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Transport Processes & Separation Process Principles (Includes Unit Operations)

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EXAMPLE 4.8-2. Condensation on a Vertical Tube

Steam saturated at 68.9 kPa (10 psia) is condensing on a vertical tube 0.305 m (1.0 ft) long having an OD of 0.0254 m (1.0 in.) and a surface temperature of 86.11°C (187°F). Calculate the average heat-transfer coefficient using English and SI units.

Solution: From Appendix A.2,

$$T_{sat} = 193°\text{F } (89.44°\text{C}) \qquad T_w = 187°\text{F } (86.11°\text{C})$$

$$T_f = \frac{T_w + T_{sat}}{2} = \frac{187 + 193}{2} = 190°\text{F } (87.8°\text{C})$$

$$\text{latent heat } h_{fg} = 1143.3 - 161.0 = 982.3 \text{ btu/lb}_m$$

$$= 2657.8 - 374.6 = 2283.2 \text{ kJ/kg} = 2.283 \times 10^6 \text{ J/kg}$$

$$\rho_l = \frac{1}{0.01657} = 60.3 \text{ lb}_m/\text{ft}^3 = 60.3(16.018) = 966.7 \text{ kg/m}^3$$

$$\rho_v = \frac{1}{40.95} = 0.0244 \text{ lb}_m/\text{ft}^3 = 0.391 \text{ kg/m}^3$$

$$\mu_l = (0.324 \text{ cp})(2.4191) = 0.784 \text{ lb}_m/\text{ft} \cdot \text{h} = 3.24 \times 10^{-4} \text{ Pa} \cdot \text{s}$$

$$k_l = 0.390 \text{ btu/ft} \cdot \text{h} \cdot °\text{F} = (0.390)(1.7307) = 0.675 \text{ W/m} \cdot \text{K}$$

$$L = 1 \text{ ft} = 0.305 \text{ m} \qquad \Delta T = T_{sat} - T_w = 193 - 187 = 6°\text{F } (3.33 \text{ K})$$

Assuming a laminar film, using Eq. (4.8-20) in English as well as SI units, and neglecting ρ_v as compared to ρ_l,

$$N_{Nu} = 1.13\left(\frac{\rho_l^2 g h_{fg} L^3}{\mu_l k_l \, \Delta T}\right)^{1/4}$$

$$= 1.13\left[\frac{(60.3)^2(32.174)(3600)^2(982.3)(1.0)^3}{(0.784)(0.390)(6)}\right]^{1/4} = 6040$$

$$N_{Nu} = 1.13\left[\frac{(966.7)^2(9.806)(2.283 \times 10^6)(0.305)^3}{(3.24 \times 10^{-4})(0.675)(3.33)}\right]^{1/4} = 6040$$

$$N_{Nu} = \frac{hL}{k_l} = \frac{h(1.0)}{0.390} = 6040 \qquad \text{SI units: } \frac{h(0.305)}{0.675} = 6040$$

Solving, $h = 2350 \text{ btu/h} \cdot \text{ft}^2 \cdot °\text{F} = 13\,350 \text{ W/m}^2 \cdot \text{K}$.

Next, the N_{Re} will be calculated to see if laminar flow occurs as assumed. To calculate the total heat transferred for a tube of area

$$A = \pi DL = \pi(1/12)(1.0) = \pi/12 \text{ ft}^2, \qquad A = \pi(0.0254)(0.305) \text{ m}^2$$

$$q = hA \, \Delta T \tag{4.8-24}$$

However, this q must also equal that obtained by condensation of m lb_m/h or kg/s. Hence,

$$q = hA \, \Delta T = h_{fg} m \tag{4.8-25}$$

Substituting the values given and solving for m,

$$2350(\pi/12)(193 - 187) = 982.3(m) \qquad m = 3.77 \text{ lb}_\text{m}/\text{h}$$

$$13.350(\pi)(0.0254)(0.305)(3.33) = 2.284 \times 10^6(m) \qquad m = 4.74 \times 10^{-4} \text{ kg/s}$$

Substituting into Eq. (4.8-21),

$$N_\text{Re} = \frac{4m}{\pi D \mu_l} = \frac{4(3.77)}{\pi(1/12)(0.784)} = 73.5 \qquad \text{N}_\text{Re} = \frac{4(4.74 \times 10^{-4})}{\pi(0.0254)(3.24 \times 10^{-4})} = 73.5$$

Hence, the flow is laminar as assumed.

