4.3. Heat Transfer and Pressure Drop in High-Finned Trufin Tube Banks

4.3.1. Heat Transfer Coefficients in Crossflow

1. *Briggs and Young Correlation*. A number of correlations have been published for heat transfer during flow across banks of finned tubes. The number and range of variables are so large that it would be surprising if a relatively simple correlation would be generally applicable. More complex correlations require correspondingly greater data sets with particular emphasis upon wide ranges of variables and multivariate interactions over these ranges, and the published data generally do not meet these criteria. Therefore, the published correlations must be used with great care to ensure that they are applicable in the range of interest.

Within the above caution, one of the best published correlations for high finned Trufin tubes is due to Briggs and Young (5).

$$\frac{h_o d_r}{k_{air}} = 0.134\left(\frac{d_r \rho_{air} V_{\max}}{\mu_{air}}\right)^{0.68} \mathrm{Pr}_{air}{}^{1/3}\left(\frac{H}{s}\right)^{-0.2}\left(\frac{Y}{s}\right)^{-0.12} \tag{4.6}$$

This correlation represents data for root diameters from 0.44 in. to 1.61 in. and fin heights from 0.056 in. to 0.652 in. Fin spacings ranged from 0.035 in. to 0. 117 in. The tubes were in equilateral triangular pitch tube banks with pitches up to 4.5 in.

V_{max} is the maximum air-side velocity going through the finned tube bank and the other quantities have their usual meaning. Fin spacing, s, is related to the number of fins per inch N_f by the equation:

$$s = \left(\frac{1}{N_f}\right) - Y \tag{4.7}$$

It will be useful here to demonstrate the use of this correlation and definitions of the terms by carrying out a typical calculation.

2. *Example*: Calculate the air side heat transfer coefficient for the case of an aluminum Trufin Type H/R tube with a 3/4 in. root diameter with 9 fins/in. with a mean fin thickness of 0.019. The tubes are in an equilateral triangular layout with a 1 7/8 in. pitch. Air is flowing at 70°F with a face velocity of 600 ft/min.

Solution:

The geometrical parameters are as follows:

d_r = 0.750 in.

H = $\frac{1}{2}$ (1.625 – 0.75) in. = 0.438 in.

Y = 0.019 in.

s = (1/9 – 0.019) in. = 0.092 in.

V_{max}: The problem was stated in terms of the "face velocity", i.e., the average velocity of the air approaching the first row of tubes. To convert this to V_{max} we must first find the free flow area between two tubes at the point of closest approach, per foot of tube. Since the centers of adjacent tubes are 1.875 in. apart and the root diameter is 0.75 in., there is a 1.125 in. clearance between the root diameters of the tubes giving a clearance area of 12(1.125) = 13.50 in.^{2}/ft. From this clearance area must be subtracted the area blocked by the fins on each tube, which per foot is

2(9 fin/in.)(12 in./ft.)(0.019 in./fin)(0.438) = 1.80 in^2/ft.

Thus the free flow area between tubes per foot of length is 11.70 in.^{2} The "face area" corresponding to these same two adjacent tubes per foot of length is simply the center-to-center distance between the tubes – the pitch – or

(1.875 in.)(12 in./ft.) = 22.50 in^2/ft.

The air flowing at 600 ft/min. approaching the face must accelerate to

$$600\left(\frac{22.50}{11.70}\right) = 1150 \ ft/\text{min}$$

to flow between the tubes and this is the value of V_{max}. The physical properties of air at 70°F are:

k = 0.0150 Btu/hr ft°F

ρ = 0.0765 lb_m/ft^3

μ = 0.439 $lb_{,m}$/ft hr

c_p = 0.240 Btu/lb_m°F

$$\text{Pr} = \frac{0.240(0.0439)}{0.0150} = 0.702$$

Finally we may calculate h_o:

$$h_o = \frac{(0.134)(0.0150)}{0.750/12}\left[\frac{(0.750/12)(0.0765)(1150)(60)}{0.0439}\right]^{0.68} \times (0.702)^{1/3}\left(\frac{0.438}{0.092}\right)^{-0.2}\left(\frac{0.019}{0.092}\right)^{-0.12}$$

= 0.0322(432)(0.889)(0.732)(1.208)

h_o = 10.9 Btu/hr ft^2°F

This is a very typical value of the air side coefficient, and it may be compared to the relatively very much higher values for tube-side (~1000 for water, 2000 + for condensing steam, ~300 for a medium organic liquid, ~100 for a heavy organic liquid). This comparison illustrates immediately the importance of high-finned Trufin in air cooled service.

