Then, using Eq. (10.4-24),
\[
\frac{1}{K'_y / 0.773} = \frac{1}{1.465 \times 10^{-3} / 0.709 + 1.967 \times 10^{-3} / 0.820}
\]
\[
= 484.0 + 384.8 = 868.8
\]
Solving, \( K'_y = 8.90 \times 10^{-4} \). The percent resistance in the gas film is \( (484.0/868.8)\times 100 = 55.7\% \) and \( 44.3\% \) in the liquid film. The flux is as follows, using Eq. (10.4-22):
\[
N_A = \frac{K'_y}{(1 - y_{A0})_{M}} (y_{AG} - y_{A0}) = \frac{8.90 \times 10^{-4}}{0.773} (0.380 - 0.052)
\]
\[
= 3.78 \times 10^{-4} \text{ kg mol/s·m}^2
\]
This, of course, is the same flux value as was calculated in Example 10.4-1 using the film equations.

4. Discussion of overall coefficients. If the two-phase system is such that the major resistance is in the gas phase as in Eq. (10.4-19), then to increase the overall rate of mass transfer, efforts should be centered on increasing the gas-phase turbulence, not the liquid-phase turbulence. For a two-phase system where the liquid film resistance is controlling, turbulence should be increased in this phase to increase rates of mass transfer.

To design mass-transfer equipment, the overall mass-transfer coefficient is synthesized from the individual film coefficients, as discussed in this section.

10.5 CONTINUOUS HUMIDIFICATION PROCESSES

10.5A Introduction and Types of Equipment for Humidification

1. Introduction to gas–liquid contactors. When a relatively warm liquid is directly contacted with gas that is unsaturated, some of the liquid is vaporized. The liquid temperature will drop mainly because of the latent heat of evaporation. This direct contact of a gas with a pure liquid occurs most often in contacting air with water. This is done for the following purposes: humidifying air for control of the moisture content of air in drying or air conditioning; dehumidifying air, where cold water condenses some water vapor from warm air; and water cooling, where evaporation of water to the air cools warm water.

In Chapter 9 the fundamentals of humidity and adiabatic humidification were discussed. In this section the performance and design of continuous air–water contactors is considered. The emphasis is on cooling of water, since this is the most important type of process in the process industries. There are many cases in industry in which warm water is discharged from heat exchangers and condensers when it would be more economical to cool and reuse it than to discard it.

2. Towers for water cooling. In a typical water-cooling tower, warm water flows countercurrently to an air stream. Typically, the warm water enters the top of a packed tower and cascades down through the packing, leaving at the bottom. Air enters at the bottom of the tower and flows upward through the descending water. The tower packing often consists of slats of wood or plastic or of a packed bed. The water is distributed by troughs and overflows to cascade over slat gratings or packing that provide large interfacial areas of contact between the water and air in the form of droplets and films.
of water. The flow of air upward through the tower can be induced by the buoyancy of the warm air in the tower (natural draft) or by the action of a fan. Detailed descriptions of towers are given in other texts (B1, T1).

The water cannot be cooled below the wet bulb temperature. The driving force for the evaporation of the water is approximately the vapor pressure of the water less the vapor pressure it would have at the wet bulb temperature. The water can be cooled only to the wet bulb temperature, and in practice it is cooled to about 3 K or more above this. Only a small amount of water is lost by evaporation in cooling water. Since the latent heat of vaporization of water is about 2300 kJ/kg, a typical change of about 8 K in water temperature corresponds to an evaporation loss of about 1.5%. Hence, the total flow of water is usually assumed to be constant in calculations of tower size.

In humidification and dehumidification, intimate contact between the gas phase and liquid phase is needed for large rates of mass transfer and heat transfer. The gas-phase resistance controls the rate of transfer. Spray or packed towers are used to give large interfacial areas and to promote turbulence in the gas phase.

10.5B Theory and Calculation of Water-Cooling Towers

1. Temperature and concentration profiles at interface. In Fig. 10.5-1 the temperature profile and the concentration profile in terms of humidity are shown at the water–gas interface. Water vapor diffuses from the interface to the bulk gas phase with a driving force in the gas phase of \((H_i - H_G)\) kg H₂O/kg dry air. There is no driving force for mass transfer in the liquid phase, since water is a pure liquid. The temperature driving force is \(T_L - T_i\) in the liquid phase and \(T_i - T_G\) K or °C in the gas phase. Sensible heat flows from the bulk liquid to the interface in the liquid. Sensible heat also flows from the interface to the gas phase. Latent heat also leaves the interface in the water vapor, diffusing to the gas phase. The sensible heat flow from the liquid to the interface equals the sensible heat flow in the gas plus the latent heat flow in the gas.

