large systems it is common practice to use a recirculating-air fan as well as a supply-air fan. However, this circumstance does not affect the discussions to be given here.



(a) (b)

Figure 8.14 Schematic winter air conditioning system.



(a) (b)

Figure 8.15 Schematic elementary summer air conditioning system.

Processing equipment in Fig. 8.14 includes a preheat coil, air washer (or evaporative cooler), and reheat coil. The heating coils would be of the finned-tube type or electric coils. The preheat coil serves two functions: it prevents water in the air washer from possible freezing, and, by regulation of its heat supply, the amount of water evaporated in the air washer may be controlled. The air washer serves as a humidifying device to offset the moisture losses of the space, and it also accomplishes a certain amount of air cleaning. Regulation of the heat supply by the reheat coil allows control of the space dry-bulb temperature.

The schematic process lines of Fig. 8.14(b) should be obvious. The line $\overline{12}$ is the space-condition line, while line $\overline{452}$ is the mixing line for the mixing of recirculated and outdoor air. The process lines for both heating coils are ones of constant humidity ratio.

Figure 8.15 shows the most elementary form of a summer-comfort air conditioning system; a cooling coil is the only processing device. This coil would be of the finned-tube type with the cooling medium being either chilled water or an evaporating refrigerant. Analysis of cooling coils is given in Chapter 11.

EXAMPLE 8.10

Assume the restaurant of Ex. 8.9 is to be provided with the simple psychrometric system of Fig. 8.15. Outside-air conditions are 92 °F dry-bulb temperature and 77 °F wet-bulb temperature. The rate of exhaust from the restaurant is 4500 standard ft³/min. As a continuation of solution 1 of Ex. 8.9, determine (a) mass flow rate of recirculated air, lbm_a/hr, (b) the thermodynamic state of the moist air entering the cooling coil, and (c) the refrigeration capacity required.

Solution: The nomenclature of Fig. 8.15 will be used. From Ex. 8.9 we have the locations of states 1 and 2 on the psychrometric chart. We also have $\dot{m}_{a,1} = 49{,}474$ lbm_a/hr.

(a) The rate of exhaust air is given as 4500 standard ft³/min. From Eq. (8.7)

$$\dot{m}_{a,3} = \rho_{std}\dot{V}_{std} = 0.075(4500) = 337.5\ \text{lbm}_a/\text{min}$$

or

$$\dot{m}_{a,3} = 20{,}250\ \text{lbm}_a/\text{hr}$$

Thus,

$$\dot{m}_{a,2} = \dot{m}_{a,1} - \dot{m}_{a,3} = 49{,}474 - 20{,}250 = 29{,}224 \text{ lbm}_a/\text{hr}$$

(b) The mass flow rate of dry air $\dot{m}_{a,4}$ introduced from outdoors is equal to the mass flow rate of exhaust air, $\dot{m}_{a,3}$. For the mixing process we can write Eq. (8.3), (8.4), or (8.6). For the purposes of this example let us use Eq. (8.3). Thus,

$$W_5 = \frac{\dot{m}_{a,2}}{\dot{m}_{a,5}} W_2 + \frac{\dot{m}_{a,4}}{\dot{m}_{a,5}} W_4$$

$$= \frac{29{,}224}{49{,}474}(0.00928) + \frac{20{,}250}{49{,}474}(0.0166)$$

$$= 0.01228 \text{ lbm}_w/\text{lbm}_a$$

State 5 is located at the intersection of W_5 and the line $\overline{24}$. We find $t_5 = 82.2$ °F, $t_5^* = 69.2$ °F, and $h_5 = 33.25$ Btu/lbm$_a$.

(c) By Eqs. (8.14) and (8.15)

$$\dot{Q}_R = |{}_5\dot{Q}_6| = \dot{m}_{a,5}[(h_5 - h_6) - (W_5 - W_6)h_{f,6}]$$

$$\dot{Q}_R = (49{,}474)[(33.25 - 22.45) - (0.01228 - 0.00740)(27.63)]$$

$$\dot{Q}_R = 528{,}000 \text{ Btu/hr}$$

As previously indicated, the capacity is occasionally expressed in units of "tons." Thus, in that terminology,

$$\dot{Q}_R = \frac{528{,}000 \text{ Btu/hr}}{12{,}000 \text{ Btu/hr} \cdot \text{ton}} = 44 \text{ tons}$$

The simple system of Fig. 8.15 has limitations. Since the cooling coil is the only processing device, only one property of the moist air can be controlled. In comfort air conditioning systems, this would always be the space dry-bulb temperature.

Figure 8.16 shows a modification to the system of Fig. 8.15 which may be useful during partial-load operation. We assume that face dampers (not shown) on the cooling coil

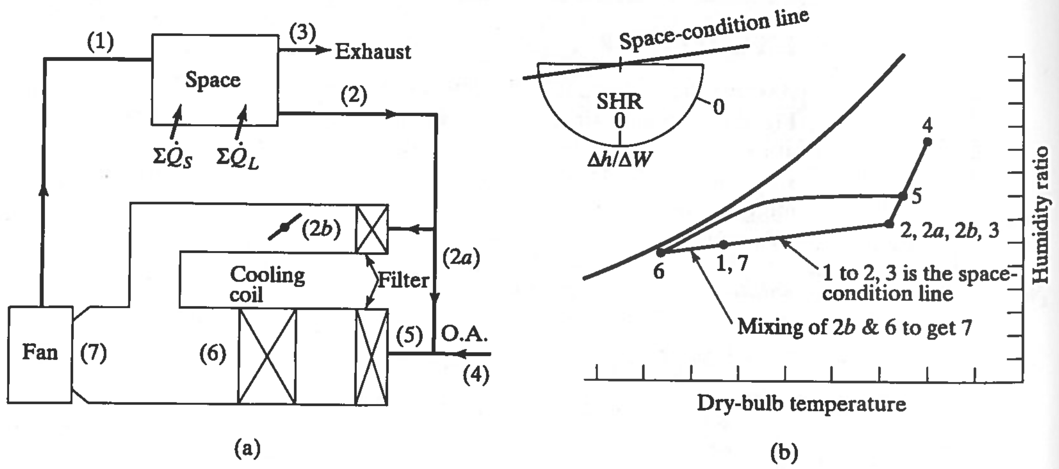


Figure 8.16 Schematic elementary summer air conditioning system with bypass of recirculated air.

and the bypass damper are controlled by a motor which positions them so as to maintain a constant space dry-bulb temperature. As the sensible-heat gain of the space decreases, more recirculated air is bypassed. However, the air which does pass across the coil may be more thoroughly dehumidified than when the full air quantity is handled. Thus, satisfactory space-humidity conditions may be maintained during some partial-load situations without the need for reheat.

Figure 8.16(b) shows the system state-points on a psychrometric chart. Although locations 2, 2*a*, 2*b*, and 3 are all at the same state, it is convenient to identify them separately to facilitate keeping track of mass flow rates in working problems. State 5 is the adiabatic mixture of return air with mass flow rate $\dot{m}_{a,2a}$ and outside air at a mass flow rate of $\dot{m}_{a,4} = \dot{m}_{a,3}$. Thus,

$$\dot{m}_{a,5} = \dot{m}_{a,2a} + \dot{m}_{a,4}$$

The supply-air condition, state 1, is assumed to be the same as that before the fan, state 7, which results from the adiabatic mixing of the air leaving the cooling coil (state 6 with a mass flow rate of $\dot{m}_{a,6} = \dot{m}_{a,5}$) and bypass air (state 2*b* with mass flow rate $\dot{m}_{a,2b}$). Thus state 7 is located along the line connecting states 6 and 2*b*. Notice that in calculating the cooling load, ${}_5\dot{Q}_6$, the mass flow rate through the coil is equal to $\dot{m}_{a,5}$, which is less than the supply air flow rate by an amount equal to the bypass air flow rate.

Figure 8.17 shows a summer air conditioning system with a reheat coil. Reheat is required when the space-condition line is so steep that a satisfactory intersection of it with the cooling-coil process line cannot be obtained. When a reheat coil is included, both temperature and humidity of a space may be controlled. However, the use of reheat results in more expensive system operation, since some of the heat added to the air as reheat must be removed in the cooling coil.

Figure 8.17(b) shows the system state-points on a psychrometric chart. In this type of system, the space condition (state 2) and the state of the air leaving the cooling coil (state 6) are often known. State 7, 1 then can be located at the intersection of the space-condition line drawn through state 2 and the sensible-heating process line (constant W) starting at state 6.

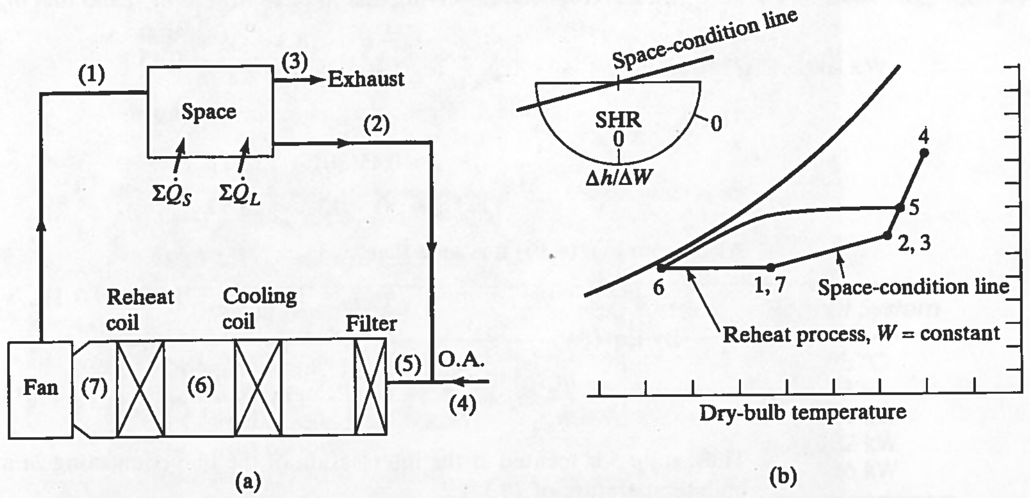


Figure 8.17 Schematic summer air conditioning system with reheat.