4.3.2. Mean Temperature Difference in Crossflow

1. *The LMTD.* It was pointed out in Chapter 1, the great simplification introduced into heat exchanger design by the Mean Temperature Difference (MTD) concept. In its simplest practical form, assuming countercurrent flow, constant overall heat transfer coefficient, etc., the correct definition of the MTD is the Logarithmic Mean Temperature Difference (LMTD) defined as:

$$LMTD = \frac{(T_i - t_o) - (T_o - t_i)}{\ell n\left(\frac{(T_i - t_o)}{(T_o - t_i)}\right)} \quad (4.8)$$

where T_i and T_o are the hot fluid inlet and outlet temperatures and t_i and t_o are the cold fluid inlet and outlet temperatures, respectively.

With this concept one may write the basic design equation (if all of the conditions are satisfied) as

$$A_o = \frac{Q}{U_o(LMTD)} \quad (4.9)$$

where A_o and U_o are referred to the same reference area, usually the total outside heat transfer area of the heat exchanger.

2. *MTD for Crossflow*. In air-cooled heat exchangers, the air and the process fluid are in crossflow to one another, not in countercurrent flow as assumed in the LMTD derivation. In this case, it is necessary to apply a correction factor F which may be obtained by mathematical analysis. The Eq. (4.9) may be written as:

$$A_o = \frac{Q}{U_o F(LMTD)} \quad (4.10)$$

where, it is important to note, the LMTD is calculated as in Eq. (4.8). The correction factor F is a function of the parameters and

$$P = \frac{t_o - t_i}{T_i - t_i} \quad (4.11)$$

and

$$R = \frac{T_i - T_o}{t_o - t_i} \quad (4.12)$$

F is plotted in Fig. 4.13 for crossflow with one tube-side pass, i.e., the tube side fluid flows in parallel through all the tubes. It is assumed that an equal amount of fluid flows through each tube.

F is plotted in Fig. 4.14 for crossflow with two tube-side passes, with an equal number of tubes in each pass. There may be more than one row of tubes in each pass. Note that the tube-side flow goes through the uppermost tubes first, and that the overall flow pattern is moving towards the tube side fluid being in countercurrent flow to the air flow. In fact, for three or more passes, the overall flow pattern is so close to countercurrent that F can be taken to equal to 1.00 with very small error.

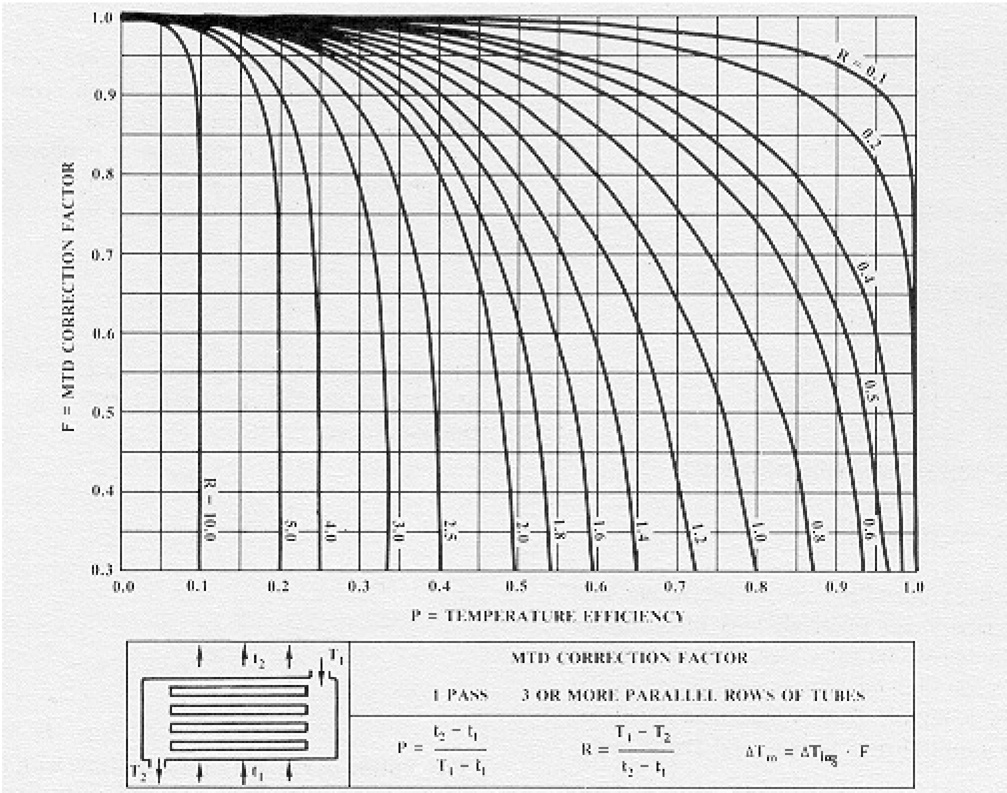

Fig. 4.13 MTD Configuration Correction Factor for Crossflow with One Tube-Side Pass