The conditions in Fig. 10.5-1 occur at the upper part of the cooling tower. In the lower part of the cooling tower the temperature of the bulk water is higher than the wet bulb temperature of the air but may be below the dry bulb temperature. Then the direction of the sensible heat flow in Fig. 10.5-1 is reversed.

2. Rate equations for heat and mass transfer. We shall consider a packed water-cooling

![Diagram of temperature and concentration profiles in upper part of cooling tower.]

**Figure 10.5-1.** Temperature and concentration profiles in upper part of cooling tower.

Sec. 10.5 Continuous Humidification Processes
tower with air flowing upward and water countercurrently downward in the tower. The total interfacial area between the air and water phases is unknown, since the surface area of the packing is not equal to the interfacial area between the water droplets and the air. Hence, we define a quantity $a$, defined as m$^2$ of interfacial area per m$^3$ volume of packed section, or m$^2$/m$^3$. This is combined with the gas-phase mass-transfer coefficient $k_G$ in kg mol/s · m$^2$ · Pa or kg mol/s · m$^2$ · atm to give a volumetric coefficient $k_G a$ in kg mol/s · m$^3$ volume · Pa or kg mol/s · m$^3$ · atm (lb mol/h · ft$^2$ · atm).

The process is carried out adiabatically and the various streams and conditions are shown in Fig. 10.5-2, where

$$L = \text{water flow, kg water/s · m}^2 \cdot (\text{lb}_m / \text{h} \cdot \text{ft}^2)$$

$$T_L = \text{temperature of water, °C or K (°F)}$$

$$G = \text{dry air flow, kg/s · m}^2 \cdot (\text{lb}_m / \text{h} \cdot \text{ft}^2)$$

$$T_G = \text{temperature of air, °C or K (°F)}$$

$$H = \text{humidity of air, kg water/kg dry air (lb water/lb dry air)}$$

$$H_y = \text{enthalpy of air–water vapor mixture, J/kg dry air (btu/lb}_m \text{ dry air)}$$

The enthalpy $H_y$ as given in Eq. (9.3-8) is

$$H_y = c_s(T - T_o) + H\lambda_o = (1.005 + 1.888H)10^3(T - 0) + 2.501 \times 10^6H \quad \text{(SI)}$$

$$H_y = c_s(T - T_o) + H\lambda_o = (0.24 + 0.45H)(T - 32) + 1075.4H \quad \text{(English)}$$

The base temperature selected is 0°C or 273 K (32°F). Note that $(T - T_o)°C = (T - T_o)$ K.

Making a total heat balance for the dashed-line box shown in Fig. 10.5-2, an operating line is obtained.
\[ G(H_y - H_{y1}) = Lc_L(T_L - T_{L1}) \quad (10.5-1) \]

This assumes that \( L \) is essentially constant, since only a small amount is evaporated. The heat capacity \( c_L \) of the liquid is assumed constant at \( 4.187 \times 10^3 \text{ J/kg} \cdot \text{K} \) (1.00 btu/lbm \cdot °F). When plotted on a chart of \( H_y \) versus \( T_L \), this Eq. (10.5-1) is a straight line with a slope of \( Lc_L/G \). Making an overall heat balance over both ends of the tower,

\[ G(H_y2 - H_{y1}) = Lc_L(T_{L2} - T_{L1}) \quad (10.5-2) \]

Again making a heat balance for the \( dz \) column height and neglecting sensible heat terms compared to the latent heat,

\[ Lc_L \frac{dT_L}{dz} = G \frac{dH_y}{dz} \quad (10.5-3) \]

The total sensible heat transfer from the bulk liquid to the interface is (refer to Fig. 10.5-1)

\[ Lc_L \frac{dT_L}{dz} = G \frac{dH_y}{dz} = h_L a \frac{d(T_L - T_i)}{dz} \quad (10.5-4) \]

where \( h_L a \) is the liquid-phase volumetric heat-transfer coefficient in \( \text{W/m}^3 \cdot \text{K} \) (btu/h \cdot ft\(^3\) \cdot °F) and \( T_i \) is the interface temperature.

