Calculation of the adiabatic saturation temperature, T_{as}: We may make an estimate of T_{as} from Eq. (10.9) taking the relevant physical properties at the given air temperature (30°C) and then repeat the calculations taking the properties at the mean gas-film temperature. Alternatively, we can proceed with a guess value of T_{as} and the properties at the mean gas-film temperature. If the calculated value is not much different from the guess value, the second cycle of calculation to obtain a refined value may not be necessary. We follow this procedure here.

Take a guess value, T_{as} = 3°C. Mean temperature, T_{av} = (30 + 3)/2 = 16.5°C

$$\text{Working equation, Eq. (10.9): } T_G - T_{as} = (Y'_{as} - Y')\frac{\lambda_{as}}{c_H} \qquad \text{(i)}$$

If we know the values of λ_{as} and c_H, we can solve the above equation together with the vapour pressure equation to calculate T_{as}. We may get the value of λ_{as} at the guess temperature of 3°C following the method given in Reid et al. (1988) and used above. λ_{as} = 985 kJ/kg.

Determination of c_H: We take the value of 'humid heat' c_H at the mean humidity of air. Here 'humidity' means kg alcohol vapour per kg dry air. The vapour pressure of alcohol at the guess value of the liquid temperature (3°C) is calculated from the vapour pressure equation, P^v = 0.01943 bar.

Corresponding humidity = $Y_s' = [P^v/(P - P^v)] \times (46/28.97)$ = 0.03105 (kg ethanol)/(kg dry air)

'Humidity' in the bulk air = Y' = 0; Mean 'humidity' = 0.01552

Specific heat of alcohol vapour at the mean temperature (291.2 K) may be calculated using the relation given by Reid et al. (1988), Appendix A, p. 677.

$$\begin{aligned} c_p &= 9.014 + 0.2141T - 8.39 \times 10^{-5}T^2 \\ &= 9.014 + (0.2141)(291.2) - 8.39 \times 10^{-5}(291.2)^2 = 64.25 \text{ kJ/kmol·K} \\ &= 1.4 \text{ kJ/(kg alcohol)(K)} \end{aligned}$$

Specific heat of dry air at the mean temperature = 1.007 kJ/kg·K

'Humid heat' (at Y' = 0.01552), c_H = 1.007 + (0.0192)(1.4) = 1.034 kJ/(kg dry air).
Now putting all the values in Eq. (i), we have

$$303.2 - T_{as} = \frac{985}{1.034} \times \frac{p_{as}}{P - p_{as}} \times \frac{46}{28.97} = (1513)\frac{p_{as}}{P - p_{as}}$$

Solving the above equation together with the vapour pressure equation for alcohol (by trial or numerically), we obtain

$$T_{as} = \boxed{275.2 \text{ K} = 2°\text{C}}$$

This value is pretty close to the guess value of 3°C. No further calculation is necessary.

10.5 DESCRIPTION OF COOLING TOWERS—CONSTRUCTION AND OPERATION

A cooling tower is a special type of *heat exchanger* in which the warm water and the air are brought in direct contact for evaporative cooling. It must provide a very good contact of air and water in terms of the contact area and mass transfer coefficient of water vapour while keeping the air pressure drop low. In the early years of industrial development, cooling of warm water for reuse was done in 'spray ponds'. In a spray pond, as the name implies, a spray system located about six to eight feet above the water surface creates small droplets of warm water that cool down by evaporation in contact with air. This is a simple and easy process but requires a large pond area. It is inefficient (the effective heat transfer coefficient is about 3.5 $\text{Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F}$ or 20 $\text{W/m}^2\cdot\text{K}$), and creates the problem of entrainment and carryover of water droplets by air. The first water cooling tower was built in the USA by George Stocker in 1898. It was a wooden packed tower. Many changes in the design and construction of cooling towers have occurred since then. A brief history of development of cooling towers has been given by Willa (1997).

The important factors that govern the operation of a cooling tower are: (i) the dry-bulb and wet-bulb temperatures of the air; (ii) the temperature of warm water; (iii) the efficiency of contact between air and water in terms of the volumetric mass transfer coefficient ($k_{Y'}\bar{a}$, see Section 10.7) and the contact time between the air and the water; (iv) the uniformity of distribution of the phases within the tower; (v) the air pressure drop; and (vi) the desired temperature of the cooled water. A tower is irrigated at the top through nozzles. A large air–water contact area and contact time are offered by a typical tower packing. Air may enter the tower driven by a density gradient (natural draft), may be pushed into the tower (forced draft) at the base or sucked into the tower (induced draft) assisted by a fan. Several types of cooling towers have been designed on the basis of the above factors and operating strategies. A number of models of each type are offered by the manufacturers. The more important types are classified (Figure 10.7) and described below.

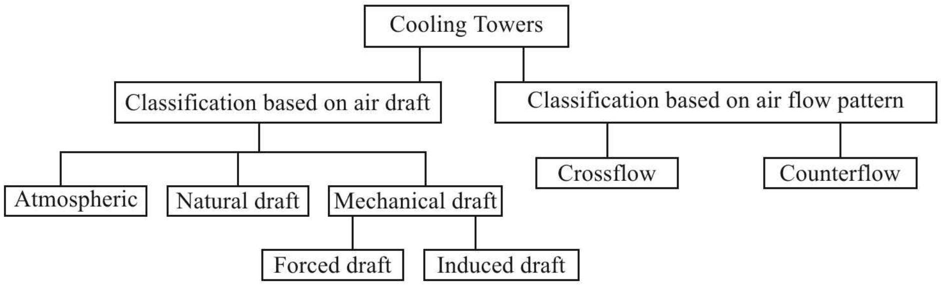

Figure 10.7 Classification of cooling towers.

10.5.1 Atmospheric Towers

An atmospheric tower consists of a big rectangular chamber with two opposite *louvered* walls. The tower is packed with a suitable *tower fill* (tower fills and louvers will be described later). Atmospheric air enters the tower through the louvers driven by its own velocity. An atmospheric tower is cheap but inefficient. Its performance largely depends upon the direction and velocity of wind.

10.5.2 Natural Draft Towers

A natural draft (also spelled 'draught') cooling tower (Figure 10.8) has a large reinforced concrete shell of hyperbolic shape (this is why such a tower is also called 'hyperbolic tower').

A small part of it near the bottom is filled with a high-void packing consisting of inclined wood or PVC battens of suitable size (1″ × 2″ cross-section is common) fitted into slots along the supporting frames. Alternative layers of battens run in perpendicular directions. The warm water splashes to form droplets as it falls from one batten to the next lower. Evaporation, and the resulting cooling of water, occurs at the water-film on the battens as well as on the surface of a droplet. A few layers of batten placed above the water distributor act as the *drift eliminator*, which arrest fine water droplets (like the *dry packing* above the distributor in a packed tower) that tend to be carried away by the up-flowing air.

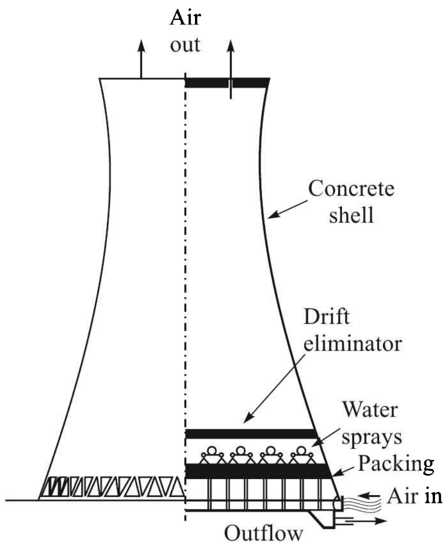

Figure 10.8 Counterflow natural draft tower.

A natural draft tower is so called because natural flow of air occurs through the tower. Two factors are responsible for creating the natural draft: (i) a rise in temperature and humidity of air in the column reduces its density and (ii) the wind velocity at the tower bottom. The pressure drop across the tower is low (a fraction of an inch of water) and the air velocity above the packing may vary from 1–1.5 m/s. The concrete tower is supported on a set of reinforced concrete columns. Air enters the tower at the bottom through the large openings in between the supporting columns and moves *countercurrent* to water. In some cases, a few fans are installed at the bottom to enhance the air flow rate. This type of tower is called 'fan-assisted' natural draft tower.

Natural draft towers are used in big installations like steam power plants requiring a huge quantity of cooling water. The diameter may be as large as 100 m and the height about 150 m and the capacity may be as high as 500,000 gpm (gallons per minute) water. The hyperbolic shape is made because of the following reasons: (i) more packing can be fitted in the bigger area at the bottom of the shell; (ii) the entering air gets smoothly directed towards the centre because of the shape of the wall, producing a strong upward draft; and (iii) greater structural strength and stability of the shell is provided by this shape. The initial cost of a hyperbolic tower is high. However, the absence of a fan reduces the fixed cost and particularly the operating and maintenance costs making it economic over a long period of time.