3. Film-condensation coefficients outside horizontal cylinders. The analysis of Nusselt can also be extended to the practical case of condensation outside a horizontal tube. For a single tube the film starts out with zero thickness at the top of the tube and increases in thickness as it flows around to the bottom and then drips off. If there is a bank of horizontal tubes, the condensate from the top tube drips onto the one below; and so on.

For a vertical tier of N horizontal tubes placed one below the other with outside tube diameter D (M1),

$$N_\text{Nu} = \frac{hD}{k_l} = 0.725\left(\frac{\rho_l(\rho_l - \rho_v)g h_{fg} D^3}{N \mu_l k_l \, \Delta T}\right)^{1/4} \tag{4.8-26}$$

In most practical applications, the flow is in the laminar region and Eq. (4.8-26) holds (C3, M1).

4.9 HEAT EXCHANGERS

4.9A Types of Exchangers

1. Introduction. In the process industries the transfer of heat between two fluids is generally done in heat exchangers. The most common type is one in which the hot and cold fluids do not come into direct contact with each other but are separated by a tube wall or a flat or curved surface. The transfer of heat from the hot fluid to the wall or tube surface is accomplished by convection, through the tube wall or plate by conduction, and then by convection to the cold fluid. In the preceding sections of this chapter we have discussed the calculation procedures for these various steps. Now we will discuss some of the types of equipment used and overall thermal analyses of exchangers. Complete, detailed design methods have been highly developed and will not be considered here.

2. Double-pipe heat exchanger. The simplest exchanger is the double-pipe or concentric-pipe exchanger. This is shown in Fig. 4.9-1, where one fluid flows inside one pipe and the other fluid flows in the annular space between the two pipes. The fluids can be in cocurrent or countercurrent flow. The exchanger can be made from a pair of single lengths of pipe with fittings at the ends or from a number of pairs interconnected in series. This type of exchanger is useful mainly for small flow rates.

3. Shell-and-tube exchanger. If larger flows are involved, a shell-and-tube exchanger is used, which is the most important type of exchanger in use in the process industries. In these exchangers the flows are continuous. Many tubes in parallel are used, where one fluid flows