For adiabatic mass transfer the rate of heat transfer due to the latent heat in the water vapor being transferred can be obtained from Eq. (9.3-16) by rearranging and using a volumetric basis.

\[ \frac{q_s}{A} = M_B k_g a P \tau_0 (H_i - H_G) \frac{dz}{dz} \quad (10.5-5) \]

where \( q_s/A \) is in \( \text{W/m}^2 \) (btu/h \cdot ft\(^2\)), \( M_B \) is molecular weight of air, \( k_g a \) is a volumetric mass-transfer coefficient in the gas in \( \text{kg mol/s} \cdot \text{m}^3 \). \( P \), \( \tau_0 \) is the latent heat of water in \( \text{J/kg} \) water, \( H_i \) is the humidity of the gas at the interface in kg water/kg dry air, and \( H_G \) is the humidity of the gas in the bulk gas phase in kg water/kg dry air. The rate of sensible heat transfer in the gas is

\[ \frac{q_s}{A} = h_g a (T_i - T_G) \frac{dz}{dz} \quad (10.5-6) \]

where \( q_s/A \) is in \( \text{W/m}^2 \) and \( h_g a \) is a volumetric heat-transfer coefficient in the gas in \( \text{W/m}^3 \cdot \text{K} \).

Now from Fig. 10.5-1, Eq. (10.5-4) must equal the sum of Eqs. (10.5-5) and (10.5-6).

\[ G dH_y = M_B k_g a P \tau_0 (H_i - H_G) \frac{dz}{dz} + h_g a (T_i - T_G) \frac{dz}{dz} \quad (10.5-7) \]

Equation (9.3-18) states that

\[ \frac{h_g a}{M_B k_g a} \cong c_s \quad (10.5-8) \]

Substituting \( P k_g a \) for \( k_g a \),

\[ \frac{h_g a}{M_B P k_g a} \cong c_s \quad (10.5-9) \]

Substituting Eq. (10.5-9) into Eq. (10.5-7) and rearranging,

\[ G dH_y = M_B k_g a P d \left[ (c_s T_i + \lambda_0 H_i) - (c_s T_G + \lambda_0 H_G) \right] \quad (10.5-10) \]

Adding and subtracting \( c_s T_0 \) inside the brackets,

\[ G dH_y = M_B k_g a P d \left[ c_s (T_i - T_0) + H_i \lambda_0 - [c_s (T_G - T_0) + H_G \lambda_0] \right] \quad (10.5-11) \]
The terms inside the braces are \((H_{yi} - H_y)\), and Eq. (10.5-11) becomes

\[
G \ aH_y = M_B \ k_G \ aP \ dz (H_{yi} - H_y) \tag{10.5-12}
\]

Integrating, the final equation to use to calculate the tower height is

\[
\int_0^z dz = \frac{G}{M_B \ k_G \ aP} \int_{H_{yi}}^{H_y} \frac{dH_y}{H_{yi} - H_y} \tag{10.5-13}
\]

If Eq. (10.5-4) is equated to Eq. (10.5-12) and the result rearranged,

\[
\frac{h_L a}{k_G \ aM_B P} = \frac{H_{yi} - H_y}{T_i - T_L} \tag{10.5-14}
\]

10.5C Design of Water-Cooling Tower Using Film Mass-Transfer Coefficients

The tower design is done using the following steps.

1. The enthalpy of saturated air \(H_{yi}\) is plotted versus \(T_i\) on an \(H\) versus \(T\) plot as shown in Fig. 10.5-3. This enthalpy is calculated with Eq. (9.3-8) using the saturation humidity from the humidity chart for a given temperature, with 0°C (273 K) as a base temperature. Calculated values are tabulated in Table 10.5-1.

2. Knowing the entering air conditions \(T_{g1}\) and \(H_i\), the enthalpy of this air \(H_{yi}\) is calculated from Eq. (9.3-8). The point \(H_{yi}\) and \(T_{L1}\) (desired leaving water temperature) is plotted in Fig. 10.5-3 as one point on the operating line. The operating line is plotted with a slope \(Lc_L/G\) and ends at point \(T_{L2}\), which is the entering water temperature. This gives \(H_{y2}\). Alternatively, \(H_{y2}\) can be calculated from Eq. (10.5-2).