10.5.3 Mechanical Draft Towers

Because of their huge shape, construction difficulties and cost, natural draft towers have been replaced by mechanical draft towers in many installations. Compact and relatively small size units of rectangular shape are now available and preferred in process industries and petroleum refineries. A mechanical draft tower uses fans to move the air through the tower. The two types of such towers are: (i) forced draft towers, and (ii) induced draft towers.

Forced draft towers

It has one or more fans located at the tower bottom to push air into the tower (Figure 10.9). This is why it is called *forced draft*. The air flows counter-current to water. The advantages of the forced draft system are the following: (i) A part of the velocity head of air thrown by the blower is converted to pressure head on entering into the tower. This makes the forced draft tower slightly more energy efficient than the induced draft type. (ii) The system is less susceptible to vibrations because the fans are installed near the ground on a solid foundation. The disadvantages of the forced draft system are: (i) the air flow through the packing or *fill* may not be uniform, particularly if the tower is big and a big fan is used; (ii) some of the warm and humid air may be recirculated back into the tower thereby reducing the performance (see Section 10.6.1). The recirculation rate becomes low if the wind velocity is high. Forced draft towers are not very popular except for small capacities.

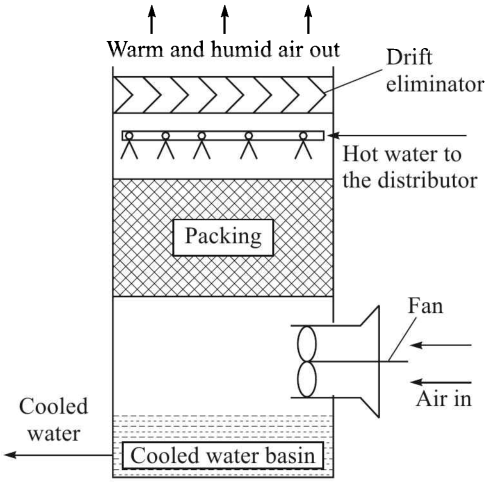

Figure 10.9 Schematic of a forced draft cooling tower.

Induced draft towers

An induced draft tower has one or more fans located at the top. The fans suck fresh air through the air inlets at the bottom. The air flow or draft is 'induced' by the suction created by the fans and hence the name. The induced draft tower may be of two types—crossflow and counterflow—depending upon the air inlet and flow pattern. In a 'counterflow induced draft tower' (Figure 10.10), a vertical movement of air countercurrent to the warm water occurs. The major advantage of this flow configuration is that the relatively dry air contacts the coldest water at the bottom, and the humid air contacts the warm water at the top of the tower. This ensures maximum average driving force for both heat and mass transfer. But more fan horsepower is required because of the restricted area of air flow at the bottom and, unlike the forced draft system, the velocity head of the air thrown out of the tower by the fan is dissipated.

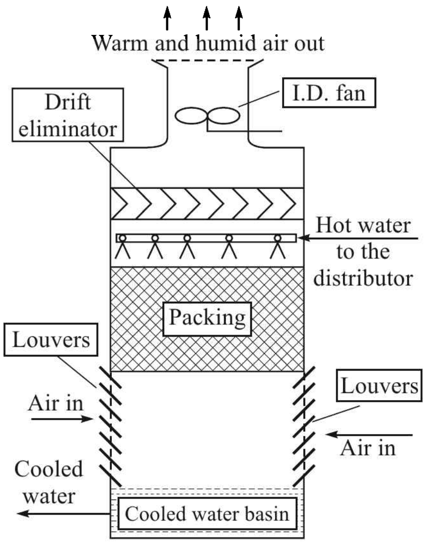

Figure 10.10 Schematic of a counterflow induced draft cooling tower.

The crossflow induced draft cooling tower provides horizontal air flow along the packed height (Figure 10.11). The air flow is crosscurrent to the down-flowing water. Louvers are provided all along the walls of the tower to allow the air to move in. For the same air flow rate, the tower requires less motor horsepower than the counterflow type. But the growth of algae on the fills is more because the tower interior gets more sunlight through the larger number of louvers.

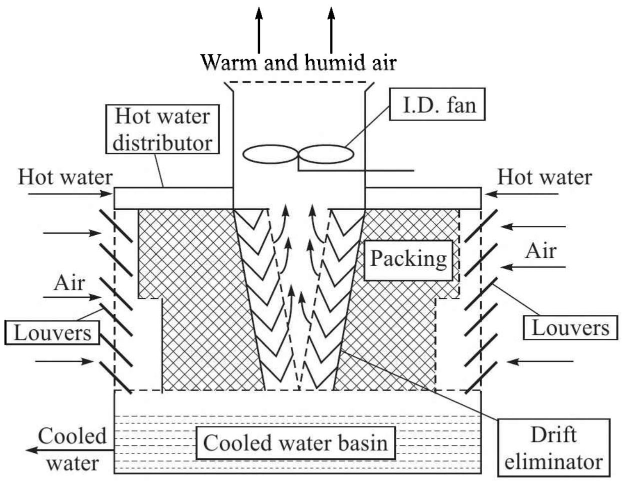

Figure 10.11 Schematic of a crossflow induced draft cooling tower.

The induced draft counterflow towers are nowadays built in compact cellular modules (Figure 10.12). Additional 'cells' may be added to increase the capacity if necessary. Each cell is provided with a fan, water distributor, and accessories. This is a big advantage of this type of towers. However, liquid distribution, in a crossflow tower is done by gravity flow through a hot water basin above the packing (see Figure 10.11) which may be easily cleaned. A counterflow tower needs distribution under pressure through spray nozzles which are not so easy to clean.

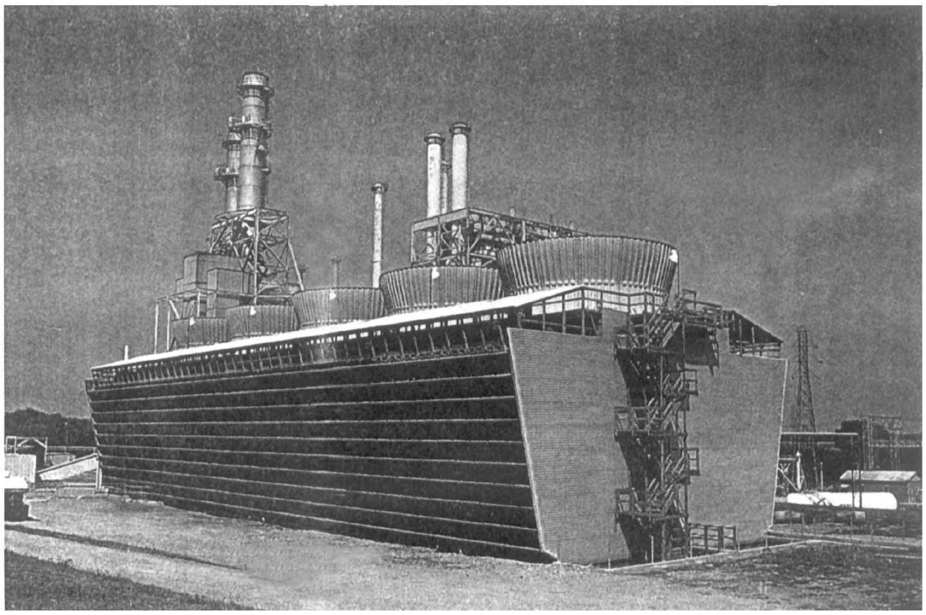

Figure 10.12 A five-cell crossflow induced draft tower (Courtesy: Marley Cooling Tower).

Spray-filled towers

This is the simplest but the least efficient induced draft cooling tower. It does not have any 'fill' or packing. Water is sprayed at the top of a tall empty chamber. The fan at the top induces air flow through the tower countercurrent to the falling spray.

10.5.4 Structural Components and Materials of Construction

The major structural components of a cooling tower (Hensley, 1985) are: (i) the shell or the framework and casing walls, (ii) the tower fill, (iii) the louvers, (iv) the drift eliminator, (v) the water distributor, (vi) the cold water basin, (vii) the fan deck and fan cylinders, and (viii) mechanical equipment supports. Some of these components and their materials of construction are briefly described here.

