2.3. Heat Transfer and Pressure Drop During Flow Across Banks of Trufin Tubes

2.3.1. Heat Transfer In Trufin Tube Banks

The design method for the shell side heat transfer coefficient and pressure drop of a baffled shell and tube heat exchanger using type S/T Trufin will be described in detail in the sample problem -following this Section. However, as an essential component of that calculation, we must have basic heat transfer and fluid flow data for flow across banks of finned tubes. Regrettably, there are very few data available in the open literature for banks of low and medium-finned tubes. * However, the data that are available indicate that, with a small modification, we can use the extensive data and correlations available on banks of plain tubes.

The database that we shall use comes originally from Williams and Katz (3), but the definitive analysis for pre sent purposes was done by Briggs, et al. (4). The latter reference interprets the earlier data in the light of the Delaware method (5) for shell-side heat transfer and pressure drop. The design method proposed later in this Section is a modified version of the Delaware method, so the database and its incorporation into a design method are at least self-consistent.

The heat transfer data for tube banks are correlated as plots of the Colburn j factor for heat transfer vs. the crossflow Reynolds number. The Colburn j factor for crossflow is defined as:

$$j_s = \left(\frac{h_o}{C_p G_m}\right)_s \left(\frac{C_{p^\mu}}{k}\right)_s^{2/3} \left(\frac{\mu_w}{\mu}\right)_s^{0.14} \tag{2.15}$$

where G_m is the mass velocity through the minimum free flow area between adjacent tubes:

$$G_m = \frac{W_s}{S_m} \tag{2.16}$$

and the subscript s indicates the shell-side flow. In a circular tube bundle, S_m will be defined as the minimum free flow area through one crossflow section (i.e., between adjacent baffles) at or near the centerline of the bundle.

The method of calculating S_m will be demonstrated by the examples given in this Section.

The crossflow Reynolds number for a finned tube bank is defined as:

$$\mathrm{Re}_s = \frac{d_r G_m}{\mu_s} \tag{2.17}$$

where d_r is the root diameter of the finned tube. When data for Trufin heat exchangers are compared (via the Delaware method) to those for an otherwise identical plain tube exchanger, Briggs, et al. (4) found results like those in Fig. 2.13, for water, oil, and glycerine.

From Fig. 2.13, we see that there is essentially no difference between the j_s curves for plain and Trufin tube banks at Re_s values above about 500. Below 500, the Trufin tube bank performance falls off compared to the plain tube bank. Comparing all of their data for three different sets of tube banks, Briggs, et al. (4) prepared the graph in Fig. 2.14 which gives the ratio

$$\frac{j_s, \textit{finned tube}}{j_s, \textit{plain tube}}$$

as a function of Re_s below 1000.

It appears that at low Reynolds Numbers, the fins tend to trap the fluid between them, reducing the local velocity between the tubes and therefore the heat transfer coefficient. It should be noted that the Briggs, et al. (4) data were only for tubes having 19 fins/in. More recently, Rabas, et al. (11) have shown an effect in the heat transfer and pressure drop of low fin tube banks due to fin density.

The designer should also note that the j_s ratio appears to be approaching a minimum value of 0.5 at very low Re_s, Since the increase in area due to the fins is always greater than a factor of 2 (more usually 3 or 4), there is still a net increase in the value of $h_o A_o$ compared to plain tubes at a comparable Re_s.

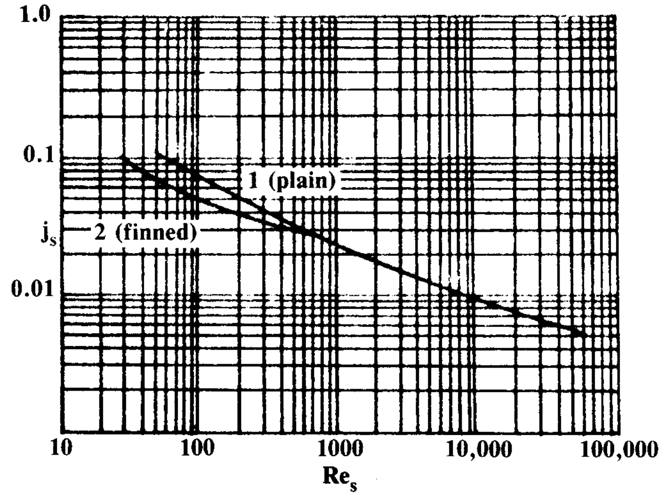

Fig. 2.13 Correlation of j_s vs. Re_s for Two Similar Shell and Tube Heat Exchangers, One Trufin Tube and One Plain Tube