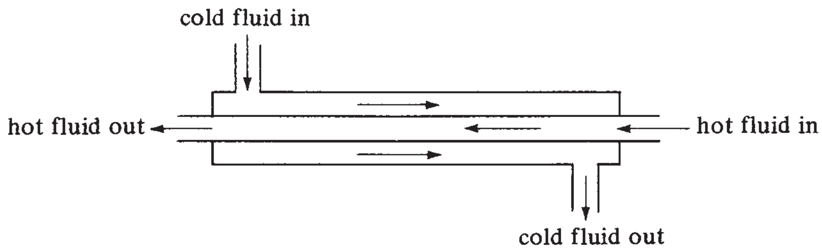

FIGURE 4.9-1. *Flow in a double-pipe heat exchanger.*

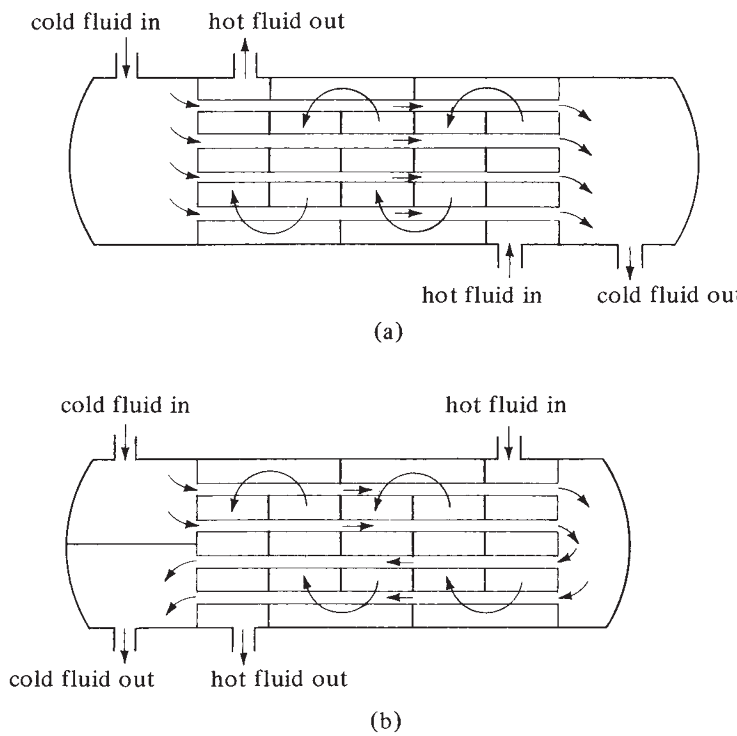

FIGURE 4.9-2. *Shell-and-tube heat exchangers: (a) 1 shell pass and 1 tube pass (1–1 exchanger); (b) 1 shell pass and 2 tube passes (1–2 exchanger).*

inside these tubes. The tubes, arranged in a bundle, are enclosed in a single shell and the other fluid flows outside the tubes in the shell side. The simplest shell-and-tube exchanger is shown in Fig. 4.9-2a for one shell pass and one tube pass, or a 1–1 counterflow exchanger. The cold fluid enters and flows inside through all the tubes in parallel in one pass. The hot fluid enters at the other end and flows counterflow across the outside of the tubes. Cross-baffles are used so that the fluid is forced to flow perpendicular across the tube bank rather than parallel with it. The added turbulence generated by this cross-flow increases the shell-side heat-transfer coefficient.

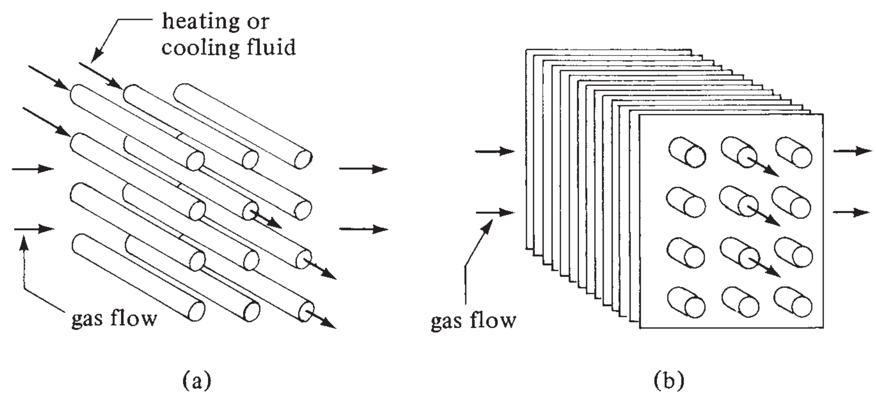

FIGURE 4.9-3. *Flow patterns of cross-flow heat exchangers: (a) one fluid mixed (gas) and one fluid unmixed; (b) both fluids unmixed.*

In Fig. 4.9-2b a 1–2 parallel–counterflow exchanger is shown. The liquid on the tube side flows in two passes as shown and the shell-side liquid flows in one pass. In the first pass of the tube side, the cold fluid is flowing counterflow to the hot shell-side fluid; in the second pass of the tube side, the cold fluid flows in parallel (cocurrent) with the hot fluid. Another type of exchanger has two shell-side passes and four tube passes. Other combinations of number of passes are also used sometimes, with the 1–2 and 2–4 types being the most common.

4. Cross-flow exchanger. When a gas such as air is being heated or cooled, a common device used is the cross-flow heat exchanger shown in Fig. 4.9-3a. One of the fluids, which is a liquid, flows inside through the tubes, and the exterior gas flows across the tube bundle by forced or sometimes natural convection. The fluid inside the tubes is considered to be unmixed, since it is confined and cannot mix with any other stream. The gas flow outside the tubes is mixed, since it can move about freely between the tubes, and there will be a tendency for the gas temperature to equalize in the direction normal to the flow. For the unmixed fluid inside the tubes, there will be a temperature gradient both parallel and normal to the direction of flow.