The shell, the framework and casing walls

A natural draft tower has a hyperbolic reinforced concrete shell. A forced draft tower of rectangular shape has a framework made of concrete or wood which is enclosed by casing walls. Treated wood has been the material for the casing walls for many decades. Chromated copper arsenate (CCA) has been used for treating wood to protect it against bacterial degradation. Since arsenic is a severe pollutant, acid copper chromate (ACC) containing the salts of chromium, chromic acid and copper is now the preferred agent for wood treatment. These chemicals diffuse into the wood during treatment in a pressure vessel and are retained in it by weak complexation. Wood is relatively insensitive to chlorides and sulphides, but may get slowly degraded by excessive free chlorine.

Polyvinyl chloride (PVC) corrugated sheets have been tried for building casing walls. But relatively high thermal expansion characteristics of PVC have limited its use. Cement asbestos board (CAB) is a good material for walls, but it is substantially heavy and sometimes susceptible to chemical attack. Further, asbestos is classified as a carcinogen and its use is not allowed in many countries. The most favoured material for casing walls and louvers at present is the corrugated glass-fibre reinforced polyester (FRP or GRP). It is lightweight (10–12 oz/ft^2). The opaque variety guards against the algal growth within the tower by preventing the entry of sunlight. Ultraviolet protective additives are often used to extend its life. The FRP made from chlorinated polyesters has the inherent flame retardant ability.

The tower fill

The *tower fill* is the single most important component of a cooling tower. A good fill should promote a high contact surface and contact time between air and water while imposing little resistance and pressure drop on the air flow. The *fills* are broadly of two types: (i) splash-type fill, and (ii) film-type fill.

The splash-type fills (Figure 10.13) are traditional. This type of fill consists of staggered rows of 'splash bars'. Wood battens or slats of suitable size (0.5–1″ thick, 3–4″ wide, and 4–6′ long) were the only material used over the decades. The battens are fitted into slots along a supporting frame. The splash bars covered with thin water-films intercept the falling droplets. On hitting a splash bar, the droplets break. The contact time is greatly increased because of repeated interception of the droplets by the splash bars. Beginning 1950s, the wood has been gradually replaced by plastics. The advantages of plastic materials are their resistance to microbial attack, corrosion and erosion, compatibility with other materials, formability, high strength-to-weight ratio, and comparatively low cost. Plastics are also least susceptible to changes in the water condition. The most commonly used plastics for use in tower fills (in the form of battens or V-bars) are PVC, polypropylene, and FRP. Thin concrete slabs (1.5–2″ thick, 3–4″ wide), coated with water-resistant paints, are also used.

Splash fills are used in both crossflow and counterflow towers. These are not prone to clogging, create only a small air pressure drop and allow easy cleaning. The splash bars must be horizontal; sagging of the bars causes 'channeling' of air and water.

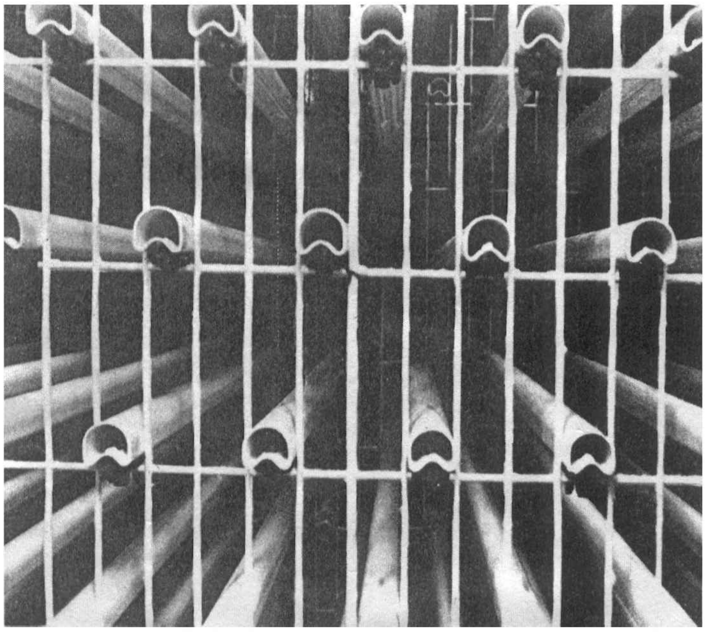

Figure 10.13 Splash-type fill—plastic splash bars (Courtesy: Marley Cooling Tower).

The film-type fill (also called *film-pack*; see Figure 10.14), which is similar to the structured packing (Section 5.11.2), consists of vertical corrugated sheets with a spacing of 18–25 mm. PVC is the most widely used material because it is inert and can be moulded easily into the desired shape. The sheets are glued to make rectangular units having sides about 1.5–2 m. The warm water spreads into thin films flowing along the fill surface. The air flows through the passage between the adjacent sheets.

The film-type fill is used in counterflow towers only and provides a higher cooling capacity for the same tower volume. Uniform spacing of the fill sheets is very important to ensure uniform distribution of air flow and to prevent 'channeling'. This type of tower fill is not recommended where the circulating cooling water contains foreign substances and debris.

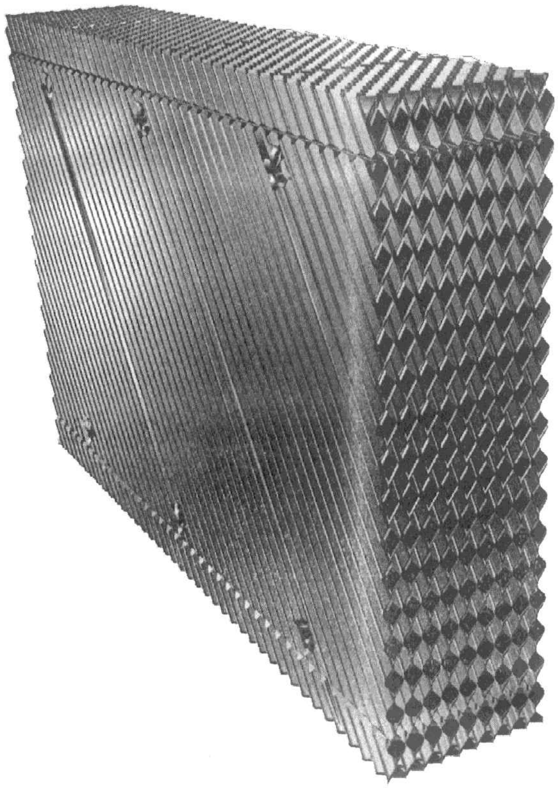

Figure 10.14 Marley MC 75 counterflow film pack.

Louvers

Louvers (see Figures 10.10 and 10.11) are inclined blade or passage type assemblies fitted at the air inlet wall of a cooling tower to promote uniform air entry into the tower while preventing water splashout. Crossflow towers are always provided with louvers to allow uniform air flow all along the wall. Closely spaced and steeply sloped louvers are very effective for water containment. The most important louver materials are corrugated fire-retardant fibre-reinforced polyester and treated wood. Asbestos cement board louvers are also in use, but the use of this material is declining.

Drift eliminators

Water droplets carried over by the outgoing air are collectively called *drift*. It is like entrainment of liquid in the gas in a conventional packed tower. Drift causes loss of cooling water. It is also a potential nuisance in the region close to a cooling tower. Drift is more likely to occur in a tower using splash fills. A drift eliminator arrests the floating water droplets by *impaction* as the air passes through it. A few layers of slats placed in the frame above the water distributor may act as the drift eliminator. A 'honeycomb' type drift eliminator made from a plastic material (predominantly

PVC) with labyrinth passages for air is used by Marley Cooling Towers. By using an effective drift eliminator, the drift level can be reduced to 0.02% of the water flow or even less.

Fans

Cooling tower fans should deliver large volumes of air efficiently. The fan blades should be properly balanced to minimize vibration. Propeller type fans are generally used; centrifugal fans are used in rather small installations. The fan diameter ranges from less than a metre to as large as 10 metre. Larger fans are equipped with 'adjustable pitch blades' to regulate the air flow rate as per requirement in order to minimize wastage of electric power if there is a change in the cooling load. A drive shaft transmits power from the motor through a gear box. The fan rpm typically ranges between 150 and 400; the tip speed of the fan blade may be as high as 4000 m/min. FRP is the common material for fan blades. In induced draft towers the air thrown by the fan leaves through a tall 'fan stack' or 'fan cylinder' (Figure 10.15) of a smooth profile to ensure 'velocity recovery'.