3. Knowing \(h_L a\) and \(k_G a\), lines with a slope of \(-h_L a/k_G aM_B P\) are plotted as shown in Fig. 10.5-3. From Eq. (10.5-14) point \(P\) represents \(H_y\) and \(T_L\) on the operating line, and point \(M\) represents \(H_{yi}\) and \(T_i\), the interface conditions. Hence, line \(MS\) or \(H_{yi} - H_y\) represents the driving force in Eq. (10.5-13).

![Figure 10.5-3. Temperature enthalpy diagram and operating line for water-cooling tower.](image)

Chap. 10  Stage and Continuous Gas–Liquid Separation Processes
<table>
<thead>
<tr>
<th>$T_L$</th>
<th>$H_y$</th>
<th>$J$</th>
<th>$T_L$</th>
<th>$H_y$</th>
<th>$J$</th>
</tr>
</thead>
<tbody>
<tr>
<td>°F</td>
<td>°C</td>
<td>lb m dry air</td>
<td>kg dry air</td>
<td>°F</td>
<td>°C</td>
</tr>
<tr>
<td>60</td>
<td>15.6</td>
<td>18.78</td>
<td>$43.68 \times 10^3$</td>
<td>100</td>
<td>37.8</td>
</tr>
<tr>
<td>80</td>
<td>26.7</td>
<td>36.1</td>
<td>$84.0 \times 10^3$</td>
<td>105</td>
<td>40.6</td>
</tr>
<tr>
<td>85</td>
<td>29.4</td>
<td>41.8</td>
<td>$97.2 \times 10^3$</td>
<td>110</td>
<td>43.3</td>
</tr>
<tr>
<td>90</td>
<td>32.2</td>
<td>48.2</td>
<td>$112.1 \times 10^3$</td>
<td>115</td>
<td>46.1</td>
</tr>
<tr>
<td>95</td>
<td>35.0</td>
<td>55.4</td>
<td>$128.9 \times 10^3$</td>
<td>140</td>
<td>60.0</td>
</tr>
</tbody>
</table>

4. The driving force $H_{y1} - H_y$ is computed for various values of $T_L$ between $T_{L1}$ and $T_{L2}$. Then by plotting $1/(H_{y1} - H_y)$ versus $H_y$ from $H_{y1}$ to $H_{y2}$, a graphical integration is performed to obtain the value of the integral in Eq. (10.5-13). Finally, the height $z$ is calculated from Eq. (10.5-13).

### 10.5D Design of Water-Cooling Tower Using Overall Mass-Transfer Coefficients

Often, only an overall mass-transfer coefficient $K_G a$ in kg mol/s·m$^3$·Pa or kg mol/s·m$^3$·atm is available, and Eq. (10.5-13) becomes

$$z = \frac{G}{M_B K_G a P} \int_{H_{y1}}^{H_{y2}} \frac{dH_y}{H_y^* - H_y} \quad (10.5-15)$$

The value of $H_y^*$ is determined by going vertically from the value of $H_y$ at point $P$ up to the equilibrium line to give $H_y^*$ at point $R$, as shown in Fig. 10.5-3. In many cases the experimental film coefficients $k_G a$ and $h_L a$ are not available. The few experimental data available indicate that $h_L a$ is quite large and the slope of the lines $-h_L a/(k_G a M_B P)$ in Eq. (10.5-14) would be very large and the value of $H_y^*$ would approach that of $H_y^*$ in Fig. 10.5-3.

The tower design using the overall mass-transfer coefficient is done using the following steps.

1. The enthalpy–temperature data from Table 10.5-1 are plotted as shown in Fig. 10.5-3.
2. The operating line is calculated as in steps 1 and 2 for the film coefficients and plotted in Fig. 10.5-3.
3. In Fig. 10.5-3 point $P$ represents $H_y$ and $T_L$ on the operating line and point $R$ represents $H_y^*$ on the equilibrium line. Hence, the vertical line $RP$ or $H_y^* - H_y$ represents the driving force in Eq. (10.5-15).
4. The driving force $H_y^* - H_y$ is computed for various values of $T_L$ between $T_{L1}$ and $T_{L2}$. Then by plotting $1/(H_y^* - H_y)$ versus $H_y$ from $H_{y1}$ to $H_{y2}$, a graphical integration is performed to obtain the value of the integral in Eq. (10.5-15). Finally, the height $z$ is obtained from Eq. (10.5-15).