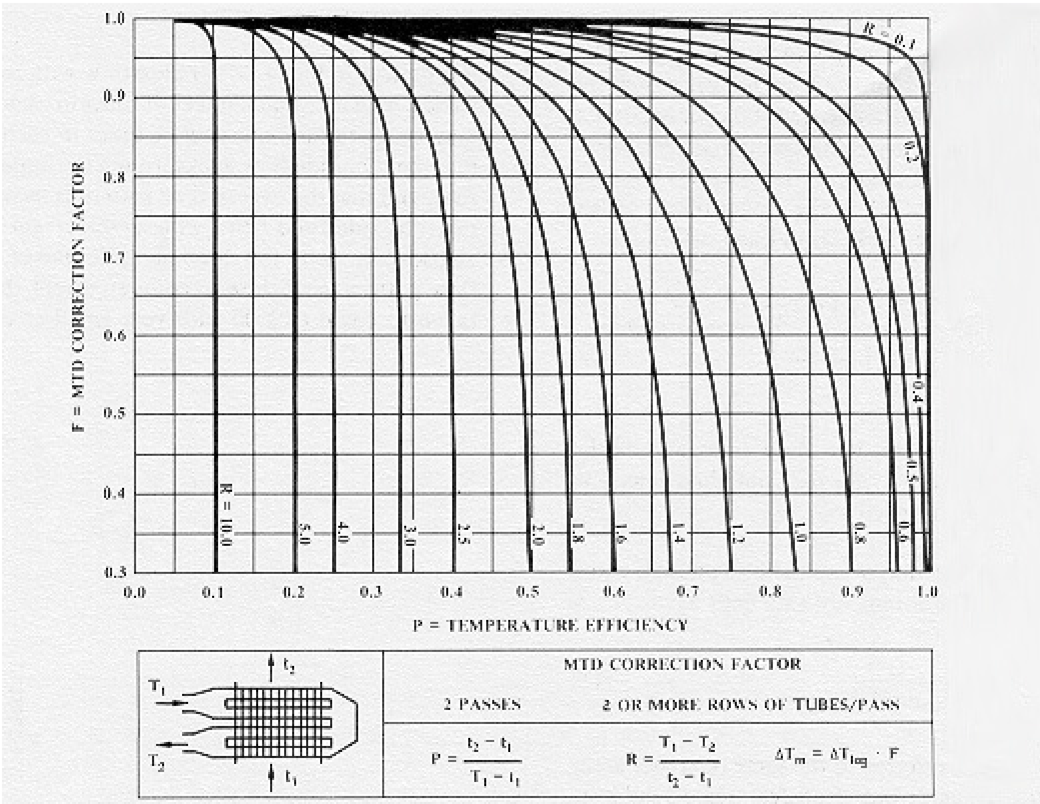

Fig. 4.14 MTD Configuration Correction Factor for Crossflow with Two Tube-Side Passes

4.3.3. Pressure Drop in Crossflow

1. *Robinson and Briggs Correlation. Correlations* for pressure drop across banks of finned tubes are subject to the same cautions as used for heat transfer, and in fact pressure drop is subject to even greater uncertainty. One of the better correlations in the open literature is due to Robinson and Briggs (6):

$$f_r = 9.47\left(\frac{d_r \rho_{air} V_{max}}{\mu_{air}}\right)^{-0.32}\left(\frac{P_t}{d_r}\right)^{-0.93} \tag{4.13}$$

where f_r is the friction factor and P_t is the transverse pitch between adjacent tubes in the same row. The friction factor is defined as

$$f_r = \frac{\Delta p_{air}\, g_c}{2n\rho_{air} V_{max}^2} \tag{4.14}$$

where ΔP_{air} is the pressure drop across the tube bank (in say lb_f/ft^2) and n is the number of tube rows in the bank.

This correlation represents data for tube banks with root diameters from 0.734 to 1.61 in., fin diameters from 1.561 in. to 2.750 in., and pitches from 1.687 to 4.500 in.

Most of the data are for equilateral triangular arrangements. However, two tube banks with isosceles triangular arrangements were tested, and it was found that the data for these two banks could be adequately cor related if the additional factor

$$\left(\frac{P_t}{P}\right)^{0.52}$$

were included on the right-hand side of Eq. (4.13). In the above expression, P is the "longitudinal" pitch of the tube bank, defined as the distance between the centers of adjacent tubes in different rows, measured along the diagonal. For an equilateral arrangement, P_t = P. Since one of the isosceles tube banks tested was close to a staggered square arrangement there is some reason to believe that this correlation will prove adequate for predicting pressure drop in such geometries also.