EXAMPLE 8.11

In order to compare the space conditions produced and the relative energy requirements of bypass and reheat systems, consider a space that has a sensible load of $\sum \dot{Q}_S = 205$ kW and a latent load of $\sum \dot{Q}_L = 88$ kW when the space is maintained at a dry-bulb temperature of 25 °C and the outdoor conditions are 35 °C dry-bulb temperature and 40 percent relative humidity. The space dry-bulb condition is to be met by using either of the systems shown in Figs. 8.16 or 8.17. Supply air is to be introduced to the space at a flow rate of 30 kg_a/s. The flow rate of exhaust air is 4.5 kg_a/s. The exit conditions of the systems cooling coils are a dry-bulb temperature of 10 °C and a relative humidity of 95 percent. For the bypass system shown in Fig. 8.16 determine (a) the relative humidity in the space and (b) the required system cooling capacity. For the reheat system shown in Fig. 8.17 determine (c) the relative humidity in the space, (d) the rate of heat required for the reheat coil, and (e) the required system cooling capacity.

Solution: The nomenclature of Fig. 8.16 will be used for parts (a) and (b). Standard atmospheric pressure is assumed and the solution carried out using psychrometric chart C-8SI. States 4 and 6 can be located from the given information. States 6, 7, and 2*b* fall on the mixing line connecting 6 and 2*b*. The slope of this line is determined by observing that it is also the space-condition line from state 1 to state 2. The sensible-heat ratio for the space-condition line is given by Eq. (8.27).

$$\text{SHR} = \frac{205}{205 + 88} = 0.70$$

(a) State 2 is located at the intersection of the line with an SHR = 0.70 drawn through state 6 and $t_2 = 25$ °C. The resulting point falls at $\phi_2 = 50$ percent. Thus using the system shown in Fig. 8.16, the space conditions will be $t_2 = 25$ °C and $\phi_2 = 50$ percent.

(b) State 5 is on the mixing line connecting states 4 and 2*a*. (Note that states 2, 2*a*, 2*b*, and 3 are the same.) In order to determine the division of the return air between $\dot{m}_{a,2a}$ and $\dot{m}_{a,2b}$, locate state 7, 1. By Eq. (8.28) with $\bar{c}_p = 1.02$ kJ/kg_a °C

$$(30)(1.02)(25 - t_1) = 205$$

$$t_1 = 18.3\ ^\circ\text{C}$$

By Eq. (8.9) and observing that $\dot{m}_{a,2b} = \dot{m}_{a,1} - \dot{m}_{a,6}$ and that $\dot{m}_{a,5} = \dot{m}_{a,6}$,

$$\frac{\dot{m}_{a,5}}{\dot{m}_{a,1}} = \frac{t_1 - t_2}{t_6 - t_2} = \frac{18.3 - 25}{10 - 25} = 0.45$$

Thus

$$\dot{m}_{a,5} = 0.45(30) = 13.5\ \text{kg}_a/\text{s}$$

$$\dot{m}_{a,2b} = 30 - 13.5 = 16.5\ \text{kg}_a/\text{s}$$

Also, from Fig. (8.16) it is seen that

$$\dot{m}_{a,2a} = \dot{m}_{a,1} - \dot{m}_{a,3} - \dot{m}_{a,2b} = 30 - 4.5 - 16.5 = 9.0\ \text{kg}_a/\text{s}$$

By Eq. (8.9)

$$t_5 = \frac{\dot{m}_{a,4}}{\dot{m}_{a,5}} t_4 + \frac{\dot{m}_{a,2a}}{\dot{m}_{a,5}} t_{2a} = \frac{4.5}{13.5}(35) + \frac{9.0}{13.5}(25) = 28.3\ ^\circ\text{C}$$

Thus, state 5 is located at the intersection of the line connecting 2*a* and 4 and a dry-bulb temperature of 28.3 °C.

By Eq. (8.14)

$$_5\dot{Q}_6 = \dot{m}_{a,5}[(h_6 - h_5) - (W_6 - W_5)h_{f,6}]$$

$$= 13.5[(28.3 - 57.8) - (0.0073 - 0.0113)(41.12)]$$

$$= -396\ \text{kW}$$

or the system cooling capacity required is

$$\dot{Q}_R = 396\ \text{kW}$$

The nomenclature of Fig. 8.17 will be used to analyze the reheat system.

(c) The dry-bulb temperature of the supply air is found in the same manner as it was in part (b) of this example. Therefore, $t_1 = 18.3$ °C. The state 1, 7 is located by following a sensible-heating process line (constant W) beginning at state 6 and ending at $t_1 = t_7 = 18.3$ °C. State 2 is located by drawing the space-condition line beginning at state 1 and ending at the prescribed space dry-bulb temperature of $t_2 = 25$ °C. In part (a) of this example the SHR for the space-condition line was calculated to be SHR = 0.70. Locating state 2 on the psychrometric chart, we read $\phi_2 = 43$ percent. Thus it is seen that use of the reheat system results in a lower relative humidity in the space than does the use of the bypass system (i.e., 43 percent vs. 50 percent).

(d) Applying Eq. (8.12), the reheat energy rate is

$$_6\dot{Q}_7 = \dot{m}_{a,1}(c_{pa} + W_6 c_{pw})(t_7 - t_6)$$

$$= 30[(1.00 + 0.0073)(1.86)](18.3 - 10)$$

$$= 252\ \text{kW}$$

(e) State 5 is the intersection of t_5 with the adiabatic mixing process line connecting states 2 and 4. Notice that in the reheat system the entire supply-airflow rate passes through the cooling coil. Therefore, $\dot{m}_{a,5} = \dot{m}_{a,1} = 30$ kg_a/s and

$$\dot{m}_{a,2} = \dot{m}_{a,1} - \dot{m}_{a,3} = 30 - 4.5 = 25.5\ \text{kg}_a/\text{s}$$

By Eq. (8.9)

$$t_5 = \frac{\dot{m}_{a,4}}{\dot{m}_{a,5}} t_4 + \frac{\dot{m}_{a,2a}}{\dot{m}_{a,5}} t_2 = \frac{4.5}{30}(35) + \frac{25.5}{30}(25) = 26.5\ ^\circ\text{C}$$

After locating state 5, we read $h_5 = 50.2$ kJ/kg_a and $W_5 = 0.0094$ kg_w/kg_a. By Eq. (8.14)

$$_5\dot{Q}_6 = 30[(28.3 - 50.5) - (0.0094 - 0.0113)(41.12)] = -664\ \text{kW}$$

or the system cooling capacity required is

$$\dot{Q}_R = 664\ \text{kW}$$

The results of the example are summarized in Table 8.1.

TABLE 8.1 Results of Ex. 8.11

Result or Setpoint	*Bypass System*	*Reheat System*
Space dry-bulb temp.	25 °C	25 °C
Space relative humidity	50%	43%
Required cooling capacity	396 kW	664 kW
Required reheat	—	252 kW
Total energy rate required	396 kW	916 kW

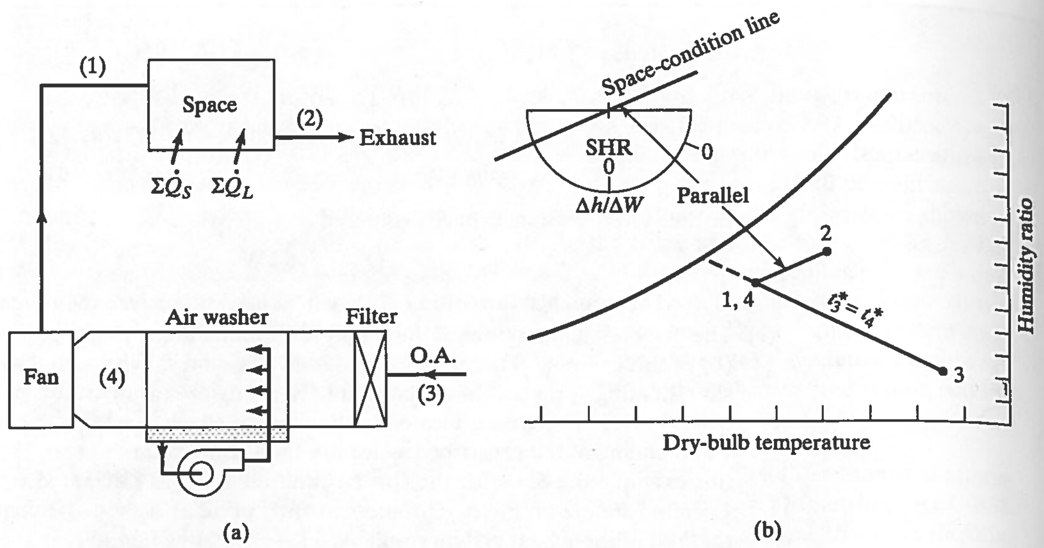


Figure 8.18 Schematic elementary evaporative-cooling system.

This example demonstrates that often a trade-off must be made between tight control of the conditions in a space and the energy needed to achieve that control.

Comfort air conditioning systems capable of maintaining optimum thermal conditions may be expensive to own and operate. Partially effective systems which involve much lesser costs may be attractive where finances preclude the installation of a completely effective system. In hot dry regions, evaporative-cooling systems may be capable of providing considerable relief in enclosed spaces. Such systems may also have application in hot industrial environments. Figure 8.18 shows an elementary evaporative-cooling system. We assume that state 2 is an acceptable space condition, although not necessarily an optimum one. State 3 of the outdoor air is assumed to be at a much higher temperature but lower humidity ratio than State 2. Through use of an air washer as the only processing device, an acceptable air conditioning system may result. Generally, a much higher flow rate of air is used with an evaporative-cooling system than with conventional systems. A high rate of air movement past a person allows the same degree of comfort but with higher effective temperatures, as compared to situations where air movement is slight.

Modifications may be made among the psychrometric systems previously discussed in order to achieve specific purposes. For example, the addition of a cooling coil between the air washer and reheater of Fig. 8.14 would provide a year-round air conditioning system.