A second type of cross-flow heat exchanger shown in Fig. 4.9-3b is typically used in air-conditioning and space-heating applications. In this type the gas flows across a finned-tube bundle and is unmixed, since it is confined in separate flow channels between the fins as it passes over the tubes. The fluid in the tubes is unmixed.

Discussions of other types of specialized heat-transfer equipment will be deferred to Section 4.13. The remainder of this section deals primarily with shell-and-tube and cross-flow heat exchangers.

4.9B Log-Mean-Temperature-Difference Correction Factors

In Section 4.5H it was shown that when the hot and cold fluids in a heat exchanger are in true countercurrent flow or in cocurrent (parallel) flow, the log mean temperature difference should be used:

$$\Delta T_{\text{lm}} = \frac{\Delta T_2 - \Delta T_1}{\ln(\Delta T_2/\Delta T_1)} \tag{4.9-1}$$

where ΔT_2 is the temperature difference at one end of the exchanger and ΔT_1 at the other end. This ΔT_{lm} holds for a double-pipe heat exchanger and a 1–1 exchanger with one shell pass and one tube pass in parallel or counterflow.

In cases where a multiple-pass heat exchanger is involved, it is necessary to obtain a different expression for the mean temperature difference, depending on the arrangement of the shell and tube passes. Considering first the one-shell-pass, two-tube-pass exchanger in Fig. 4.9-2b, the cold fluid in the first tube pass is in counterflow with the hot fluid. In the second tube pass, the cold fluid is in parallel flow with the hot fluid. Hence, the log mean temperature difference, which applies either to parallel or to counterflow but not to a mixture of both types, as in a 1–2 exchanger, cannot be used to calculate the true mean temperature drop without a correction.

The mathematical derivation of the equation for the proper mean temperature to use is quite complex. The usual procedure is to use a correction factor F_T which is so defined that when it is multiplied by the ΔT_{lm}, the product is the correct mean temperature drop ΔT_m to use. In using the correction factors F_T, it is immaterial whether the warmer fluid flows through the tubes or the shell (K1). The factor F_T has been calculated (B4) for a 1–2 exchanger and is shown in Fig. 4.9-4a. Two dimensionless ratios are used as follows:

$$Z = \frac{T_{hi} - T_{ho}}{T_{co} - T_{ci}} \qquad \textbf{(4.9-2)}$$

$$Y = \frac{T_{co} - T_{ci}}{T_{hi} - T_{ci}} \qquad \textbf{(4.9-3)}$$

where T_{hi} = inlet temperature of hot fluid in K (°F), T_{ho} = outlet of hot fluid, T_{ci} inlet of cold fluid, and T_{co} = outlet of cold fluid.

In Fig. 4.9-4b, the factor F_T (B4) for a 2–4 exchanger is shown. In general, it is not recommended to use a heat exchanger for conditions under which $F_T < 0.75$. Another shell-and-tube arrangement should be used. Correction factors for two types of cross-flow exchanger are given in Fig. 4.9-5. Other types are available elsewhere (B4, P1).

Using the nomenclature of Eqs. (4.9-2) and (4.9-3), the ΔT_{lm} of Eq. (4.9-1) can be written as

$$\Delta T_{lm} = \frac{(T_{hi} - T_{co}) - (T_{ho} - T_{ci})}{\ln[(T_{hi} - T_{co})/(T_{ho} - T_{ci})]} \qquad \textbf{(4.9-4)}$$

Then the equation for an exchanger is

$$q = U_i A_i \, \Delta T_m = U_o A_o \, \Delta T_m \qquad \textbf{(4.9-5)}$$

where

$$\Delta T_m = F_T \, \Delta T_{lm} \qquad \textbf{(4.9-6)}$$

EXAMPLE 4.9-1. Temperature Correction Factor for a Heat Exchanger

A 1–2 heat exchanger containing one shell pass and two tube passes heats 2.52 kg/s of water from 21.1 to 54.4°C by using hot water under pressure entering at 115.6 and leaving at 48.9°C. The outside surface area of the tubes in the exchanger is $A_o = 9.30\ \text{m}^2$.

(a) Calculate the mean temperature difference ΔT_m in the exchanger and the overall heat-transfer coefficient U_o.