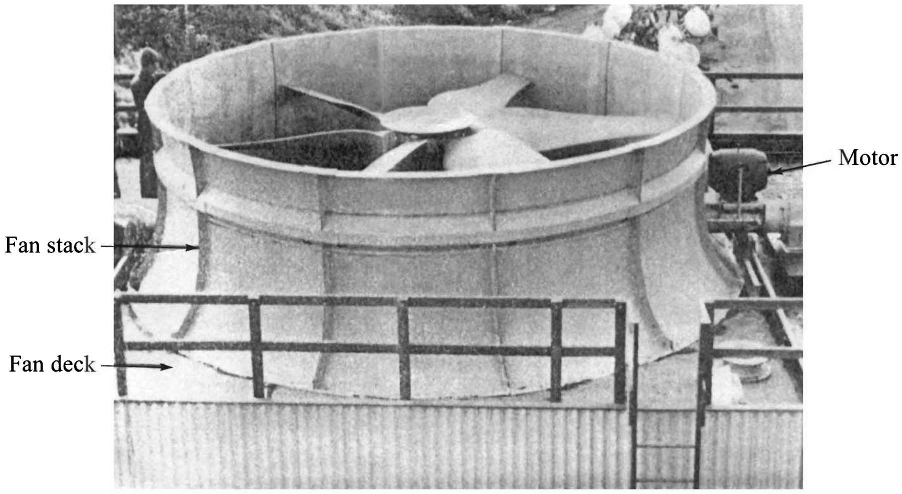

Figure 10.15 Typical fibre-reinforced plastic fan cylinder (Courtesy: Marley Cooling Tower).

Water distribution

Water enters through a main header and is supplied to the individual compartments or cells of a mechanical draft tower. A spray distributor is common in counterflow towers. A crossflow tower uses an open distribution deck.

10.5.5 Interference and Recirculation

A number of cooling towers may have to be used if the hot water load is large. The towers are very often installed at one place in the plant for convenience. Interference and recirculation are two common problems with cooling towers placed in proximity. Some of the moist air leaving a tower may be sucked into another tower installed 'downwind' in the vicinity of the former [Figure 10.16(a)] affecting the performance. This is called 'interference'. Sometimes a part of the moist air from a tower may be sucked into itself by the 'forced-draft fan'. This is 'recirculation'

[Figure 10.16(b)]. The ratio of the 'plume velocity' to the 'wind velocity' is an important factor to determine the extent of recirculation. If this ratio is small, a forced-draft tower becomes more susceptible to recirculation [Figure 10.16(c)]. Both the phenomena of interference and recirculation reduce the enthalpy driving for cooling and adversely affect the performance of cooling towers.

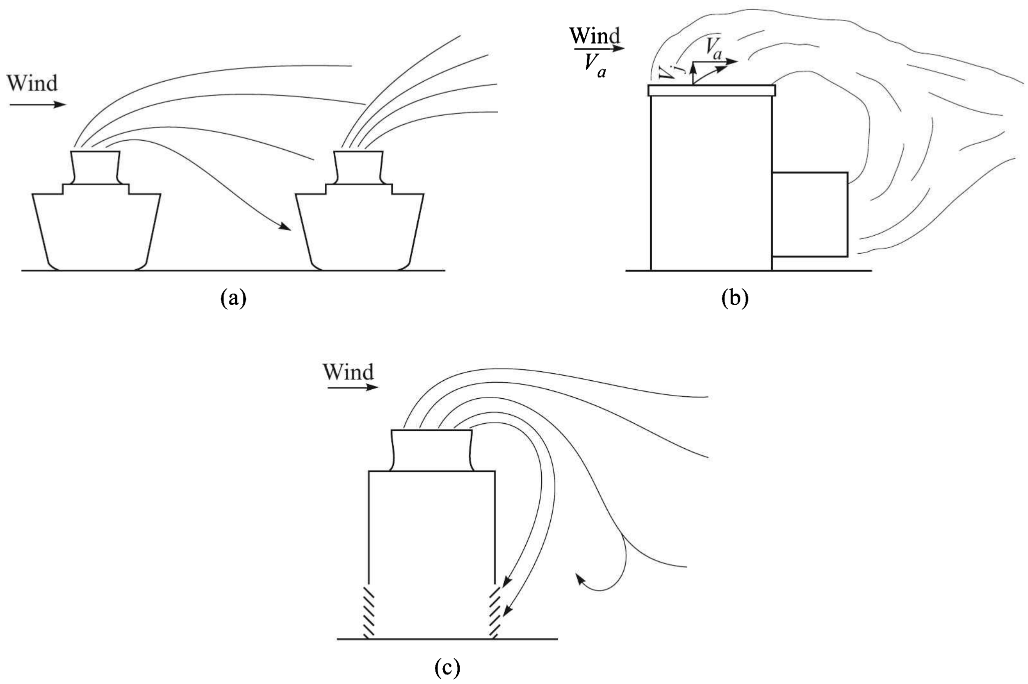

Figure. 10.16 (a) Interference; (b) and (c) recirculation.

10.5.6 Cooling Range and Approach

The difference in the temperatures of the inlet hot water and the outlet cooled water is called the 'range of cooling' or simply 'range'. The difference between the cooled water temperature leaving a tower and the wet-bulb temperature of the inlet air is the 'approach'. For example, if hot water is cooled from 40°C to 30°C by air of wet-bulb temperature 27°C, the range is 10°C and the approach is 3°C. If a small 'approach' is targeted, the height of packing increases rapidly. Theoretically, the approach is zero if a tower has an 'infinite' packed height. The range and approach are illustrated in Figure 10.17.

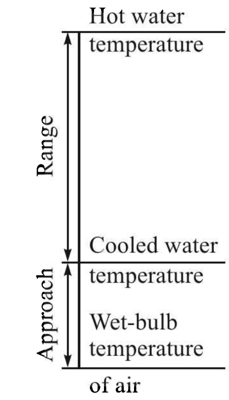

Figure 10.17 Cooling 'range' and 'approach' illustrated.

10.6 COOLING TOWER CALCULATIONS

An air–water contacting tower may have three major applications: (i) evaporative cooling of warm water from the cooling water circuit in a plant, (ii) cooling of a hot gas by direct contact with cool water, and (iii) dehumidification of a warm moist gas in contact with cool water. The design procedures are described below.

10.6.1 Sizing of a Cooling Tower

For sizing a cooling tower we primarily need to calculate: (i) the tower cross-section required to take the given load of warm water, and (ii) the height of packing required to achieve the desired cooling of the water. Simultaneous heat and mass transfer are involved. We have to write down the mass and enthalpy balance equations for the air and the water streams over a thin section of the tower and integrate these equations for the calculation of the required height of packing. The following analysis is based on a few assumptions: (i) The rate of vaporization of water is much less than the rate of water input to the tower (typically about 1% loss of the feed water occurs due to vaporization, hence the water flow rate within the tower remains essentially constant); and (ii) adiabatic cooling of water occurs in the tower (there is no heat gain or loss at the tower wall).

Let L (kg/m^2·s) be the 'constant' rate of water flow and G_s be the air flow rate [expressed in (kg dry air)/m^2·s].† The temperature of water decreases by dT_L and the enthalpy of the air increases by dH' across a differential thickness dz of the bed. Then the change in the enthalpy of the water = $Lc_{wL}dT_L$, and the change in the enthalpy of the gas = $G_s dH'$. The differential enthalpy balance over a thin slice of the bed in Figure 10.18 is given by

$$Lc_{wL}dT_L = G_s dH' \tag{10.15}$$

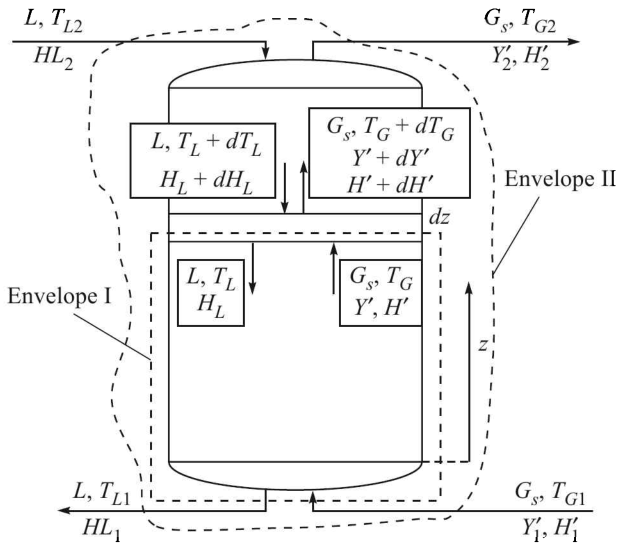

Figure 10.18 Mass and enthalpy balance in a cooling tower.

The enthalpy balance over the envelope I gives

$$Lc_{wL}(T_L - T_{L1}) = G_s(H' - H'_1) \tag{10.16}$$

† The flow rates L and G_s are on unit area basis, no prime (′) is used.