In designing a psychrometric system, every effort should be made to utilize heat transfers internal to the system where economically feasible so as to reduce the need for external heat transfers. Figure 8.19 shows one case. We assume winter operation and a system which requires 100 percent outdoor air. The outdoor air is preheated by waste heat in the exhaust air. The same recovery scheme of Fig. 8.19 might be applied for an opposite purpose in Fig. 8.18. Here, the outdoor air might be precooled by the exhaust air, allowing state 2 to be at a lower effective temperature for the same airflow rate. Other possibilities for using internal heat transfers might include the use of refrigerant-condenser heat for reheating of moist air. When outdoor air temperatures are high, reheat-

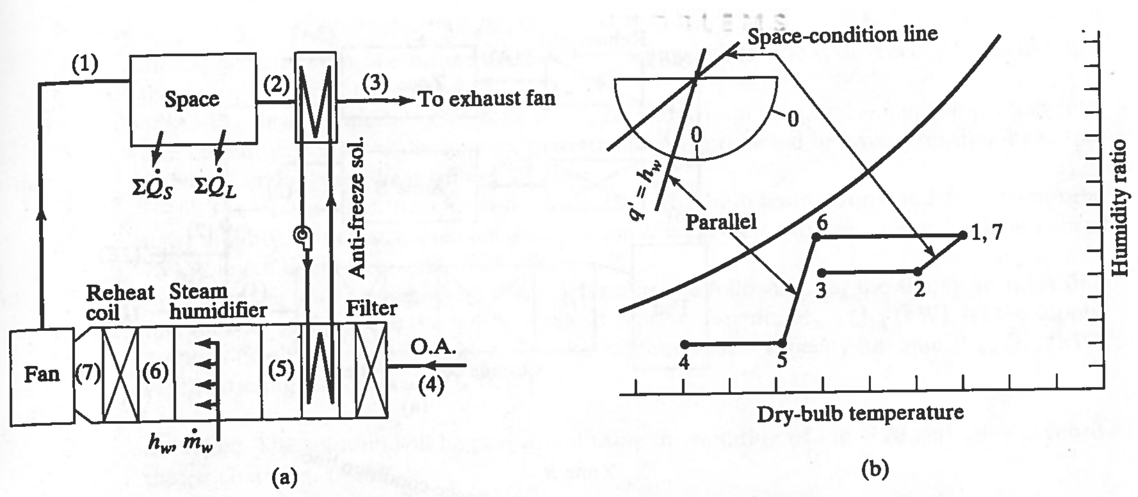


Figure 8.19 Schematic winter air conditioning system using 100 percent outdoor air with preheating by waste heat from the exhaust air.

ing may be accomplished internally by a heat-exchanger system involving the reheat coil and a coil in the outdoor-air duct.

In many applications the required refrigerating capacity chargeable to the outdoor air taken into the system is a large fraction of the total. In such cases, and in winter systems where heating is required, the use of outdoor air should be kept to a minimum. However, for spaces which may require cooling during the entire year, provision should be made for flexible use of outdoor air, so that refrigerating equipment may be shut down during cold weather.

8.4 PSYCHROMETRIC SYSTEMS—MULTIPLE ZONE

It was previously mentioned that, in most large buildings, the heating and cooling loads will vary throughout different locations in the building and that in order to maintain comfort in the entire building it is necessary to subdivide the building into zones. This section presents descriptions of three multizone systems which can be used to meet the loads in the various zones and thus provide the desired space conditions in each of them. The three systems described are simplified versions of reheat, variable-air-volume, and dual-duct systems. Again, the purpose is to illustrate general principles rather than attempt to cover all combinations and aspects of multizone systems. Multizone systems are described in greater detail in the *HVAC Systems and Applications Volume* of the *ASHRAE Handbook* series [3].

Reheat. A multizone reheat system is a modification of the single-zone system previously presented in Fig. 8.17. As the term "reheat" implies, heat is added as a secondary process. Figure 8.20 schematically shows a simple two-zone reheat system. Generally, multizone systems consist of more than two zones. However, the principles associated with them can be illustrated considering just two zones. The system produces conditioned air from a central unit at a fixed cold-air temperature and low humidity level, state 5. The mass flow rates are generally selected to offset the maximum cooling loads

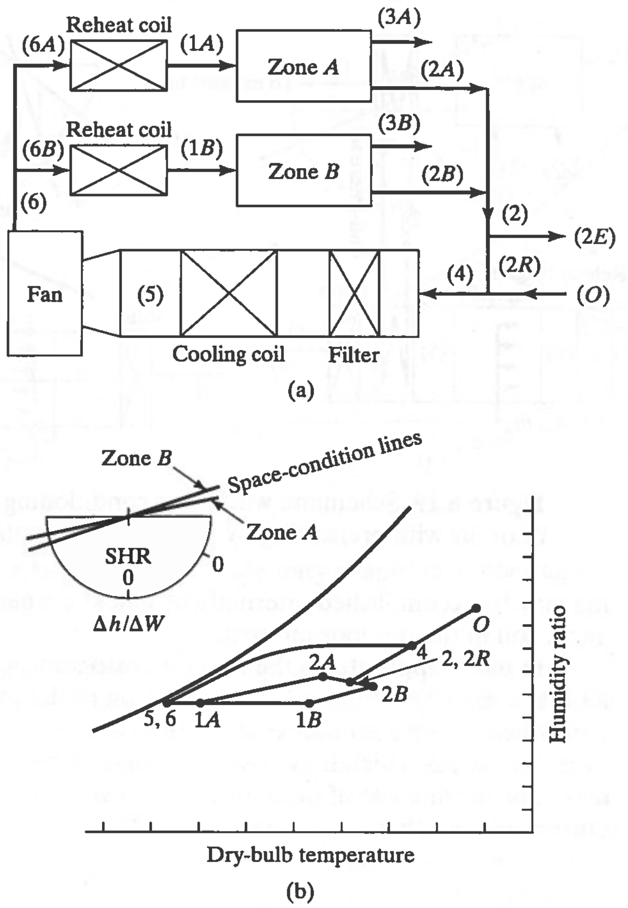


Figure 8.20 Schematic of a two-zone reheat system.

in the zones. A control thermostat in each zone calls for reheat when the cooling load in the zone drops below the design value. Return air from the zones ($\dot{m}_{a,2A}$ and $\dot{m}_{a,2B}$) mixes in the return-air duct, resulting in state 2. A portion of this air ($\dot{m}_{a,2E}$) is expelled to the outside through a main system exhaust duct, while the balance ($\dot{m}_{a,2R}$) is returned and mixed with the outside air, resulting in state 4. In addition to the main system exhaust, each zone may have its own direct exhaust flow. These are shown in Fig. 8.20 as mass flows 3A and 3B. In general, these flows are relatively small and are present to discharge localized air contaminants. The mass flow rate of outside air is equal to the sum of all exhaust flow rates.

EXAMPLE 8.12

A two-zone reheat system similar to the one shown in Fig. 8.20 is to be designed for Minneapolis summer outdoor design conditions of 32 °C dry-bulb temperature and 23 °C coincident wet-bulb temperature. In this design the only exhaust is the main system exhaust. Therefore, the mass flow rates labeled 3A and 3B in Fig. 8.20 are nonexistent. Twenty-five percent of the supply-air mass flow rate to each zone is to be fresh air (outside air). Standard

atmospheric pressure is assumed to exist. The cooling coil for the system is to be selected such that the air will exit the coil saturated at 5 °C.

The design conditions for zone *A* are 22 °C dry-bulb temperature and 40 percent relative humidity. For these design conditions, zone *A* is predicted to have a sensible-heat gain of 80 kW and a latent-heat gain of 20 kW.

The design conditions for zone *B* are 26 °C dry-bulb temperature and 30 percent relative humidity. For these design conditions zone *B* is predicted to have a sensible-heat gain of 75 kW and a latent-heat gain of 25 kW.

For the above design conditions, determine the following: (a) the supply-air mass flow rate for zone *A*, kg$_a$/s, (b) the required reheat capacity for zone *A*, ${}_{6A}\dot{Q}_{1A}$ (kW), (c) the supply-air mass flow rate for zone *B*, kg$_a$/s, (d) the required reheat capacity for zone *B*, ${}_{6B}\dot{Q}_{1B}$ (kW), (e) the cooling-coil capacity, $\dot{Q}_R$ (kW).

Solution: The solution will be performed using the notation of Fig. 8.20 and using psychrometric chart Fig. C-8SI.

(a) The mass flow rate for zone *A* can be determined using Eq. (8.25) once state 1A is determined. Since only sensible processes occur between state 5 and state 1A, $W_{1A} = W_5 = 0.0054$ kg$_w$/kg$_a$. By Eq. (8.30) the SHR of the condition line for zone *A* is calculated.

$$\text{SHR} = \frac{80}{80 + 20} = 0.80$$

With the aid of the SHR protractor, the space-condition line is drawn through the given state 2A. State 1A is the intersection of the condition line with humidity ratio W_{1A}. The result is $t_{1A} = 11.0$ °C. By Eq. (8.28)

$$\dot{m}_{a,1A} = \frac{\sum \dot{Q}_S}{c_p(t_{2A} - t_{1A})} = \frac{80}{1.02(22.0 - 11.0)} = 7.13 \text{ kg}_a/\text{s}$$

(b) By Eq. (8.12)

$${}_{6A}\dot{Q}_{1A} = 7.13[1.00 + 1.86(0.0054)](11.0 - 5.0) = 43.2 \text{ kW}$$

(c) Using the same procedure as that used in part (a), we calculate the following values for Zone *B*.

$$\text{SHR} = 0.75$$
$$t_{1B} = 20.0 \text{ °C}$$
$$\dot{m}_{a,1B} = 12.25 \text{ kg}_a/\text{s}$$

(d) Using the same procedure as that used in part (b),

$${}_{6B}\dot{Q}_{1B} = 12.25[1.00 + 1.86(0.0054)](20.0 - 5.0) = 185.6 \text{ kW}$$

(e) The refrigeration or cooling-coil capacity is calculated combining Eqs. (8.14) and (8.15):

$$\dot{Q}_R = \dot{m}_{a,5}[(h_4 - h_5) - (W_4 - W_5)h_{f,5}]$$

In order to determine state 5, we must first determine state 4, which requires knowledge of state 2R. State 2R is the same as state 2, which results from the adiabatic mixing of states 2A and 2B. Thus, state 2 is found by drawing a line connecting points 2A and 2B and then using any one of the mixing equations (8.3), (8.4), or (8.9) to establish the proper location on the line. By Eq. (8.9)

$$t_2 = \frac{7.13}{19.38}(22.0) + \frac{12.25}{19.38}(26.0) = 24.5 \text{ °C}$$

State 4 is a mix of 75 percent return air, state $2R$, and 25 percent outside air, state O. By Eq. (8.9)

$$t_4 = (0.75)(24.5) + (0.25)(32.0) = 26.4\ ^\circ\mathrm{C}$$

Thus, using the mixing line connecting states $2R$ and O and the temperature $t_4 = 26.4\ ^\circ\mathrm{C}$, state 4 is established. From the psychrometric chart

$$h_4 = 47.6\ \mathrm{kJ/kg}_a \quad \text{and} \quad W_4 = 0.0083\ \mathrm{kg}_w/\mathrm{kg}_a$$