If experimental cooling data in an actual run in a cooling tower with known height $z$ are available, then using Eq. (10.5-15), the experimental value of $K_G a$ can be obtained.
EXAMPLE 10.5-1. Design of Water-Cooling Tower Using Film Coefficients

A packed countercurrent water-cooling tower using a gas flow rate of $G = 1.356$ kg dry air/s·m² and a water flow rate of $L = 1.356$ kg water/s·m² is to cool the water from $T_{L2} = 43.3$°C (110°F) to $T_{L1} = 29.4$°C (85°F). The entering air at 29.4°C has a wet bulb temperature of 23.9°C. The mass-transfer coefficient $k_{G}a$ is estimated as $1.207 \times 10^{-7}$ kg mol/s·m³·Pa and $h_{L}a/k_{G}aM_{B}P$ as $4.187 \times 10^{4}$ J/kg·K (10.0 btu/lb·°F). Calculate the height of packed tower $z$. The tower operates at a pressure of $1.013 \times 10^{5}$ Pa.

Solution: Following the steps outlined, the enthalpies from the saturated air–water vapor mixtures from Table 10.5-1 are plotted in Fig. 10.5-4. The inlet air at $T_{G1} = 29.4$°C has a wet bulb temperature of 23.9°C. The humidity from the humidity chart is $H_{i} = 0.0165$ kg H₂O/kg dry air. Substituting into Eq. (9.3-8), noting that $(29.4 - 0)$°C = $(29.4 - 0)$ K,

$$H_{y1} = (1.005 + 1.88 \times 0.0165)10^{3}(29.4 - 0) + 2.501 \times 10^{6}(0.0165)$$

$$= 71.7 \times 10^{3}$$ J/kg

The point $H_{y1} = 71.7 \times 10^{3}$ and $T_{L1} = 29.4$°C is plotted. Then substituting into Eq. (10.5-2) and solving,

$$1.356(H_{y2} - 71.7 \times 10^{3}) = 1.356(4.187 \times 10^{4})(43.3 - 29.4)$$

$H_{y2} = 129.9 \times 10^{3}$ J/kg dry air (55.8 btu/lbₜ). The point $H_{y2} = 129.9 \times 10^{3}$ and $T_{L2} = 43.3$°C is also plotted, giving the operating line. Lines with slope $-h_{L}a/k_{G}aM_{B}P = -41.87 \times 10^{3}$ J/kg·K are plotted giving $H_{yi}$ and $H_{y}$ values, which are tabulated in Table 10.5-2 along with derived values as shown. Values of $1/(H_{yi} - H_{y})$ are plotted versus $H_{y}$ and the area under the curve from $H_{y1} = 71.7 \times 10^{3}$ to $H_{y2} = 129.9 \times 10^{3}$ is

$$\int_{H_{y1}}^{H_{y2}} \frac{dH_{y}}{H_{yi} - H_{y}} = 1.82$$

![Graphical solution of Example 10.5-1.](image)

Figure 10.5-4. Graphical solution of Example 10.5-1.
TABLE 10.5-2. Enthalpy Values for Solution to Example 10.5-1 (enthalpy in J/kg dry air)

<table>
<thead>
<tr>
<th>$H_{yi}$</th>
<th>$H_y$</th>
<th>$H_{yi} - H_y$</th>
<th>$1/(H_{yi} - H_y)$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$94.4 \times 10^3$</td>
<td>$71.7 \times 10^3$</td>
<td>$22.7 \times 10^3$</td>
<td>$4.41 \times 10^{-5}$</td>
</tr>
<tr>
<td>$108.4 \times 10^3$</td>
<td>$83.5 \times 10^3$</td>
<td>$24.9 \times 10^3$</td>
<td>$4.02 \times 10^{-5}$</td>
</tr>
<tr>
<td>$124.4 \times 10^3$</td>
<td>$94.9 \times 10^3$</td>
<td>$29.5 \times 10^3$</td>
<td>$3.39 \times 10^{-5}$</td>
</tr>
<tr>
<td>$141.8 \times 10^3$</td>
<td>$106.5 \times 10^3$</td>
<td>$35.3 \times 10^3$</td>
<td>$2.83 \times 10^{-5}$</td>
</tr>
<tr>
<td>$162.1 \times 10^3$</td>
<td>$118.4 \times 10^3$</td>
<td>$43.7 \times 10^3$</td>
<td>$2.29 \times 10^{-5}$</td>
</tr>
<tr>
<td>$184.7 \times 10^3$</td>
<td>$129.9 \times 10^3$</td>
<td>$54.8 \times 10^3$</td>
<td>$1.82 \times 10^{-5}$</td>
</tr>
</tbody>
</table>