Equation (10.16) is the *operating line* for the air–water contact. Since L, c_{wL} and G_s remain unchanged along the tower, it is a straight line on the T_L–H plane having a slope of Lc_{wL}/G_s. Also making an enthalpy balance over the entire tower, we get the following relation

$$Lc_{wL}(T_{L2} - T_{L1}) = G_s(H'_2 - H'_1) \tag{10.17}$$

It appears from Eqs. (10.16) and (10.17) that the operating line may the obtained by joining the terminal points (T_{L1}, H'_1) and (T_{L2}, H'_2). The 'equilibrium curve' for the air–water system on the T_L–H' plane is the plot of enthalpy of saturated air versus the liquid temperature at equilibrium. As in other heterogeneous contacting, we assume that equilibrium prevails at the gas–liquid interface, i.e. the temperature and enthalpy of air at the air–water interface (T_{Li}, H'_i) lie on the equilibrium line. Typical operating and equilibrium curves are shown in Figure 10.19.

The rate of transfer of water vapour to air in the differential volume is

$$G_s dY' = k_{Y'}\bar{a}\, dz(Y'_i - Y') \tag{10.18}$$

where

$k_{Y'}$ = mass transfer coefficient of water vapour
$\bar{a}$ = specific interfacial area of the air–water contact
Y_i' = humidity of air at the air–liquid interface.

We shall use two other relations in the following analysis.

The decrease in the temperature of the gas because of sensible heat transfer to the water is given by

$$-G_s c_H dT_G = h_G\bar{a}\, dz(T_G - T_i) \tag{10.19}$$

The differential form of Eq. (10.7) is multiplied by G_s and then Eqs. (10.18) and (10.19) are used to yield

$$G_s dH' = G_s\lambda_0 dY' + G_s c_H dT_G = G_s\lambda_0 dY' - h_G\bar{a}\, dz(T_G - T_i)$$

$$= -h_G\bar{a}\, dz(T_G - T_i) + \lambda_0 k_{Y'}\bar{a}\, dz(Y_i' - Y') \tag{10.20}$$

$$\Rightarrow \quad G_s dH' = k_{Y'}\bar{a}\, dz[(h_G/k_{Y'})(T_i - T_G) + \lambda_0(Y_i' - Y')] \tag{10.21}$$

Using the result $(h_G/k_{Y'}) = c_H$ in the above equation,

$$G_s dH' = k_{Y'}\bar{a}\, dz[c_H(T_i - T_G) + \lambda_0(Y_i' - Y')]$$

$$= k_{Y'}\bar{a}\, dz[\{Y_i'\lambda_0 + c_H(T_i - T_0)\} - \{\lambda_0 Y' + c_H(T_G - T_0)\}]$$

Using the definition of enthalpy of the gas [see Eq. (10.7)],

$$G_s dH' = k_{Y'}\bar{a}\, dz(H_i' - H') \tag{10.22}$$

The above equation is unique in the sense that the change in the energy content of the gas over the differential thickness dz is expressed in terms of a transport coefficient and the 'enthalpy driving force'. The height of packing in the cooling tower can be obtained by integrating the above equation.

$$\int_{H'_1}^{H'_2} \frac{dH'}{H'_i - H'} = \frac{k_{Y'}\bar{a}}{G_s}\int_0^Z dz = \frac{k_{Y'}\bar{a}}{G_s} Z \tag{10.23}$$

Invoking the concept of transfer units, we can define

$$N_{tG} = \int_{H_1'}^{H_2'} \frac{dH'}{H_i' - H'} \quad \text{and} \quad H_{tG} = \frac{G_s}{k_{Y'}\bar{a}} \tag{10.24}$$

where

N_{tG} = number of gas-enthalpy transfer units
H_{tG} = height of a gas-enthalpy transfer unit.

Height of the packed section,

$$z = H_{tG}N_{tG} \tag{10.25}$$

The volumetric mass or enthalpy transfer coefficient ($k_{Y'}\bar{a}$) should be known or has to be estimated from a suitable correlation. So the height of a transfer unit may be calculated from the given gas mass flow rate. In order to determine the number of transfer units, the integral in Eq. (10.24) is to be evaluated. There is no direct relation available between the enthalpy of the bulk gas H' and that at the interface H_i'. So the integral cannot be evaluated analytically. For numerical or graphical evaluation of the integral, we have to know the values of H_i' (the enthalpy of air at the condition of the air–liquid interface or the 'interfacial enthalpy') for a set of values of H' within the range of enthalpy values given by the operating line ($H_1' \le H' \le H_2'$). The values of H_i' can be obtained by following a procedure similar to that for the determination of the interfacial gas-phase concentration of a solute described in Section 6.4.1.

If $h_L\bar{a}$ is the volumetric individual heat transfer coefficient on the water-side of the interface, a differential heat balance equation over a small section of the packing gives

$$Lc_{wL}dT_L = h_L\bar{a}\, dz(T_L - T_{Li}) \tag{10.26}$$

Combining Eqs. (10.15), (10.22) and (10.26),

$$k_{Y'}\bar{a}\, dz(H_i' - H') = -h_L\bar{a}\, dz(T_{Li} - T_L) \Rightarrow \frac{H_i' - H'}{T_{Li} - T_L} = -\frac{h_L}{k_{Y'}} \tag{10.27}$$

The above equation indicates that a line of slope $-h_L/k_{Y'}$ emanating from a point (T_L, H') on the operating line meets the equilibrium line at the point (T_{Li}, H_i') that represents the liquid temperature and air humidity at the gas–liquid interface. A very similar procedure was used in Section 6.4.1 for the determination of the interfacial concentrations of the gas and the liquid phases at a section of a gas absorption column if both the phases offer resistance to mass transfer. If a set of parallel lines of slope $-h_L/k_{Y'}$ are drawn from several points on the operating line, a set of values of (H', H_i') can be obtained. These values can be used to numerically or graphically evaluate the integral in Eq. (10.24). The value of the integral is the number of gas-phase transfer units, N_{tG}.

An alternative form of N_{tG} can be obtained by substituting $G_s dH' = Lc_{wL}dT_L$ [see Eq. (10.15)] in Eq. (10.22).

$$Lc_{wL}dT_L = k_{Y'}\bar{a}\, dz(H_i' - H') \quad \Rightarrow \quad \int_{T_{Li}}^{T_{Lo}} \frac{dT_L}{H_i' - H'} = \frac{k_{Y'}\bar{a}}{Lc_{wL}} \int_0^z dz = \frac{k_{Y'}\bar{a}}{Lc_{wL}} z \tag{10.28}$$

The above integral of the inverse of the enthalpy driving force also represents a type of transfer units. Kern (1950) preferred to call it the number of 'diffusion units' (NDU). The procedure of evaluation of the NDU is similar to N_{tG}.

A simplified design equation based on the overall enthalpy transfer coefficient

If an overall enthalpy transfer coefficient $K_{Y'}$ is used, the differential mass balance Eq. (10.22) becomes

$$G_s dH' = K_{Y'}\bar{a}\,dz(H'^* - H') \tag{10.29}$$

Here H'^* is the enthalpy of the saturated air at a temperature equal to the bulk liquid temperature, T_L.

On integration,

$$\int_{H_1'}^{H_2'} \frac{dH'}{H'^* - H'} = \frac{K_{Y'}\bar{a}}{G_s}\int_0^z dz = \frac{K_{Y'}\bar{a}}{G_s} z \tag{10.30}$$

The above integral is the number of overall gas-phase enthalpy transfer units.

$$\int_{H_1'}^{H_2'} \frac{dH'}{H'^* - H'} = N_{tOG}; \quad \frac{G_s}{K_{Y'}\bar{a}} = H_{tOG}; \quad z = N_{tOG}H_{tOG} \tag{10.31}$$

The above method is often called the 'Marked method' and the integral representing N_{tOG}, the 'Marked Integral'.

Expression of the overall enthalpy transfer coefficient in terms of the individual coefficients

If the individual gas-phase enthalpy transfer coefficient $k_{Y'}$ and the liquid-phase heat transfer coefficient h_L are known, these can be combined to express the overall coefficient (compare with Section 4.4).

Since T_L is the bulk water temperature and T_{Li} is the temperature at the air–water interface, the flux of energy transport from the liquid may be written as

$$q = k_{Y'}(H_i' - H') = h_L(T_L - T_{Li}) = K_{Y'}(H'^* - H') \tag{10.32}$$

Driving force: $$(H'^* - H') = (H'^* - H_i') + (H_i' - H') \tag{10.33}$$

From Eqs. (10.32) and (10.33),

$$\frac{q}{K_{Y'}} = q\frac{H'^* - H_i'}{h_L(T_L - T_{Li})} + \frac{q}{k_{Y'}} \Rightarrow \frac{1}{K_{Y'}} = \frac{H'^* - H_i'}{h_L(T_L - T_{Li})} + \frac{1}{k_{Y'}} \tag{10.34}$$

If the individual coefficients $k_{Y'}$ and h_L are known, the overall coefficient $K_{Y'}$ can be determined. The quantity $(H'^* - H')$ gives the overall enthalpy driving force for the water cooling process.