The conditions of the moist air leaving the cooling coil, state 5, were specified as part of the design. After locating state 5 on the psychrometric chart, it is found that

$$h_5 = 18.6\ \mathrm{kJ/kg}_a \quad \text{and} \quad W_5 = 0.0054\ \mathrm{kg}_w/\mathrm{kg}_a$$

From Table A.6SI $h_{f,5} = 20.44\ \mathrm{kJ/kg}_w$. Therefore, the required refrigeration capacity is

$$\dot{Q}_R = (19.38)[(47.6 - 18.6) - (0.0083 - 0.0054)(20.44)] = 560.9\ \mathrm{kW}$$

Variable Air Volume. A variable-air-volume (VAV) system controls the dry-bulb temperature in a zone by varying the supply-airflow rate rather than the supply-air temperature. A space thermostat provides a control signal to a "VAV unit," which controls the supply airflow. Several VAV unit designs exist. Some are duct-mounted devices which control the flow by varying the settings of dampers or pressure-reducing boxes. Other VAV units control the flow at the terminal diffuser or grille. The fan system is designed to handle the largest simultaneous block load, not the sum of the peaks. At a given time, zones experiencing peak loads essentially borrow the extra air from the zones experiencing off-peak loads. Depending on the system size and complexity and the lowest expected flow rate, additional fan controls may be used to reduce fan power and to limit system noise at part-load operating conditions. Many methods are available to achieve this, including fan-speed control, variable-inlet-vane control, fan bypass, fan-discharge dampers, and variable-pitch fan control. These control methods are described in detail in the *1988 Equipment Volume* of the *ASHRAE Handbook* series [4].

Simple VAV systems typically cool only and have no requirement for simultaneous heating and cooling in various zones. In instances where there are exterior zones in a building that require heating, secondary systems are used to provide the necessary heat. Secondary systems used include baseboard heaters, radiant heaters, and independent constant-volume, variable-temperature air systems. Figure 8.21(a) schematically shows a simple two-zone VAV system. Figure 8.21(b) is a sketch of a psychrometric chart showing the state-points and process lines for the simple VAV system where the two zones are to be maintained at different temperatures, t_{2A} and t_{2B}. The locations of states $2A$ and $2B$ on a psychrometric chart can be determined by constructing the space-condition lines for the zones. Since the VAV units do not change the state of the moist air, states $1A$, $1B$, and 6 are the same. Thus, the space-condition lines can be drawn starting at state 6 and ending at the respective zone dry-bulb temperatures.

The slopes of the space-condition lines are established with the aid of the protractor, using the sensible-heat ratios for the two zones as calculated from their respective loads. Once the conditions in a zone are established, the mass flow rate for that zone can be calculated using Eq. (8.25) or Eq. (8.28). Return air from the zones ($\dot{m}_{a,2A}$ and $\dot{m}_{a,2B}$) mix in the return-air duct, resulting in state 2. A portion of this air ($\dot{m}_{a,2E}$) is expelled to

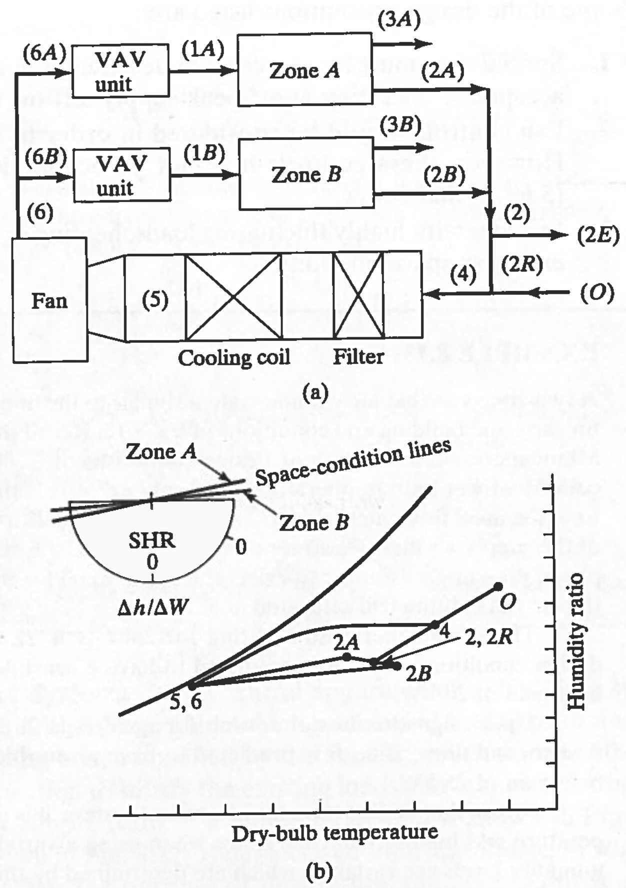


Figure 8.21 Schematic of a two-zone variable-air-volume system.

the outside through a main system exhaust duct, while the balance ($\dot{m}_{a,2R}$) is returned and mixed with the outside air, resulting in state 4. In addition to the main system exhaust, each zone may have its own direct exhaust flow. These are shown in Fig. 8.21 as mass flows $3A$ and $3B$. In general, these flows are relatively small and are present to discharge localized air contaminants. The mass flow rate of outside air ($\dot{m}_{a,O}$) is equal to the sum of all exhaust flow rates.

The *HVAC Systems and Applications Volume* of the *ASHRAE Handbook* series [3] presents an extensive list of advantages and design precautions for variable-air-volume systems. Some of the advantages listed are:

1. The VAV concept, when combined with one of the perimeter heating systems, offers inexpensive temperature control for multiple zoning.
2. Advantage may be taken of changing loads from lights, occupancy, solar, and equipment to lower the cost for fans, refrigeration, heating, and associated plant auxiliaries.
3. VAV system designs can be made virtually self-balancing.
4. It is easy and inexpensive to subdivide into new zones and to handle increased loads with new tenancy or usage if the system is designed for a potential load increase.

Some of the design precautions listed are:

1. Special care must be exercised in designing the air distribution in a zone to ensure acceptable operation at off-peak supply-airflow rates.
2. Fan controls should be considered in order to save power and for noise control. However, these controls may not be economical for systems of 10,000 ft^3/min ($5\ m^3/s$) and below.
3. In zones with highly fluctuating loads, heating or reheat may be required to prevent excessive space humidity.

EXAMPLE 8.13

A two-zone variable-air-volume system similar to the one shown in Fig. 8.21 is to be designed for the same building and conditions of Ex. 8.12. Recall that the design is to be conducted for Minneapolis summer outdoor design conditions of 32 °C dry-bulb temperature and 23 °C coincident wet-bulb temperature. The only exhaust is the main system exhaust and, therefore, the mass flow rates labeled 3*A* and 3*B* in Fig. 8.21 are nonexistent. Twenty-five percent of the supply-air mass flow rate to each zone is to be fresh air (outside air). Standard atmospheric pressure is assumed to exist. The cooling coil for the system is to be selected such that the air will exit the coil saturated at 5 °C.

The design thermostat setting for zone *A* is 22 °C dry-bulb temperature. For the design conditions, zone *A* is predicted to have a sensible-heat gain of 80 kW and a latent-heat gain of 20 kW.

The design thermostat setting for zone *B* is 26 °C dry-bulb temperature. For the design conditions, zone *B* is predicted to have a sensible-heat gain of 75 kW and a latent-heat gain of 25 kW.

Notice that, unlike the case of a reheat system, it is not possible to specify both the temperature and humidity in each space when using a variable-air-volume system. Rather, the humidity levels are variables which are determined by the exit conditions of the cooling coil and the space loads.

For the above design conditions, determine the following: (a) the relative humidities in the zones, (b) the mass flow rates of supply air for each zone, $\dot{m}_{a,1A}$ and $\dot{m}_{a,1B}$, (c) the cooling-coil capacity, $\dot{Q}_R$ (kW).

Solution: The solution will be performed using the notation of Fig. 8.21 and using the psychrometric chart, Fig. C-8SI.

(a) The locations of states 2*A* and 2*B* are found by constructing the space-condition lines, using the appropriate sensible-heat ratios, starting at the supply-air state (6) and terminating at the respective zone temperatures (t_{2A} and t_{2B}). As previously determined in Ex. 8.12, the sensible-heat ratios for the two zones are:

$$(\text{SHR})_A = 0.80 \quad \text{and} \quad (\text{SHR})_B = 0.75$$

The construction results in:

$$W_{2A} = 0.0072\ \text{kg}_w/\text{kg}_a, \quad \phi_{2A} = 0.41, \quad h_{2A} = 40.3\ \text{kJ/kg}_a$$

and

$$W_{2B} = 0.0084\ \text{kg}_w/\text{kg}_a, \quad \phi_{2B} = 0.40, \quad h_{2B} = 47.4\ \text{kJ/kg}_a$$

(b) The mass flow rates can be calculated using either Eq. (8.25) or (8.28). Using Eq. (8.25)

$$\dot{m}_{a,1A} = \frac{80 + 20}{40.3 - 18.6} = 4.61\ \text{kg}_a/\text{s}$$

and

$$\dot{m}_{a,1B} = \frac{75 + 25}{47.4 - 18.6} = 3.47\ \text{kg}_a/\text{s}$$

(c) In order to calculate the cooling-coil capacity, it is necessary to determine state 4, which results from the adiabatic mixing of 25 percent outside air (state *O*) and 75 percent of the return air (state 2). Applying Eq. (8.4).

$$h_2 = \frac{4.61}{8.08}(40.3) + \frac{3.47}{8.08}(47.4) = 43.3\ \text{kJ/kg}_a$$

and

$$h_4 = (0.75)(43.3) + (0.25)(67.8) = 49.4\ \text{kJ/kg}_a$$

and from the psychrometric chart

$$W_4 = 0.0091\ \text{kg}_w/\text{kg}_a$$

Therefore the refrigeration capacity required is

$$\dot{Q}_R = (8.08)[(49.4 - 18.6) - (0.0091 - 0.0054)(20.44)] = 248.3\ \text{kW}$$

Dual-Duct Systems. The central apparatus of a dual-duct system produces two air streams, one hot and one cold, which are carried by separate ducts to each zone. At each zone a mixing box, controlled by the zone thermostat, mixes the two air streams in the correct proportion to satisfy the existing loads and maintain the desired zone dry-bulb temperature. A simple form of a dual-duct system is shown in Fig. 8.22. The psychrometric chart shown in Fig. 8.22(b) depicts a cooling application in which the humidifier is not operational. For the purpose of illustration, the temperatures of the two zones are shown being maintained at different values, t_{2A} and t_{2B}. The return airflows from each zone mix adiabatically, resulting in state 2. A portion of the mixed air is exhausted, and the balance is mixed with outside air. The dampers that control the percentages of return and outside air that are mixed have several settings, ranging from a minimum opening for very high outdoor-air temperatures to a maximum opening when outdoor air can be used for cooling without the need for mechanical refrigeration.