2. *Example.* Using the same tube and arrangement of the previous example and the same air flow rate, calculate the pressure drop across a tube bank of five rows of tubes.

Solution: The geometrical and flow parameters (including Reynolds number) are unchanged. Then the friction factor f_r is, by Eq. (4.13):

$$f_r = 9.47\left[\frac{(0.750/12)(0.0765)(1150)(60)}{0.439}\right]^{-0.32}\left(\frac{1.875}{0.750}\right)^{-0.93} = 9.47(0.0575)(0.426) = 0.232$$

and the pressure drop is, by Eq. (4.14):

$$\Delta p_{air} = \frac{0.232(2)(5)\left(0.0765\,\mathrm{lb_m/ft^3}\right)(1150\ \mathrm{ft/min})^2}{\left(32.2\,\frac{\mathrm{lb_m}\,ft}{lb_f\ \mathrm{sec}^2}\right)\left(\frac{60\,\mathrm{sec}}{\mathrm{min}}\right)^2} = 2.03\,lb_f\ /\ ft^2 = 0.0141\,lb_f\ /\ in.^2$$

It is customary in stating the air-side pressure drop in air-cooled heat transfer equipment to quote it in "inches of water", that is, as the number of inches of height of a column of water at the earth's sur face that would be supported by the pressure difference. The pressure drop is converted from $lb_f/\ ft^2$ to inches of water by multiplying the reciprocal of the density of water (commonly taken as 62.4 lb_m/ft^3), by (12 in./ft), and by g_c/g which has a numerical value of unity but converts the units properly. So, for this case:

$$\Delta p_{in.\ of\ H_2O} = \left(\Delta p_{air}\,\frac{lb_f}{ft^2}\right)\left(\frac{1}{\rho_{H_2O}}\right)\left(\frac{g_c}{g}\right) \qquad (4.15)$$

$$\Delta p_{in.\ of\ H_2O} = \left(2.03\frac{lb_f}{ft^2}\right)\left(\frac{1}{62.4\,lb_m\ /\ ft^3}\right)\left(\frac{12\ in.}{ft}\right)\left(1\frac{lb_m}{lb_f}\right)$$

$$\Delta p_{in.\ of\ H_2O} = 0.39\ in.\ H_2O$$

This value is well within the capability of fans that would be used in this service. A pressure drop of 1/2 in. of water is a common design value.

4.3.4. Other Air-Side Pressure Effects

There can be other sources of pressure loss on the air-side of an air-cooled exchanger, such as the louver system (even when it is open), fan guards, structural elements, plenums and shrouds. Few data or correlations are generally available in the literature to permit accurate evaluations of these effects. Fortunately, the losses should be very small and a rough estimate of their magnitude is usually sufficient, mainly as reassurance that they will not be major features in limiting the performance of the units.

The following is a useful rule of thumb procedure for any flow situation in which the major pressure effect is from a sudden acceleration of the flow followed by a sudden deceleration.

Calculate the increase in air velocity that must be achieved in order for the air to flow through the restriction (say the open louver system, for example). Then allow twice the velocity head calculated from this velocity increase as the irrecoverable pressure loss. Thus, suppose the flow in the previous example had to accelerate from 800 feet/minute to 900 feet/minute to pass through a fan guard. The velocity head represented by that velocity changeis:

$$\Delta P = \frac{\rho_{air}\left(V_2^2 - V_1^2\right)}{2g_c}$$

$$= \frac{\left(0.0765\ \mathrm{lb_m/ft^3}\right)\left[(900\ \mathrm{ft/min})^2 - (800\ \mathrm{/min})^2\right]}{2\left(32.2\,\dfrac{\mathrm{lb_m\,ft}}{\mathrm{lb_f\ sec^2}}\right)\left(\dfrac{60\,\mathrm{sec}}{\mathrm{min}}\right)^2}$$

$$= 0.056\,\frac{\mathrm{lb_f}}{\mathrm{ft^2}} = 3.9\times10^{-4}\,\frac{\mathrm{lb_f}}{\mathrm{in.^2}} = 0.011\,\mathrm{in.\ H_2O}$$

Doubling this loss gives 0.02 in. H_2O as the probable maximum effect. This is a sufficiently small loss (compared to 0.39 in. of H_2O across the tube bank) that it would not cause a major problem in the design or operation.

If the pressure drop thus calculated turns out to be substantial by this estimate, manufacturers' data or a much more detailed calculation of the effect is required.

Finally, if the discharge air plume is to be accelerated strongly in a smoothly converging duct in order to cause it to travel far above the unit before it mixes with the surrounding air, the additional pressure drop required for acceleration can be conservatively taken to be one velocity head based upon the desired discharge velocity.