Step-by-step design procedure (also see Examples 10.3 and 10.4)

(a) Specify the inlet and outlet temperatures and the flow rate of warm water. These quantities are governed by the requirement of cooling water in the heat exchangers and other cooling devices in a plant.

(b) Select the design values of the dry-bulb and the wet-bulb temperature of air at the proposed geographical location of the cooling tower. The relevant data are available at the weather office in the region.

(c) Draw the 'equilibrium curve', i.e. the saturation–humidity curve. The enthalpy data can be calculated using the vapour pressure equation for water and the physical properties of air and water vapour [see Eq. (10.8)].

(d) Locate the lower terminal of the operating line Q on the T_L–H' plane by the point (T_{L1}, H_1'). This point indicates the condition at the bottom of the tower in the case of water cooling.

(e) Draw a tangent to the equilibrium line through the point Q. The slope of this tangent gives the ratio of the liquid and the minimum gas flow rates [see Eq. (10.16)]. The water rate is given. The minimum air rate is calculated. The actual air rate taken is usually 1.25 to 1.5 times the minimum. This step is not relevant if the air rate is given.

(f) The upper terminal of the operating line is located by the point $P(T_{L2}, H_2')$. It is the point where the operating line of the slope determined in step (e) meets the vertical line through T_{L2}. It can also be located by calculating the top end enthalpy H_2' from Eq. (10.17).

(g) Evaluate the integral in Eq. (10.24) or (10.28). This is the number of gas-phase enthalpy transfer units. The volumetric transfer coefficients $k_{Y'}\bar{a}$ and $h_L\bar{a}$ are required for this purpose. As described above, a set of parallel lines (sometimes called *tie lines*) of slope $-h_L\bar{a}/k_{Y'}\bar{a}$ is drawn between the operating line and the equilibrium line. The values of H' and H_i' are taken from the terminals of these lines. The integral is evaluated numerically or graphically.

(h) If the overall enthalpy transfer coefficient $K_{Y'}$ is known and used, the 'tie lines' are vertical. For a given value of H', the value of H'^* is given by the point on the equilibrium line vertically above it. The integral in Eq. (10.31) gives the number of overall transfer units in this case.

(i) The height of a transfer unit ($G_s/k_{Y'}\bar{a}$ or $G_s/K_{Y'}\bar{a}$) is calculated from Eq. (10.24) or (10.31). The packed height is the product of the height of a transfer unit and the number of transfer units.

The approximate packed heights of towers for different values of approach and cooling range are given below:

Approach to wet-bulb temperature (°F)	Cooling range (°F)	Packed height (ft)
15–20	25–35	15–20
10–15	25–35	25–30
5–10	25–35	35–40

EXAMPLE 10.3 (*Calculation of the height of packing, substantial liquid-phase heat transfer resistance*) Warm water at 45°C is to be cooled to 30°C by countercurrent contact with air in a tower packed with wood slats. The inlet air has a dry-bulb temperature of 31°C and a wet-bulb temperature of 22°C. The mass flow rate of water is 6000 kg/m^2·h and that of air is 1.4 times the minimum. The individual gas-phase mass transfer coefficient is $k_{Y'}\bar{a}$ = 6000 kg/(m^3)(h)($\Delta Y'$). The volumetric water-side heat transfer coefficient is given by $h_L\bar{a} = 0.059L^{0.51}G_s$, in kcal/m^3·h·K, where L and G_s are mass flow rates of water and air (dry basis).

Determine (a) the dry air flow rate to be used, (b) the height of packing, and (c) the wet-bulb depression at the bottom of the tower.

Solution

(a) Inlet air: $T_G = 31°C$; $T_w = 22°C = T_{as}$; humidity [from Figure 10.6(a)], $Y_1' = 0.01295$ kg/kg

$$\text{Enthalpy, } H_1' = 2500Y_1' + (1.005 + 1.88Y_1')(31 - 0) = 64.3 \text{ kJ/(kg dry air)};$$

$$\text{Exit water temperature, } T_{L1} = 30°C$$

Draw the saturation humidity curve (the equilibrium line) from the calculated values of saturation enthalpies at different temperatures (the procedure is illustrated in Example 10.1), as shown in Figure 10.19. For example, take $T_L = 35°C$ (= 308.2 K). Vapour pressure of water at this temperature, $P^v = 0.05625$ bar, and

$Y' = [(0.05625)/(1.013 - 0.05625)]\,[18.02/28.97] = 0.03657$ (kg moisture)/(kg dry air).

Enthalpy of saturated air at 35°C [ref. temp. = 0°C; see Eq. (10.8)]

$= (2500)(0.03657) + [1.005 + (1.88)(0.03657)]35$

$= 129$ kJ/(kg dry air)

Locate the point Q (30°C; 64.3 kJ/kg) on the T_L–H' plane (Q is the lower terminal of the operating line). In order to determine the minimum air rate, draw the tangent to the equilibrium line from the point Q (not shown in Figure 10.19).

$$\text{Slope of the tangent} = 10.76 = \frac{Lc_{wL}}{G_{s,\,\min}}$$

$$c_{wL} = 4.187 \text{ kJ/kg·K}$$

$$L = 6000 \text{ kg/m}^2\text{·h}$$

$\Rightarrow$ $$G_{s,\,\min} = 2335 \text{ kg/m}^2\text{·h}$$

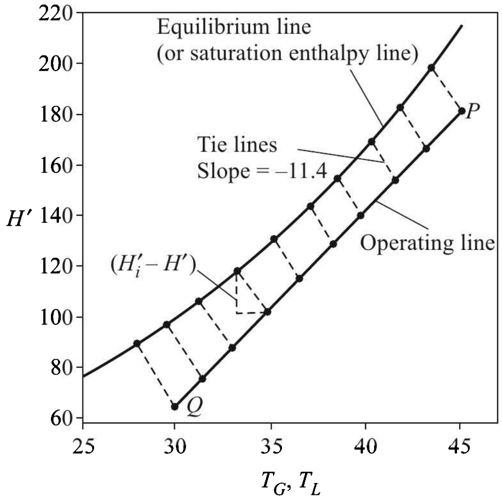

Figure 10.19 Equilibrium and operating lines.

Actual air rate to be used,

$$G_s = 1.4G_{s,\,\min} = (1.4)(2335)$$

$$= \boxed{3270 \text{ kg/m}^2 \cdot \text{h(dry basis)}}$$

(b) Given, feed water temperature, $T_{L2} = 45°C$. Determine H_2' (enthalpy of the exit air stream) from the equation of the operating line, Eq. (10.17).

$$Lc_{wL}(T_{L2} - T_{L1}) = G(H_2' - H_1') \Rightarrow (6000)(4.187)(45 - 30) = (3270)(H_2' - 64.3)$$

$\Rightarrow$ $$H_2' = 179.6 \text{ kJ/kg}$$

Locate the point P (45°C, 179.6 kJ/kg) which is the upper terminal of the operating line. Join PQ.

[The point P can be reached by an alternative method. Calculate the actual slope of the operating line (10.76/1.4 = 7.685 in this case). A line of this slope drawn through the point Q meets the vertical line through T_L = 45°C at the point P which is the upper terminal of the operating line.]

Calculate the liquid-phase heat transfer coefficient from the given correlation.

$$h_L\bar{a} = (0.059)(6000)^{0.51}(3270) = 16{,}300 \text{ kcal/h}\cdot\text{m}^3\cdot{}^\circ\text{C} = 68{,}260 \text{ kJ/h}\cdot\text{m}^3\cdot{}^\circ\text{C}$$

$$\text{Slope of the 'tie lines'} = -h_L\bar{a}/k_{Y'}\bar{a} = -68{,}250/6000 = -11.4$$

A set of tie lines of this slope (–11.4) is drawn from several points on the operating line (including the terminal points) as shown in Figure 10.19. Any such line meets the equilibrium line at (T_{Li}, H_i'). The set of points (H', H_i') thus obtained are given below. The values of $1/(H_i' - H')$ are plotted against T_L and the integral in Eq. (10.28) is evaluated graphically. The graphical integration is not shown here.