When a dual-duct system is used for cooling, the temperature of the air in the cold duct typically is maintained between 50 and 60 °F (10 to 15.5 °C) and the temperature of the air in the warm duct, t_8, typically is maintained at approximately 5 °F (3 °C) higher than the temperature of the return air. Since the proper supply-air state for each zone is achieved by mixing the hot and cold air streams, the locations of the supply-air states 1*A* and 1*B* fall on the adiabatic mixing line connecting states 6 and 8. The lines connecting states 1*A* and 2*A* and states 1*B* and 2*B* are the space-condition lines for zones *A* and *B*, respectively.

The system shown in Fig 8.22(a) also provides heating capabilities for winter operation. In this case, the hot-duct temperature is automatically set progressively higher as the outdoor temperature drops. The temperature of the air in the cold duct is maintained between 50 and 60 °F (10 to 15.5 °C) either by using mechanical refrigeration or, when outdoor conditions permit, by using the proper mix of outside and return air.

The principal advantage of a dual-duct system is its ability to maintain good control of temperature and humidity in several zones which may have a relatively wide range of

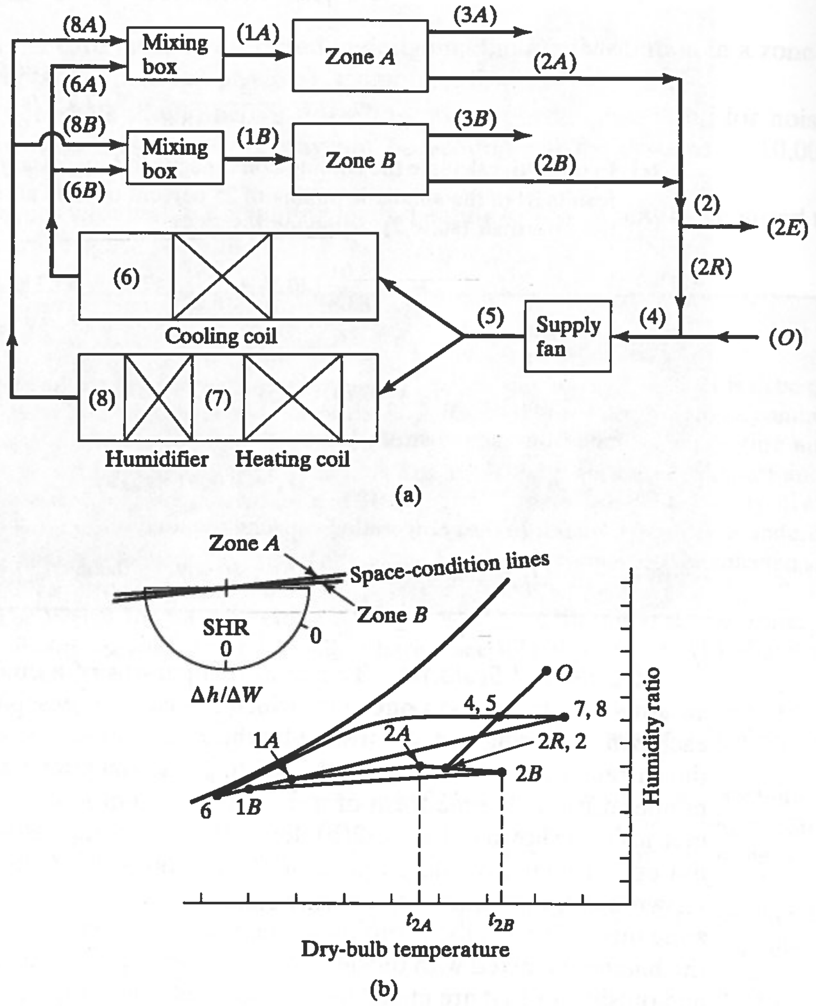


Figure 8.22 Schematic of a two-zone dual-duct system.

loads, including the condition where cooling is required in some zones while other zones require heating. On the other hand, some of the disadvantages include the relatively high energy use associated with both heating and cooling the air streams and the high initial cost associated with the installation of two air ducts.

EXAMPLE 8.14

A two-zone dual-duct system similar to the one shown in Fig. 8.22 is to be designed for the same building and conditions of Ex. 8.12 and 8.13. Recall that the design is to be conducted for Minneapolis summer outdoor design conditions of 32 °C dry-bulb temperature and 23 °C coincident wet-bulb temperature. The only exhaust is the main system exhaust, and, therefore, the mass flow rates labeled 3A and 3B in Fig. 8.22 are nonexistent. Twenty-five percent of the supply-air mass flow rate to each zone is to be fresh air (outside air). Standard atmospheric pressure is assumed to exist. The cooling coil for the system is to be selected such that

the air will exit the coil saturated at 5 °C. The humidifier shown in Fig. 8.22 is not used, and the temperature in the hot duct is maintained at 28 °C.

The design thermostat setting for zone A is 22 °C dry-bulb temperature. For the design conditions, zone A is predicted to have a sensible-heat gain of 80 kW and a latent-heat gain of 20 kW. The design supply-airflow rate for zone A is 7.1 kg_a/s.

The design thermostat setting for zone B is 26 °C dry-bulb temperature. For the design conditions, zone B is predicted to have a sensible-heat gain of 75 kW and a latent-heat gain of 25 kW. The design supply-airflow rate for zone B is 12.2 kg_a/s.

This problem is posed in a manner that represents how the system would actually operate; i.e., the exiting temperature of the heating coil, the exiting temperature and humidity of the cooling coil, and thermostat settings are specified. These parameters represent control settings and the operational characteristics of the cooling coil. The humidity levels in the two zones will be determined by the system equilibrium conditions and cannot be specified.

For the above design conditions, determine the following: (a) the mass flow rates of cold and hot air that make up the supply air for zones A and B, (b) the rate of heat addition for the heating coil, ${}_5\dot{Q}_7$ (kW), (c) the humidity levels in the zones, and (d) the cooling-coil capacity, $\dot{Q}_R$ (kW).

Solution: The solution will be performed using the notation of Fig. 8.22 and using psychrometric chart Fig. C-8SI. Efforts to locate all the state-points on the psychrometric chart reveal that all the temperatures can be calculated but there is insufficient information to establish the humidity levels. If, on the other hand, we knew one additional humidity value, e.g., ϕ_4, we could completely solve the problem. Therefore, the approach will be to calculate as many quantities as possible, using only temperatures, and then use an iterative method to determine the balance of the state-points and thus complete the problem. The iterative method will consist of guessing a humidity level for state 4 and using this value to establish all the states including a calculated value for state 4. When the guess and calculated values for state 4 agree to within the reading accuracy of the psychrometric chart, the location of the state-points is complete. This iterative process is somewhat similar to how the system operates, in that humidity levels would continually adjust until equilibrium was achieved.

(a) First determine the supply-air temperatures for each zone using Eq. (8.28).

$$t_{1A} = t_{2A} - \frac{\sum \dot{Q}_S}{c_p(\dot{m}_{a,1A})} = 22.0 - \frac{80}{1.02(7.1)} = 11.0\ ^\circ\mathrm{C}$$

Similarly

$$t_{1B} = 26.0 - \frac{75}{1.02(12.2)} = 20.0\ ^\circ\mathrm{C}$$

Using Eqs. (8.9) and (8.2),

$$\frac{\dot{m}_{a,6A}}{\dot{m}_{a,1A}} = \frac{t_{1A} - t_8}{t_6 - t_8} = \frac{11.0 - 28.0}{5.0 - 28.0} = 0.74$$

or

$$\dot{m}_{a,6A} = 0.74(7.1) = 5.25\ \mathrm{kg_a/s\ of\ cold\ air}$$

The mass flow rate of hot air for zone A is

$$\dot{m}_{a,8A} = 7.1 - 5.25 = 1.85\ \mathrm{kg_a/s}$$

A similar calculation for zone B results in

$$\dot{m}_{a,6B} = 4.27\ \mathrm{kg_a/s\ of\ cold\ air}$$

and

$$\dot{m}_{a,8B} = 7.93\ \text{kg}_a/\text{s of hot air}$$

Equation (8.9) can also be used to calculate the temperatures at states 2 and 4:

$$t_2 = \frac{7.1}{19.3}(22) + \frac{12.2}{19.3}(26) = 24.5\ ^\circ\text{C}$$

and

$$t_4 = t_5 = 0.75(24.5) + 0.25(32) = 26.4\ ^\circ\text{C}$$

(b) The temperatures and mass flow rates are sufficient to calculate the heat addition of the heating coil. Since the humidifier is not in operation, states 7 and 8 are the same. Therefore:

$$\begin{aligned} {}_5\dot{Q}_7 &= (\dot{m}_{a,8A} + \dot{m}_{a,8B})\bar{c}_p(t_8 - t_5) \\ &= (1.85 + 7.93)(1.02)(28.0 - 26.4) \\ &= 16.0\ \text{kW} \end{aligned}$$

(c) As previously indicated, it is necessary to iterate to establish the humidity levels in the zones. A number of options exist. We could assume humidity levels in both zones, complete the calculations, and then check the validity of our assumption. Another approach is to assume (guess) a humidity level for the mixed-air condition, state 4, complete the calculations, and check our assumption. Since the latter approach requires guessing only one value, it is the one selected. When we refer to the psychrometric chart and recall that the mixed air is 75 percent return and 25 percent outside air, it appears that a reasonable first iteration would be to guess a relative humidity of $\phi_4 = 40$ percent. State 8 is found by following a line of constant humidity ratio from state 4 to $t_8 = 28\ ^\circ$C. States 1*A* and 1*B* are located along the mixing line connecting states 6 and 8 at temperatures $t_{1A} = 11.0\ ^\circ$C and $t_{1B} = 20.0\ ^\circ$C, respectively. As previously determined in Ex. 8.12, the sensible-heat ratios for zones *A* and *B* are 0.80 and 0.75, respectively. Drawing the $(\text{SHR})_A = 0.80$ line starting at state 1*A* and ending at t_{2A} and drawing the $(\text{SHR})_B = 0.75$ line starting at state 1*B* and ending at $t_{2B} = 26.0\ ^\circ$C results in $\phi_{2A} = 43$ percent and $\phi_{2B} = 40$ percent. State 2 is located on the mixing line connecting 2*A* and 2*B* at $t_2 = 24.5\ ^\circ$C. Thus, $\phi_2 = 41$ percent. Locating state 4 at $t_4 = 26.4\ ^\circ$C along the line connecting states 2*R* and *O* results in $\phi_4 = 44$ percent as compared to our guess value of 40 percent. This variation is considered large enough to warrant a continuation of the solution, using a second guess value for the humidity of state 4. Recall that convergence of the iterative procedure is considered to have been achieved when the guessed and calculated values agree within the reading accuracy of the psychrometric chart. In this case, we should be able to read the relative humidity to ± 0.02.