Substituting into Eq. (10.5-13),

$$z = \frac{G}{M_g \cdot k \cdot a \cdot \rho} \int_{H_{yi}}^{H_y} \frac{dH_y}{H_{yi} - H_y} = \frac{1.356}{29(1.207 \times 10^{-7})(1.013 \times 10^5)} (1.82)$$

$$= 6.98 \text{ m (22.9 ft)}$$

10.5E Minimum Value of Air Flow

Often the air flow $G$ is not fixed but must be set for the design of the cooling tower. As shown in Fig. 10.5-5 for a minimum value of $G$, the operating line $MN$ is drawn through the point $H_{yi}$ and $T_{L1}$ with a slope that touches the equilibrium line at $T_{L2}$, point $N$. If the equilibrium line is quite curved, line $MN$ could become tangent to the equilibrium line at a point farther down the equilibrium line than point $N$. For the actual tower, a value of $G$ greater than $G_{\text{min}}$ must be used. Often, a value of $G$ equal to 1.3 to 1.5 times $G_{\text{min}}$ is used.

10.5F Design of Water-Cooling Tower Using Height of a Transfer Unit

Often another form of the film mass-transfer coefficient is used in Eq. (10.5-13):

$$z = H_G \int_{H_{yi}}^{H_y} \frac{dH_y}{H_{yi} - H_y}$$

(10.5-16)

![Diagram](image-url)

**Figure 10.5-5. Operating-line construction for minimum gas flow.**

Sec. 10.5 Continuous Humidification Processes 609
\[ H_G = \frac{G}{M_b k_G a p} \]  

(10.5-17)

where \( H_G \) is the height of a gas enthalpy transfer unit in m and the integral term is called the number of transfer units. The term \( H_G \) is often used since it is less dependent upon flow rates than \( k_G a \).

Often another form of the overall mass-transfer coefficient \( K_G a \) in kg mol/s \cdot m^3 \cdot Pa or kg mol/s \cdot m^3 \cdot atm is used and Eq. (10.5-15) becomes

\[ z = \frac{G}{M_b K_G a p} \int_{H_y}^{H_{y_2}} \frac{dH_y}{H_y^* - H_y} = H_{OG} \int_{H_y}^{H_{y_2}} \frac{dH_y}{H_y^* - H_y} \]  

(10.5-18)

where \( H_{OG} \) is the height of an overall gas enthalpy transfer unit in m. The value of \( H_y^* \) is determined by going vertically from the value of \( H_y \) up to the equilibrium line as shown in Fig. 10.5-3. This method should be used only when the equilibrium line is almost straight over the range used. However, the \( H_{OG} \) is often used even if the equilibrium line is somewhat curved because of the lack of film mass-transfer coefficient data.

10.5G Temperature and Humidity of Air Stream in Tower

The procedures outlined above do not yield any information on the changes in temperature and humidity of the air–water vapor stream through the tower. If this information is of interest, a graphical method by Mickley (M2) is available. The equation used for the graphical method is derived by first setting Eq. (10.5-6) equal to \( G c_s dT_G \) and combining it with Eqs. (10.5-12) and (10.5-9) to yield Eq. (10.5-19).

\[ \frac{dH_y}{dT_G} = \frac{H_y - H_y^*}{T_i - T^*} \]  

(10.5-19)

10.5H Dehumidification Tower

For the cooling or humidification tower discussed, the operating line lies below the equilibrium line and water is cooled and air humidified. In a dehumidification tower cool water is used to reduce the humidity and temperature of the air that enters. In this case the operating line is above the equilibrium line. Similar calculation methods are used (T1).

10.6 ABSORPTION IN PLATE AND PACKED TOWERS

10.6A Equipment for Absorption and Distillation

1. Introduction to Absorption. As discussed briefly in Section 10.1B, absorption is a mass-transfer process in which a vapor solute \( A \) in a gas mixture is absorbed by means of a liquid in which the solute is more or less soluble. The gas mixture consists mainly of an inert gas and the solute. The liquid also is primarily immiscible in the gas phase; i.e., its vaporization into the gas phase is relatively slight. A typical example is absorption of the solute ammonia from an air–ammonia mixture by water. Subsequently, the solute is recovered from the solution by distillation. In the reverse process of desorption or stripping, the same principles and equations hold.