T_L	30	31.45	33	34.8	36.5	38.2	39.7	41.5	43.1	45
H'	64.3	75	87.2	101.2	114	127.5	138.7	152.6	165	179.6
T_{Li}	27.8	29.5	31.2	33.2	35.1	37	38.4	40.2	41.7	43.5
H_i'	89.7	96.7	105.7	118	130	142.5	153.5	168	181.1	196.7
$1/(H_i' - H')$	0.0394	0.0461	0.054	0.0595	0.0625	0.0667	0.0676	0.0641	0.0621	0.0585

$$\text{Area under the curve} = 6.76 = N_{tG} = \text{number of gas-phase transfer units}$$

$$\text{Height of a transfer unit, } H_{tG} = G_s/k_{Y'}\bar{a} = 3270/6000 = \boxed{0.545 \text{ m}}$$

$$\text{Packed height, } z = H_{tG}N_{tG} = (0.545)(6.76) = \boxed{3.68 \text{ m}}$$

(c) Wet-bulb depression at the bottom of the tower

= Air temperature – Wet-bulb temperature

$= 31 - 22 = \boxed{9^\circ\text{C}}$

10.6.2 Determination of the Gas Temperature Profile

In most cases, the temperature of the warm feed water (T_{w2}) to a cooling tower is well above that of the ambient air (T_{G1}). However, the temperature of the cooled water (T_{w1}) may be above or below T_{G1}. In either case, the heat supplied by the warm water is mostly utilized to vaporize water at the air–water interface. If $T_{G1} > T_{w1}$, a small part of the heat of vaporization comes from the air and its temperature decreases with height in the lower part of a cooling tower followed by a rise in contact with the warm water. However, if $T_{G1} < T_{w1}$, the air temperature increases continuously along the height.

The bulk liquid temperature at any axial position in the tower can be determined by a slight manipulation of Eq. (10.28). But the determination of the bulk gas temperature is not straightforward. It can be determined by the following graphical procedure.

Dividing Eq. (10.22) by Eq. (10.19),

$$-\frac{G_s\,dH'}{G_s\,c_H\,dT_G} = \frac{k_{Y'}\,\bar{a}\,dz(H_i' - H')}{h_G\,\bar{a}\,dz(T_G - T_i)}$$

Using the result $h_G/k_{Y'} = c_H$ for the air–water system,

$$\frac{dH'}{dT_G} = \frac{H_i' - H'}{T_G - T_i} \tag{10.35}$$

A step-by-step procedure for the determination of the gas temperature profile in a cooling tower using the above equation is given by Skelland (1974). This is illustrated in Example 10.4.

EXAMPLE 10.4 (*Determination of packed height, the overall coefficient being given*) A cooling tower receives warm water at 43°C at a mass flow rate of 7000 kg/m^2·h. A cooling range of 13°C is to be achieved by countercurrent contact with air. The air enters at a rate of 4200 kg/m^2·h at a dry-bulb temperature of 31°C and a humidity of 0.01516 kg/(kg dry air). The overall volumetric mass transfer coefficient is $K_{Y'}\bar{a}$ = 2500 kg/(m^3)(h)($\Delta Y'$).

(a) Determine the number of overall gas-phase transfer units and the height of packing.

(b) Plot the gas temperature profile along the tower.

Solution

(a) Dry-bulb temperature of air, T_G = 31°C

Humidity, Y_1' = 0.01516; enthalpy, H_1' = 70 kJ/(kg dry air), calculated; cooling range = 13°C

Water temperature: inlet, T_{L2} = 43°C; outlet, T_{L1} = 43 – 13 = 30°C

Water feed rate, L = 7000 kg/h·m^2

Total rate of air input = 4200 kg/h·m^2(on dry basis)

G_s = 4200/1.01516 = 4137 (kg dry air)/m^2·h

Heat balance [see Eq. (10.17)]: (7000)(4.187)(43 – 30) = (4137)(H_2' – 70)

$\Rightarrow$ H_2' = 162 kJ/(kg dry air)

Draw the 'equilibrium line' and draw the operating lines through the points Q(30, 70) and P(43, 162).

The integral in Eq. (10.31) is now evaluated graphically. A set of vertical lines (tie lines) are drawn between the operating line and the equilibrium line [Figure 10.20(a)]. A set of points (H', H'^*) obtained from the terminals of these lines are tabulated. Graphical integration is shown in Figure 10.20(b).

T_L	30	33	35	37	39	41	43
H'	70	84.5	99.6	115.2	130.2	145.5	162
H'^*	105	117	129	143	158.7	175.5	196
$1/[H'^* - H']$	0.0286	0.0308	0.034	0.036	0.0351	0.0333	0.0295

Area under the graph, $\displaystyle\int_{H_1'}^{H_2'} \frac{dH'}{H'^* - H'} = N_{tOG} = 3.04$; $H_{tOG} = \dfrac{G_s}{K_{Y'}\bar{a}} = \dfrac{4137}{2500} = 1.655$ m

Packed height, $z = (1.665)(3.04) = \boxed{5.06 \text{ m}}$

(b) The gas temperature profile may be obtained by discrete graphical construction on the H'–T_G,T_L plane following the principles described in Section 10.7.2. Refer to Figure 10.20(c).

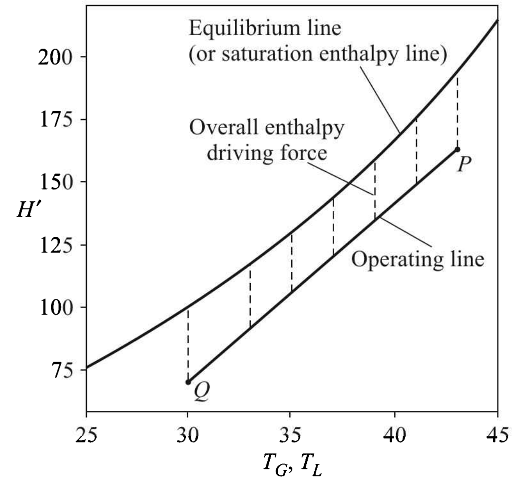

Figure 10.20(a) Equilibrium and operating lines.

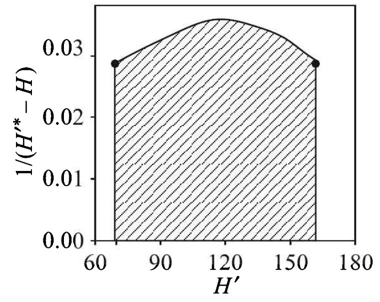

Figure 10.20(b) Graphical integration for N_{tOG}.

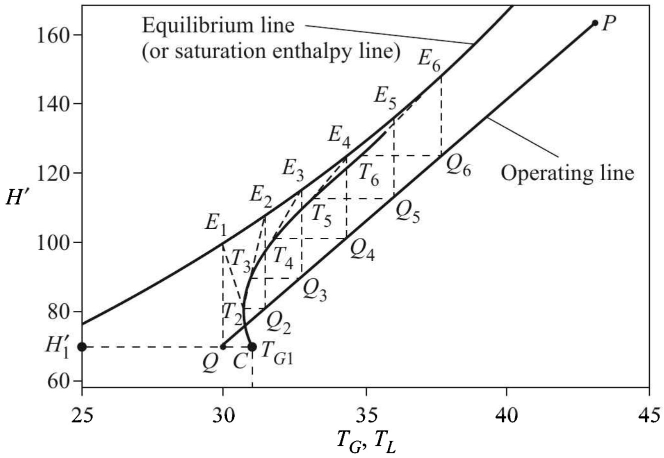

Figure 10.20(c) Gas temperature profile along the tower.

(i) The inlet state of the gas is given by the point C (H_1', T_{G1}). A vertical line through the end of the operating line, Q, meets the 'equilibrium line' at the point E_1 which denotes the interfacial condition at the bottom (*neglecting the liquid-phase resistance*).

(ii) Join CE_1. Equation (10.35) stipulates that the gas temperature in the 'film' follows the line CE_1.

(iii) Move a 'small' distance to the point T_2 along CE_1. Move horizontally to Q_2 on the operating line and then move vertically to E_2 on the equilibrium line; the point E_2 gives the interfacial condition when the bulk gas temperature is T_2. In the next step the bulk gas temperature changes along T_2E_2, and so on. Proceeding in this way, the gas temperature profile can be obtained as shown in Figure 10.20(c).

It may be noted that the temperature difference between the gas and the liquid decreases quickly along the bed, although the enthalpy driving force does not decrease much.