Repeating the above calculations using a second guess value of $\phi_4 = 0.45$ satisfies the convergence criteria, and the following state-points are found.

$$\begin{aligned}
t_{1A} &= 11.0\ ^\circ\text{C}, & \phi_{1A} &= 0.80 & W_{1A} &= 0.0065\ \text{kg}_w/\text{kg}_a, & h_{1A} &= 27.4\ \text{kJ/kg}_a \\
t_{1B} &= 20.0\ ^\circ\text{C}, & \phi_{1B} &= 0.56 & W_{1B} &= 0.0081\ \text{kg}_w/\text{kg}_a, & h_{1B} &= 40.6\ \text{kJ/kg}_a \\
t_{2A} &= 22.0\ ^\circ\text{C}, & \phi_{2A} &= 0.45 & W_{2A} &= 0.0119\ \text{kg}_w/\text{kg}_a, & h_{2A} &= 52.2\ \text{kJ/kg}_a \\
t_{2B} &= 26.0\ ^\circ\text{C}, & \phi_{2B} &= 0.42 & W_{2B} &= 0.0088\ \text{kg}_w/\text{kg}_a, & h_{2B} &= 48.4\ \text{kJ/kg}_a \\
t_2 &= 24.5\ ^\circ\text{C}, & \phi_2 &= 0.43 & W_2 &= 0.0082\ \text{kg}_w/\text{kg}_a, & h_2 &= 45.4\ \text{kJ/kg}_a \\
t_4 &= 26.4\ ^\circ\text{C}, & \phi_4 &= 0.45 & W_4 &= 0.0097\ \text{kg}_w/\text{kg}_a, & h_4 &= 51.1\ \text{kJ/kg}_a \\
t_8 &= 28.0\ ^\circ\text{C}, & \phi_8 &= 0.41 & W_8 &= 0.0097\ \text{kg}_w/\text{kg}_a, & h_8 &= 52.8\ \text{kJ/kg}_a
\end{aligned}$$

(d) The cooling capacity can now be calculated by combining Eqs. (8.14) and (8.15) and observing that states 4 and 5 are equal.

$$\begin{aligned} \dot{Q}_R &= (\dot{m}_{a,6A} + \dot{m}_{a,6B})[(h_5 - h_6) - (W_5 - W_6)h_{f,6}] \\ &= (5.25 + 4.27)[(51.1 - 18.6) - (0.0097 - 0.0054)(20.44)] \\ &= 308.6\ \text{kW} \end{aligned}$$

TABLE 8.2 Comparative Results for Reheat, VAV, and Dual-Duct Systems

	Reheat	*VAV*	*Dual Duct*
Zone *A* temperature and humidity	22 °C, 40%	22 °C, 41%	22 °C, 45%
Zone *B* temperature and humidity	26 °C, 30%	26 °C, 40%	26 °C, 42%
Required reheat or heating-coil capacity, kW	228.8	N/A	16.0
Required cooling capacity, kW	560.9	248.3	308.6

Examples 8.12, 8.13, and 8.14 demonstrate the application of three different types of HVAC systems to the same building. The key results of using the different systems are tabulated in Table 8.2.

The use of either the VAV or the dual-duct system results in a higher space humidity than the use of the reheat system, but at a significant reduction in the rate of energy consumption. The supply-airflow rates for the dual-duct system in Ex. 8.14 were selected as approximately equal to those used by the reheat system. A better design may have been to use lower flow rates. This results in lower supply-air temperatures and humidities and will reduce the humidity levels in the two zones. However, the minimum flow rates allowable would be those equal to the ones determined for the VAV system, since at those flow rates the supply-air conditions correspond to the cooling-coil exit conditions (i.e., 100 percent cold air and no air from the hot deck). Thus the minimum space humidities available using the dual-duct system correspond to those of the VAV system. One feature of the dual-duct system not demonstrated in the example is its ability to simultaneously heat and cool different zones, which cannot be readily accomplished with the other two systems.

ENDNOTES

1. *ASHRAE Handbook, Fundamentals Volume* (Atlanta: American Society of Heating, Refrigerating and Air Conditioning Engineers, 1989), 26.12 (IP Edition).
2. *ASHRAE Handbook, Equipment Volume* (Atlanta: American Society of Heating, Refrigerating and Air Conditioning Engineers, 1988), 4.1.
3. *ASHRAE Handbook, HVAC Systems and Applications Volume* (Atlanta: American Society of Heating, Refrigerating and Air Conditioning Engineers, 1987).
4. *ASHRAE Handbook, Equipment Volume* (Atlanta: American Society of Heating, Refrigerating and Air Conditioning Engineers, 1988).

PROBLEMS

8.1 In a heating/humidifying system moist air first flows through a heating coil and then through an air washer. The air enters the system at 55 °F dry-bulb temperature and 40 percent relative humidity. The air exits the system at 90 °F dry-bulb temperature and 65 °F thermodynamic wet-bulb temperature. The flow rate through the system is 2200 lbm_a/hr and the process occurs at a pressure of 29.921 in. Hg. Determine:

(a) the dry-bulb temperature of the air as it exits the heating coil,

(b) the rate of heat addition to the air by the heating coil, and

(c) the rate of moisture addition to the air by the adiabatic saturator.

8.2 A space to be conditioned has a sensible-heat loss of 80,000 Btu/hr and a latent-heat loss of 34,000 Btu/hr. The conditions in the space are maintained at 70 °F dry-bulb temperature and 44 °F dew-point temperature. Supply air is introduced into the space at a dry-bulb temperature of 95 °F (barometric pressure = 29.921 in. Hg). Determine

(a) the required relative humidity of the supply air, and

(b) the flow rate of supply air in lbm_a/hr.

8.3 Clearly show and label the following process lines and state-points on a psychrometric chart.

(a) Outdoor air at 35 °F dry-bulb temperature and 100 percent relative humidity is heated in a furnace to 100 °F dry-bulb temperature.

(b) Saturated steam at 200 °F is then sprayed into the air to increase its relative humidity to 40 percent.

(c) The air is then supplied to a space with a condition line having a sensible-heat ratio of 0.7, and the air exits the space at a dry-bulb temperature of 70 °F.

8.4 Moist air enters a cooling coil at a dry-bulb temperature of 88 °F and a relative humidity of 45 percent at a volumetric flow rate of 4500 ft³/min. Barometric pressure is 12.75 psia. The air exits the coil at a dry-bulb temperature of 52 °F and a dew-point temperature of 48 °F. *Without using a psychrometric chart*, calculate the cooling capacity of the coil, Btu/hr, for these operating conditions.

8.5 Moist air enters a cooling coil at 28 °C dry-bulb temperature and 50 percent relative humidity and exits the coil at 13 °C dry-bulb temperature and 90 percent relative humidity. The flow rate through the coil is 1.50 kg_a/s and the process occurs at a pressure of 101.325 kPa. Determine:

(a) the sensible-heat ratio for the process,

(b) the cooling-coil capacity (heat-transfer rate),

(c) the apparatus dew-point temperature, and

(d) the bypass factor for the coil.

8.6 Moist air at 84 °F dry-bulb temperature and 70 °F thermodynamic wet-bulb temperature enters a perfect-contact refrigeration coil at a rate of 3500 ft³/min. The air leaves the coil at 54 °F. Assume 14.696 psia pressure. Determine the tons of refrigeration required.

8.7 Moist air enters a refrigeration coil at 89 °F dry-bulb temperature and 65 °F thermodynamic wet-bulb temperature at a rate of 1400 ft³/min. The surface temperature of the coil is 55 °F. If 3.5 tons of refrigeration are available, find the dry-bulb and wet-bulb temperatures of the air leaving the coil. Assume sea-level pressure.

8.8 Saturated steam at a pressure of 25 psia is sprayed into a stream of moist air. The initial condition of the air is 55 °F dry-bulb temperature and 35 °F dew-point temperature. The mass rate of air flow is 2000 lbm_a/min. Barometric pressure is 14.696 psia. Determine:

(a) how much steam must be added in lbm_w/min to produce a saturated air condition, and

(b) the resulting temperature of the saturated air.

8.9 Moist air at 70 °F dry-bulb temperature and 45 percent relative humidity is recirculated from a room and mixed with outdoor air at 97 °F dry-bulb temperature and 83 °F thermodynamic wet-bulb temperature. Determine the mixture state dry-bulb and wet-bulb temperatures if the volume of recirculated air (ft³/min) is three times the volume of outdoor air. Assume sea-level pressure.

8.10 A condition exists where it is necessary to cool and dehumidify air from 80 °F db, 67 °F wb to 60 °F db and 54 °F wb.

(a) Discuss the feasibility of doing this in one process with a cooling coil. [*Hint:* Determine the apparatus dew-point temperature for the process.]

(b) Describe a practical method of achieving the required process, and sketch it on a psychrometric chart.

8.11 Moist air is heated by steam condensing inside the tubes of a heating coil, as shown by Fig. 8.23. Part of the air passes through the coil and part is bypassed around the coil. Barometric pressure is 14.696 psia. Determine

(a) the lbm_a/min which bypass the coil, and

(b) the heat added by the coil in Btu/hr.

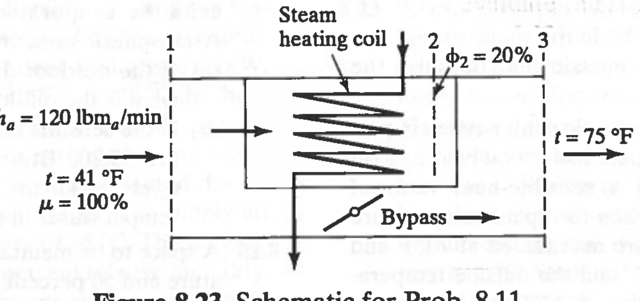


Figure 8.23 Schematic for Prob. 8.11.

