Flow Across Tube Banks

The experimental results for the flow across tube banks are represented in the form of

$$Nu = C.\mathrm{Re}^{n}.\mathrm{Pr}^{1/3}$$

The Re in this equation is base on the maximum velocity in the tube bank,

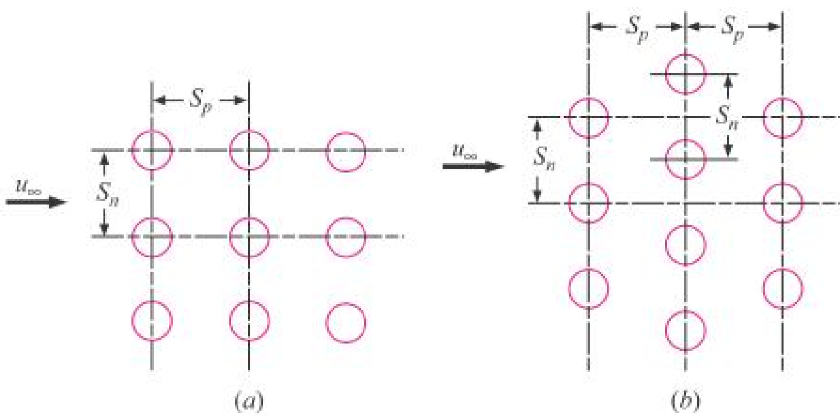
$$\mathrm{Re} = \frac{\rho.u_{\max}.d}{\mu}$$

$$u_{\max} = u_{\infty}[S_n/(S_n - d)]$$

The values of the constant C and the exponent n are given in Table 6-4 in terms of the geometric parameters.

The data of Table 6-4 are used for tube banks having 10 or more rows of tubes in the direction of flow. For fewer rows the ratio of h for N rows deep to that for 10 rows is given in Table 6-5.

Figure 6-14 | Nomenclature for use with Table 6-4: (*a*) in-line tube rows; (*b*) staggered tube rows.



EXAMPLE 6-10 Heating of Air with In-Line Tube Bank

Air at 1 atm and 10°C flows across a bank of tubes 15 rows high and 5 rows deep at a velocity of 7 m/s measured at a point in the flow before the air enters the tube bank. The surfaces of the tubes are maintained at 65°C. The diameter of the tubes is 1 in [2.54 cm]; they are arranged in an in-line manner so that the spacing in both the normal and parallel directions to the flow is 1.5 in [3.81 cm]. Calculate the total heat transfer per unit length for the tube bank and the exit air temperature.

■ **Solution**

The constants for use with Equation (6-17) may be obtained from Table 6-4, using

$$\frac{S_p}{d} = \frac{3.81}{2.54} = 1.5 \qquad \frac{S_n}{d} = \frac{3.81}{2.54} = 1.5$$

so that

$$C = 0.278 \qquad n = 0.620$$

The properties of air are evaluated at the film temperature, which at entrance to the tube bank is

$$T_{f_1} = \frac{T_w + T_\infty}{2} = \frac{65 + 10}{2} = 37.5°\text{C} = 310.5\ \text{K} \quad [558.9°\text{R}]$$

Then

$$\rho_f = \frac{p}{RT} = \frac{1.0132 \times 10^5}{(287)(310.5)} = 1.137\ \text{kg/m}^3$$
$$\mu_f = 1.894 \times 10^{-5}\ \text{kg/m} \cdot \text{s}$$
$$k_f = 0.027\ \text{W/m} \cdot °\text{C} \quad [0.0156\ \text{Btu/h} \cdot \text{ft} \cdot °\text{F}]$$
$$c_p = 1007\ \text{J/kg} \cdot °\text{C} \quad [0.24\ \text{Btu/lb}_m \cdot °\text{F}]$$
$$\text{Pr} = 0.706$$

To calculate the maximum velocity, we must determine the minimum flow area. From Figure 6-14 we find that the ratio of the minimum flow area to the total frontal area is $(S_n - d)/S_n$. The maximum velocity is thus

$$u_{\max} = u_\infty \frac{S_n}{S_n - d} = \frac{(7)(3.81)}{3.81 - 2.54} = 21\ \text{m/s} \quad [68.9\ \text{ft/s}] \qquad \textbf{[a]}$$

where u_∞ is the incoming velocity before entrance to the tube bank. The Reynolds number is computed by using the maximum velocity.

$$\text{Re} = \frac{\rho u_{\max} d}{\mu} = \frac{(1.137)(21)(0.0254)}{1.894 \times 10^{-5}} = 32{,}020 \qquad \textbf{[b]}$$

The heat-transfer coefficient is then calculated with Equation (6-17):

$$\frac{hd}{k_f} = (0.278)(32{,}020)^{0.62}(0.706)^{1/3} = 153.8 \qquad \textbf{[c]}$$

$$h = \frac{(153.8)(0.027)}{0.0254} = 164\ \text{W/m}^2 \cdot °\text{C} \quad [28.8\ \text{Btu/h} \cdot \text{ft}^2 \cdot °\text{F}] \qquad \textbf{[d]}$$

This is the heat-transfer coefficient that would be obtained if there were 10 rows of tubes in the direction of the flow. Because there are only 5 rows, this value must be multiplied by the factor 0.92, as determined from Table 6-5.

The total surface area for heat transfer, considering unit length of tubes, is

$$A = N\pi d(1) = (15)(5)\pi(0.0254) = 5.985\ \text{m}^2/\text{m}$$

where N is the total number of tubes.

Before calculating the heat transfer, we must recognize that the air temperature increases as the air flows through the tube bank. Therefore, this must be taken into account when using

$$q = hA(T_w - T_\infty) \qquad \textbf{[e]}$$

As a good approximation, we can use an arithmetic average value of T_∞ and write for the energy balance*

$$q = hA\left(T_w - \frac{T_{\infty,1} + T_{\infty,2}}{2}\right) = \dot{m}c_p(T_{\infty,2} - T_{\infty,1}) \qquad \textbf{[f]}$$

where now the subscripts 1 and 2 designate entrance and exit to the tube bank. The mass flow at entrance to the 15 tubes is

$$\dot{m} = \rho_\infty u_\infty (15) S_n$$

$$\rho_\infty = \frac{p}{RT_\infty} = \frac{1.0132 \times 10^5}{(287)(283)} = 1.246\ \text{kg/m}^3 \qquad \textbf{[g]}$$

$$\dot{m} = (1.246)(7)(15)(0.0381) = 4.99\ \text{kg/s} \quad [11.0\ \text{lb}_m/\text{s}]$$

so that Equation (f) becomes

$$(0.92)(164)(5.985)\left(65 - \frac{10 + T_{\infty,2}}{2}\right) = (4.99)(1006)(T_{\infty,2} - 10)$$

that may be solved to give

$$T_{\infty,2} = 19.08^\circ\text{C}$$

The heat transfer is then obtained from the right side of Equation (f):

$$q = (4.99)(1006)(19.08 - 10) = 45.6\ \text{kW/m}$$

This answer could be improved somewhat by recalculating the air properties based on a mean value of T_∞, but the improvement would be small and well within the accuracy of the empirical heat-transfer correlation of Equation (6-17).