10.6.3 Dehumidifiaction and Cooling of a Hot Moist Gas

Dehumidification and cooling of a hot, moist gas stream can be conveniently done by direct contact with cool water in a packed tower. This is a practically useful technique adopted in many applications. Consider the case of drying of a moist solid prone to air-oxidation. Drying may be done in contact with a hot inert gas, say N_2. The gas (N_2) has value and cannot be discarded after use. It is recycled. For this purpose, the warm and moist gas from the dryer is needed to be dehumidified (otherwise it cannot be used again). This can be done in direct contact with cool water in a tower. The dehumidified gas is heated again (using a steam heater, for example) and recycled to the dryer.

There are many uses of air of controlled humidity typically in pharmaceutical industries and for the storage of hygroscopic materials (e.g. fertilizers like urea, ammonium nitrate, etc.). A cooling tower cannot supply dehumidified air for such cases. Dehumidification by chilling or by a desiccant is preferred in practice (Harrison and Simkins, 1997; Speltz, 1998).

The procedure of calculation of packed height is very similar to that of water cooling described in the previous section. The operating line is given by Eq. (10.17). The only difference is that for dehumidification, the operating line lies above the equilibrium line on the T_L–H' plane so that transport of moisture as well as sensible heat occurs from the gas- to the liquid-phase. The water temperature increases down the column.

10.6.4 Humidification and Gas Cooling with Water at the Adiabatic Saturation Temperature

In some applications, a hot gas having a rather small quantity of moisture may have to be cooled. This can be done in a conventional heat exchanger by indirect contact with a cold fluid, for example, cooling water. If an *increase* in the moisture is *not undesirable*, cooling may be conveniently done by direct contact with cool water. The gas is cooled, but simultaneously gets humidified. In such an arrangement in practice, the water in the gas cooling tower is continuously recycled with the addition of makeup water to compensate for the evaporation loss (Figure 10.3). If we neglect heat loss or gain through the tower wall, cooling of the gas will follow the adiabatic saturation temperature line. The gas temperature will *approach* its adiabatic saturation temperature, T_{as}, at the exit or top end of the tower. So, at the steady-state operating condition, the cooling water will effectively remain at its adiabatic saturation temperature.

The temperature and humidity of the gas change along the adiabatic saturation line, but its enthalpy remains constant. The liquid temperature also remains constant at T_{as}. So the terminals of the operating line merge and the operating line shrinks to a point on the equilibrium line. The enthalpy driving force is zero as a result. So the design calculation is to be based on either the temperature or the humidity driving force. If we use the humidity driving force as the basis of calculation, we take help of the following differential mass balance equation

$$G_s dY' = k_{Y'}\bar{a}(Y'_{as} - Y')dz \tag{10.36}$$

Integrating
$$G_s \int_{Y'_1}^{Y'_2} \frac{dY'}{Y'_{as} - Y'} = k_{Y'}\bar{a}z \Rightarrow z = \frac{G_s}{k_{Y'}\bar{a}} \ln\frac{Y'_{as} - Y'_1}{Y'_{as} - Y'_2} = H_{tG}N_{tG} \tag{10.37}$$

Example 10.5 shows an application of the above equation.

EXAMPLE 10.5 (*Humidification of a moist gas with water at the adiabatic saturation temperature*) Nitrogen gas (dry-bulb temp. = 70°C; wet-bulb temp. = 30°C; pressure = 1 atm) is supplied to a cooling tower at a rate of 5500 $m^3/h \cdot m^2$. The water is continuously recirculated (with necessary makeup to compensate for the vaporization loss) and its temperature is maintained essentially at the adiabatic saturation temperature of the gas. The humidified gas leaves the tower 97% saturated. Calculate (a) the rate of supply of makeup water, (b) the number of overall gas-phase transfer units, and (c) the packed height if the overall volumetric mass transfer coefficient is given as $K_{Y'}\bar{a}$ = 2850 kg/(h)(m^2)($\Delta Y'$).

Hints: Vapour pressure of water at 30°C = 0.04244 bar; humidity of saturated N_2 at 30°C = 0.0281.

Enthalpy of saturated N_2 at 30°C [from Eq. (10.7)] = 102 kJ/(kg dry gas) = enthalpy of the feed N_2 at 70°C (since the hot gas gets 'adiabatically' saturated). If the humidity of the feed nitrogen is Y_1',

$102 = 2500Y_1' + c_H(70)$ [c_H = humid heat of nitrogen] $\Rightarrow Y_1' = 0.012$ kg/(kg dry nitrogen).

Since the nitrogen leaving the tower is 97% saturated, $Y_2' = (0.97)(0.0281) = 0.0273$.

From Eq. (10.37), $N_{tOG} = 3$. $H_{tOG} = G_s/K_{Y'}\bar{a} = 1.952$ m; $z = 5.85$ m.

10.7 SOME ADDITIONAL INFORMATION ON COOLING TOWERS

10.7.1 Blowdown

About 1–2% of the water circulating through the cooling tower evaporates in the cooling process. This leads to a buildup of the total dissolved solids (TDS) in the water. The TDS content has to be kept below a certain limit in order to avoid excessive fouling of heat exchangers and other water-cooled equipment. This is done by 'blowdown', i.e. by continuously discarding a part of the circulating water (Figure 10.1). Makeup water is simultaneously added to compensate for the blowdown, evaporation and other losses like drift loss and leakages. The blowdown rate may be calculated using the following simple equation.

Let B = blowdown rate; E = rate of evaporation loss; D = rate of losses due to 'drift' and leakages; C_1 = dissolved solid concentration in the makeup water; C_2 = that in the circulating water; M = makeup water rate.

Total water balance: $$M = B + D + E \tag{10.38}$$

Solids balance: $$MC_1 = (B + D)C_2 + (E)(0) \tag{10.39}$$

The water vaporized (E) does not have any solids in it, and the TDS in the blowdown and in the drift is the same as that in the circulating water. From the above equations,

$$(B + D + E)C_1 = (B + D)C_2 \quad \Rightarrow \quad B = \frac{E - D(r-1)}{r-1};\ r = \frac{C_2}{C_1} \tag{10.40}$$

Once the blowdown rate B is known, the makeup rate M may be calculated from Eq. (10.39).

The evaporation loss may be estimated by a thumb rule (Hensley, 1985) as follow:

$$E = (\text{water flow rate, gpm})(\text{range, °F})(0.0008), \text{ in gallon per minute (gpm)} \quad (10.41)$$

EXAMPLE 10.6 (*Blowdown calculation*) An induced draft crossflow tower is rated to cool 15,000 gpm of water from 40°C to 29°C. The total solids concentration must not exceed 900 ppm. The TDS of the makeup water is 300 ppm. About 0.1% of the water is lost by 'drift' from the tower and leakages in the circulation system. Calculate the blowdown and makeup rate.

Solution

The range = 40 – 29 = 11°C = 19.8°F

Evaporation rate, E = (15,000)(19.8)(0.0008) = 237.6 gpm [from Eq. (10.41)]

Drift and leakages, D = 0.1% of 15,000 gpm = 15 gpm; $r = C_2/C_1$ = 900/300 = 3

Blowdown rate [Eq. (10.40)], B = [237.6 – (15)(3 – 1)]/(3 – 1) = 104 gpm

Makeup water rate, $M = B + D + E$ = 104 + 15 + 237.6 = 356.6 gpm

10.7.2 The Problem of Legioellosis

A cooling tower offers an environment congenial for the growth of a rod-shaped bacteria called *legionella pneumophila* that causes respiratory diseases. Infection may be caused by mist of drift from a cooling tower. The disease is reported to have claimed lives in many places. Cooling tower water must be tested periodically for the presence of this bacteria. Dosing of ozone, hydrogen peroxide, or the use of a UV source have all been found to be useful to control the growth of this bacteria. Also, a cooling tower must be cleaned periodically and stagnant pockets of water must be avoided (Ondrey et al., 1999; Silverberg, 1997). In some countries a code of practice (COP) for cooling tower operation has been introduced (Ondrey et al., 1999) to check the problems of bacterial growth, plume, etc.

10.7.3 Typical Values of Design Variables

The practical values of the important design variables and parameters of cooling towers are given below.

Range = 8–15°C; approach = 5–6°C; evaporation loss = 1.5–2%; drift loss = 0.02–0.2%; overall volumetric gas-phase mass transfer coefficient, $K_{Y'}\bar{a}$ = 1500–3000 kg/(h)(m^2)($\Delta Y'$); packing depth = 5–10 m for splash packings, 1.5–2 m for filmpack; liquid mass flow rate = 2–5 gallon/min·ft^2; L/G = 0.75 to 1.5; free space above the drift eliminator = 2–3 m; height of the fan stack (or fan cylinder) = 3–5 m; spacing of corrugated sheets in a film pack = 19–25 mm.

NOTATIONS

$\bar{a}$: specific interfacial area of contact between water and air, m^2/m^3

A : water vapour