8.12 Figure 8.24 schematically shows part of a winter-type air conditioning system. Barometric pressure is 14,696 psia. Determine:

(a) the temperature t_3 of the mixed air entering the heating coil, and

(b) the rate of heat addition to the air by the heating coil in Btu/hr.

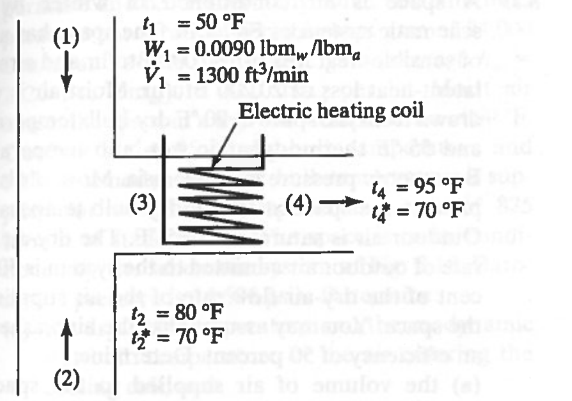


Figure 8.24 Schematic system for Prob. 8.12.

8.13 A heating/humidifying system with heat recovery is shown in Fig. 8.25. The heat-recovery unit transfers 80 percent of the heat that is removed from the exhaust air to the incoming outdoor air. The space is to be maintained at a dry-bulb temperature of 68 °F and a relative humidity of 40 percent. The space has a total load of 100,000 Btu/hr and a sensible-heat ratio 0.8. Air is supplied to the space at a dry-bulb temperature of 95 °F. The mass flow rate of air leaving the space is split such that 70 percent is recirculated and 30 percent is exhausted. Outdoor air is at a dry-bulb temperature of 40 °F and a relative humidity of 30 percent. As the exhaust air passes through the heat-recovery unit, its temperature drops to 48 °F. The heating and humidification occurs using a heating coil followed by a humidifier using saturated vapor at 240 °F.

(a) Clearly sketch and label all the points and processes on a psychrometric chart.

(b) Determine the rate at which heat is added by the heating coil, Btu/hr.

(c) Determine the rate at which moisture is added by the humidifier, lbm_w/hr.

8.14 Air at 50 °F, 40 percent relative humidity, and 13.5 psia pressure is heated in a furnace to 120 °F. The air is then supplied to a room in a building with a sensible-heat loss of 16,000 Btu/hr and a space-condition line having a sensible-heat ratio of 0.8. If the air in the room is 70 °F, determine:

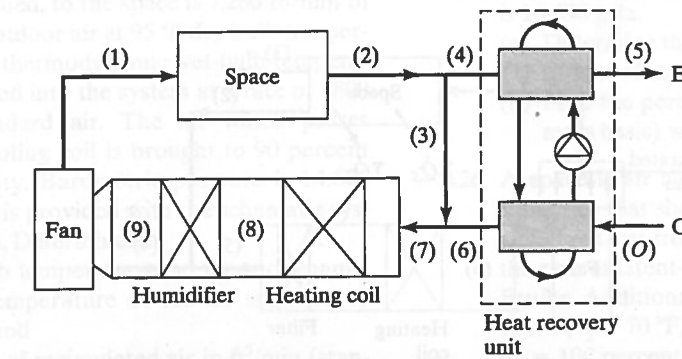


Figure 8.25 Schematic system for Prob. 8.13.

(a) the mass flow rate of dry air, and
(b) the humidity ratio, W, in the room.

Do not use the psychrometric chart to solve the problem.

8.15 Figure 8.26 shows a schematic winter-type air conditioning system. The space has a total heating load of 98,000 Btu/hr and a sensible-heat ratio of SHR = 0.6 on a day when the space temperature and relative humidity are maintained at 70 °F and 50 percent, respectively, and the outside temperature and relative humidity are 45 °F and 10 percent, respectively. The system uses 60 percent return air and 40 percent outside air (based on mass flow rates of dry air). Supply air is provided to the space at a rate of 15,000 lbm_a/hr. The saturated steam of the humidifier is at 250 °F.
(a) Carefully sketch and label all the points and process on a psychrometric chart.
(b) Calculate the rate of heat addition by the heating coil.
(c) Calculate the rate of moisture addition by the humidifier.

8.16 Air enters an air washer at a dry-bulb temperature of 100 °F and a wet-bulb temperature of 65 °F. The air exits the air washer at a dry-bulb temperature of 70 °F. Barometric pressure is 14.3 psia. *Without using a psychrometric chart*, answer the following:
(a) Calculate the relative humidity of the air *exiting* the air washer.
(b) If the mass flow rate of dry air through the air washer is 3000 lbm_a/hr, calculate the rate at which water is being added to the air stream.

8.17 An evaporative cooler (air washer) is used to cool a bus located in Phoenix. The system, similar to that shown in Fig. 8.18, takes outside air, first passes it through an evaporative cooler, then passes it through the bus, and finally exhausts it back to the outside. The system operates with a mass flow rate of 11,400 lbm_a/hr. On a very hot day the air exits the evaporative cooler saturated at 65 °F. Atmospheric pressure is 14.7 psia.
(a) If the outdoor dry-bulb temperature is 95 °F, what is the outdoor relative humidity.
(b) If the sensible and latent heat gains of the bus are 55,200 Btu/hr and 16,800 Btu/hr, respectively, what are the dry-bulb and dew-point temperatures in the bus?

8.18 A space to be maintained at 75 °F dry-bulb temperature and 50 percent relative humidity has a rate of sensible-heat gain of 82,000 Btu/hr and a rate of moisture gain (average h_w = 1100 Btu/lbm_w) of 12.0 lbm_w/hr. Barometric pressure is 14.696 psia. Moist air is supplied to the room at 58 °F dry-bulb temperature. Determine:
(a) the dew-point temperature and thermodynamic wet-bulb temperature of the supply air, and
(b) the volume of supply air required in ft³/min of standard air.

8.19 A space is air conditioned in winter by the schematic system of Fig. 8.14. The space has a rate of sensible-heat loss of 180,000 Btu/hr and a rate of latent-heat loss of 20,200 Btu/hr. Moist air is withdrawn from the space at 70 °F dry-bulb temperature and 55 °F thermodynamic wet-bulb temperature. Barometric pressure is 14.696 psia. Moist air is supplied to the space at 100 °F dry-bulb temperature. Outdoor air is saturated at 35 °F. The dry-air flow rate of outdoor air admitted to the system is 50 percent of the dry-air flow rate of the air supplied to the space. You may assume that the air washer has an efficiency of 50 percent. Determine:
(a) the volume of air supplied to the space in ft³/min of standard air,
(b) the spray-water temperature,
(c) the lbm/hr of makeup water required for the air washer, and
(d) the rate of heat added to the air by each heating coil in Btu/hr.

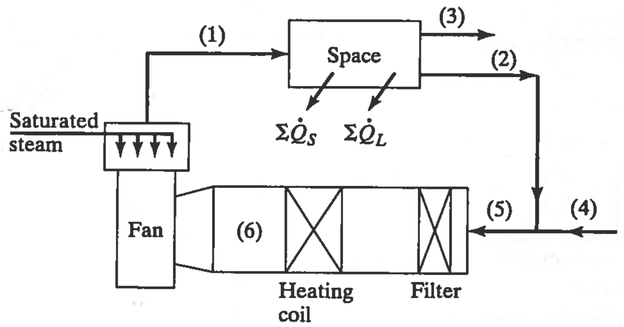


Figure 8.26 Schematic air conditioning system for Prob. 8.15.

8.20 A building is heated and humidified using the system shown in Fig. 8.14. The building has a sensible-heat loss of 260 kW and a latent-heat loss of 29 kW. The building space is maintained at 20 °C dry-bulb and 40 percent relative humidity. Outside air is at 3 °C dry-bulb and 30 percent relative humidity. Forty-five percent of the mass flow rate of dry air supplied to the space is outside air. The supply air is at a dry-bulb temperature of 35 °C. The air exiting the air washer is at 80 percent relative humidity.
(a) Sketch and label all points and processes on a psychrometric chart.
(b) Calculate the rate of heat addition to the moist air by the preheat coil.
(c) Calculate the rate of moisture addition by the air washer.
(d) Calculate the rate of heat addition to the moist air by the reheat coil.

8.21 A space to be maintained at 76 °F dry-bulb temperature and 65 °F thermodynamic wet-bulb temperature has a rate of sensible-heat gain of 84,000 Btu/hr and a rate of moisture gain (average h_w = 1120 Btu/lbm_w) of 20.0 lbm_w per hr. Moist air enters the space at a dry-bulb temperature of 60 °F. Outdoor air at 95 °F dry-bulb temperature and 78 °F thermodynamic wet-bulb temperature is supplied for ventilation purposes at a rate of 825 ft³/min of standard air. The space is to be conditioned by the schematic system of Fig. 8.15. Barometric pressure is 14.696 psia. Determine:
(a) the dry-bulb temperature and thermodynamic wet-bulb temperature of the air entering the cooling coil, and
(b) the tons of refrigeration required.

8.22 A space to be maintained at 80 °F dry-bulb temperature and 50 percent relative humidity has a rate of sensible-heat gain of 73,500 Btu/hr and a rate of latent-heat gain of 16,500 Btu/hr. The volume of air supplied, to the space is 7,200 ft³/min of standard air. Outdoor air at 95 °F dry-bulb temperature and 75 °F thermodynamic wet-bulb temperature is introduced into the system at a rate of 1800 ft³/min of standard air. The air which passes through the cooling coil is brought to 90 percent relative humidity. Barometric pressure is 14.696 psia. The space is provided with the schematic system of Fig. 8.16. Determine:
(a) the dry-bulb temperature and thermodynamic wet-bulb temperature of the air supplied to the space, and
(b) the volume of recirculated air in ft³/min (standard air) which should bypass the cooling coil.

8.23 A space to be maintained at 27 °C dry-bulb temperature has a rate of sensible-heat gain of 13kW and a rate of latent-heat gain of 8.5 kW. The system used to condition the space is shown in Fig. 8.16. The mass flow rate of air supplied to the space is 1.1 kg_a/s. Outdoor air at 38 °C dry-bulb temperature and 17 °C dew-point temperature is introduced into the system at a rate of 0.28 kg_a/s. The air exiting the cooling coil is saturated and at a temperature of 7 °C. Standard atmospheric pressure exists.
(a) Clearly sketch and label all points and process lines on a psychrometric chart.
(b) Determine the dry-bulb temperature and relative humidity of air supplied to the space.
(c) Determine the relative humidity in the space.
(d) Determine the refrigeration load.

8.24 A space to be maintained at 75 °F dry-bulb temperature and 65 °F thermodynamic wet-bulb temperature has a rate of sensible-heat gain of 377,000 Btu/hr and a rate of moisture gain (average h_w = 1100 Btu/lbm_w) of 330.0 lbm_w per hr. The chilled air leaves the cooling coil at 54 °F dry-bulb temperature and 90 percent relative humidity. Barometric pressure is 14.696 psia. The space is conditioned by the system shown in Fig. 8.17. Determine:
(a) the dry-bulb temperature and thermodynamic wet-bulb temperature of the air supplied to the space, and
(b) the rate of heat addition by the reheat coil in Btu/hr.

8.25 An interior space of a building is to be maintained at 72 °F during winter. The space has a rate of sensible-heat gain of 60,000 Btu/hr during working hours; moisture gain is negligible. The space is to be cooled by mixing outdoor air with recirculated air. Air is to be supplied to the space at a dry-bulb temperature of 55 °F. On a particular day, the outdoor air is saturated at −10 °F. Barometric pressure is 14.696 psia.
(a) Determine the volume of supply air required in ft³/min of standard air.
(b) Find the percentage of the supply air (dry-air mass basis) which is outdoor air.

8.26 A space is air conditioned in winter by a system similar to that shown by Fig. 8.19. The rate of sensible-heat loss from the space is 180,000 Btu/hr and the rate of latent-heat loss from the space is 19,700 Btu/hr. Additional known data are as follows: t_1 = 100 °F, t_2 = 70 °F, t_2^* = 55 °F, t_3 = 45 °F, t_4 = 20 °F, ϕ_4 = 100 percent, and ϕ_6 = 100 percent. Barometric pressure is 14.696 psia.

(a) Locate all state-points on the psychrometric chart and read values of t_1^*, t_5^*, t_5, t_2^*, and t_6.

(b) Determine the required specific enthalpy for the steam admitted to the air by the humidifier.

8.27 Figure 8.27 schematically depicts a two-zone variable-air-volume heating/humidifying system. The following conditions exist.

Space temperatures: t_2 = 78 °F, t_5 = 68 °F

Supply air: t_{12} = 95 °F, ϕ_{12} = 30 percent

Air-washer exit humidity: ϕ_{11} = 95 percent

Outside conditions: t_8 = 40 °F, ϕ_8 = 50 percent

Zone A heat losses, Btu/hr: 48,000 sensible; 32,000 latent

Zone B heat losses, Btu/hr: 24,000 sensible; 6,000 latent

Exhaust air: For both zones, the exhaust mass flow rate of dry air is 15 percent of the supply

Barometric pressure: 29.9 in. Hg

(a) Carefully sketch and label the points and processes on a psychrometric chart.

(b) Determine the rate of heat addition by the preheat coil, Btu/hr.

(c) Determine the rate of moisture addition by the air washer, lbm$_w$/hr.

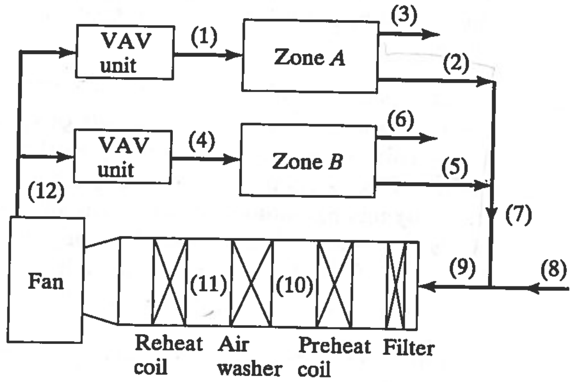


Figure 8.27 Schematic air conditioning system for Prob. 8.27.

8.28 Figure 8.28 depicts a two-zone heating/humidifying system. The following conditions exist.

Zone A is at 68 °F dry-bulb temperature and 50 percent relative humidity.

The mass flow rate of supply air into zone A is 6000 lbm$_a$/hr.

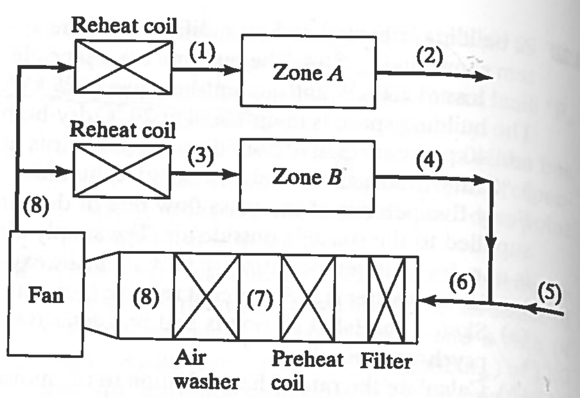


Figure 8.28 Schematic air conditioning system for Prob. 8.28.

The sensible-heat loss from zone A is 32,000 Btu/hr.

Zone B is at 72 °F dry-bulb temperature and 40 percent relative humidity.

The mass flow rate of supply air into zone B is 4000 lbm$_a$/hr.

The sensible- and latent-heat losses from zone B result in an SHR = 0.7.

Outside air is at 35 °F dry-bulb temperature and 20 percent relative humidity.

The preheat coil is controlled so that the air leaves the coil at a dry-bulb temperature of 80 °F.

The air exits the air washer at saturated conditions.

The system operates at standard atmospheric pressure.

Carefully sketch and label all the points and process lines on a psychrometric chart and determine:

(a) the dry-bulb temperature and relative humidity of state 6,

(b) the rate of heat input by the preheat coil,

(c) the dry-bulb and wet-bulb temperatures of state 8,

(d) the rate of moisture added to the air by the air washer,

(e) the dry-bulb temperature and relative humidity of the supply for zone A, and

(f) the rate of heat input by the reheat coil for zone B.

8.29 A dual-duct system similar to the one shown in Fig. 8.22 is used to air condition a building. On a day when the outdoor dry-bulb temperature is 60 °F and the outdoor relative humidity is 40 percent, zone A requires cooling while zone B requires heating. The following conditions exist.

Zone A	*Zone B*
70 °F dry-bulb temp.	72 °F dry-bulb temp.
60% relative humidity	40% relative humidity
20,000 Btu/hr sensible-heat *gain*	25,000 Btu/hr sensible-heat *loss*
16,400 Btu/hr latent-heat *gain*	Negligible latent-heat *loss*

The system operates with 50 percent exhaust and 50 percent return air flows for each zone. The air in the hot deck has a dry-bulb temperature of 95 °F and a relative humidity of 20 percent, and the air in the cold deck is saturated at 40 °F. Standard atmospheric pressure exists. Determine:

(a) the supply-airflow rates for each zone,

(b) the supply-air conditions for each zone (temperature and relative humidity), and

(c) the mass flow rate of air from the hot deck used in conditioning zone A.

8.30 Figure 8.22 shows a schematic of a dual-duct system. For design purposes, suppose the zones shown are two of five zones, each having identical operating design conditions. The only exhaust is the main system exhaust. The zones are to be maintained at a dry-bulb temperature of 75 °F and 50 percent relative humidity when the total heat gain of each zone is 200,000 Btu/hr and the sensible-heat ratio is 0.60. Outdoor air conditions are 95 °F dry-bulb temperature and 40 percent relative humidity. The system is designed to operate with a mass flow rate of dry air that consists of 25 percent outside air and 75 percent return air. The hot deck provides sensible heating only, and the air leaves the heating coil at a dry-bulb temperature of 105 °F. The cold deck is designed such that air leaves the cooling coil at a dry-bulb temperature of 50 °F and 90 percent relative humidity. Barometric pressure is standard atmospheric pressure.

For the above design conditions, carefully sketch and label all the points and processes on a psychrometric chart and determine:

(a) the mass flow rate, lbm$_a$/hr, through the heating coil,

(b) the heating-coil capacity, Btu/hr,

(c) the mass flow rate, lbm$_a$/hr, through the cooling coil, and

(d) the cooling-coil capacity, Btu/hr.

SYMBOLS

b	Bypass factor.
c_p	Specific heat of moist air at constant pressure, Btu/lbm$_a$ · °F or kJ/kg$_a$ · °C.
$\bar{c}_p$	Average specific heat of moist air at constant pressure for a process, Btu/lbm$_a$ · °F or kJ/kg$_a$ · °C.
c_{pa}	Specific heat of dry air at constant pressure, Btu/lbm$_a$ · °F or kJ/kg$_a$ · °C.
c_{pw}	Specific heat of water vapor at constant pressure, Btu/lbm$_w$ · °F or kJ/kg$_w$ · °C.
e_c	Saturation effectiveness for an evaporative cooler.
H	Enthalpy, Btu or kJ; H_a for dry air; H_w for water or water vapor.
h	Specific enthalpy of moist air, Btu/lbm$_a$ or kJ/kg$_a$; h_a for dry air; h_s for saturated moist air at t; $h_{as} = h_s - h_a$.
h_f	Specific enthalpy of liquid water, Btu/lbm$_w$ or kJ/kg$_w$.
h_g	Specific enthalpy of saturated water vapor, Btu/lbm$_w$ or kJ/kg$_w$.
$\bar{h}_g$	Average enthalpy of water vapor defined by Eq. (7.44), Btu/lbm$_w$ or kJ/kg$_w$.
h_{fg}^*	$h_g^* - h_f^*$, Btu/lbm$_w$ or kJ/kg$_w$.
h_{g0}	Enthalpy of saturated water vapor at 0 °F or 0 °C, Btu/lbm$_w$ or kJ/kg$_w$.
Δh_L	Latent change of specific enthalpy of moist air for a process, Btu/lbm$_a$ or kJ/kg$_a$.
Δh_S	Sensible change of specific enthalpy of moist air for a process, Btu/lbm$_a$ or kJ/kg$_a$.
h_w	Enthalpy of water added to moist air, Btu/lbm$_w$ or kJ/kg$_w$; $h_w = h_g$ for low-pressure water vapor.
$\dot{m}$	Mass flow rate, lbm/hr or kg/s.