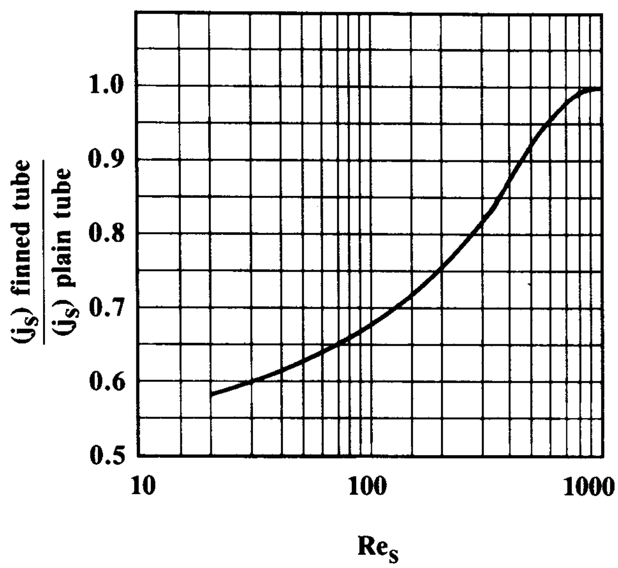

Fig. 2.14 The Relative Heat Transfer Correlation for Trufin Tube Bundles to that for Plain Tube Bundles

*The equations used for banks of high-finned tubes cannot be applied to low-finned tubes, partly because the large amount of data on which they are based was only obtained for air over narrow ranges of flow rates, temperatures, and pressures. Also, the geometrical proportions of the fins and tubes are quite different.

Using the above data, we can now construct the basic design curves for the most common tube bank geometries. In Ref. (5), and more recently in a somewhat revised form (e.g., Perry's Chemical Engineer's Handbook (6)), curves of j_s vs. Re_s have been presented for the most common tube layouts used in shell-and-tube heat exchangers. Applying the ratio shown in Fig. 2.14, we generate the basic j_s vs. Re_s curves for Trufin Tube banks as shown in Fig. 2.15.

Note that the determining factor for which curve is used from Fig. 2.15 is the tube layout - equilateral triangular $(30°)$, rotated square $(45°)$, or inline square $(90°)$. There are small differences among the various pitch ratios (the ratio of the distance between centers of adjacent tubes to the tube outside diameter), but these are secondary to other effects in the normal range (1.2-1.5) used in shell and tube exchangers.

2.3.2. Pressure Drop During Flow Across Banks of Low-Finned Trufin Tubes

The basis for the pressure drop correlation for banks of type S/T & W/H Trufin is essentially the same as for the heat transfer correlation: the Briggs, et al. paper (4) interpretation of the results of Williams and Katz (3) in the light of the Delaware method (5).

The correlating quantity for pressure drop during crossflow is the friction factor defined by Eq. (2.18):

$$f_s = \frac{2\Delta P_s g_c \rho_s G^2{}_m}{4N_c}\left(\frac{\mu}{\mu_w}\right)_s^{0.14} \tag{2.18}$$

where ΔP_s is the pressure drop for flow across a tube bundle, G_m is defined by Eq. (2.16). and N_c is the number of major restrictions crossed by the fluid in one tube bundle. N_c is equal to the number of rows of tubes in the bundle for inline square or equilateral triangular arrays, and equal to one less than the number of rows of tubes in a rotated square $(45°)$ array. The viscosity gradient term is included to account for the non-isothermal effects, ρ_s is the density of the shell-side fluid, and g_c is the gravitational conversion constant.

The friction factor is correlated against the shell side or crossflow Reynolds number defined by Eq. (2.17). Typical results from that study are shown in Fig. 2.16. Unfortunately, the result is not as clear-cut and satisfying for pressure drop as for heat transfer. The basic friction factor curves are found to be about twice the value that would have been predicted from the Delaware work on plain tube banks. However, as can be seen from Fig. 2.16 the Trufin result is quite comparable to that found for the corresponding plain tube bank in the Briggs et al. study (4).

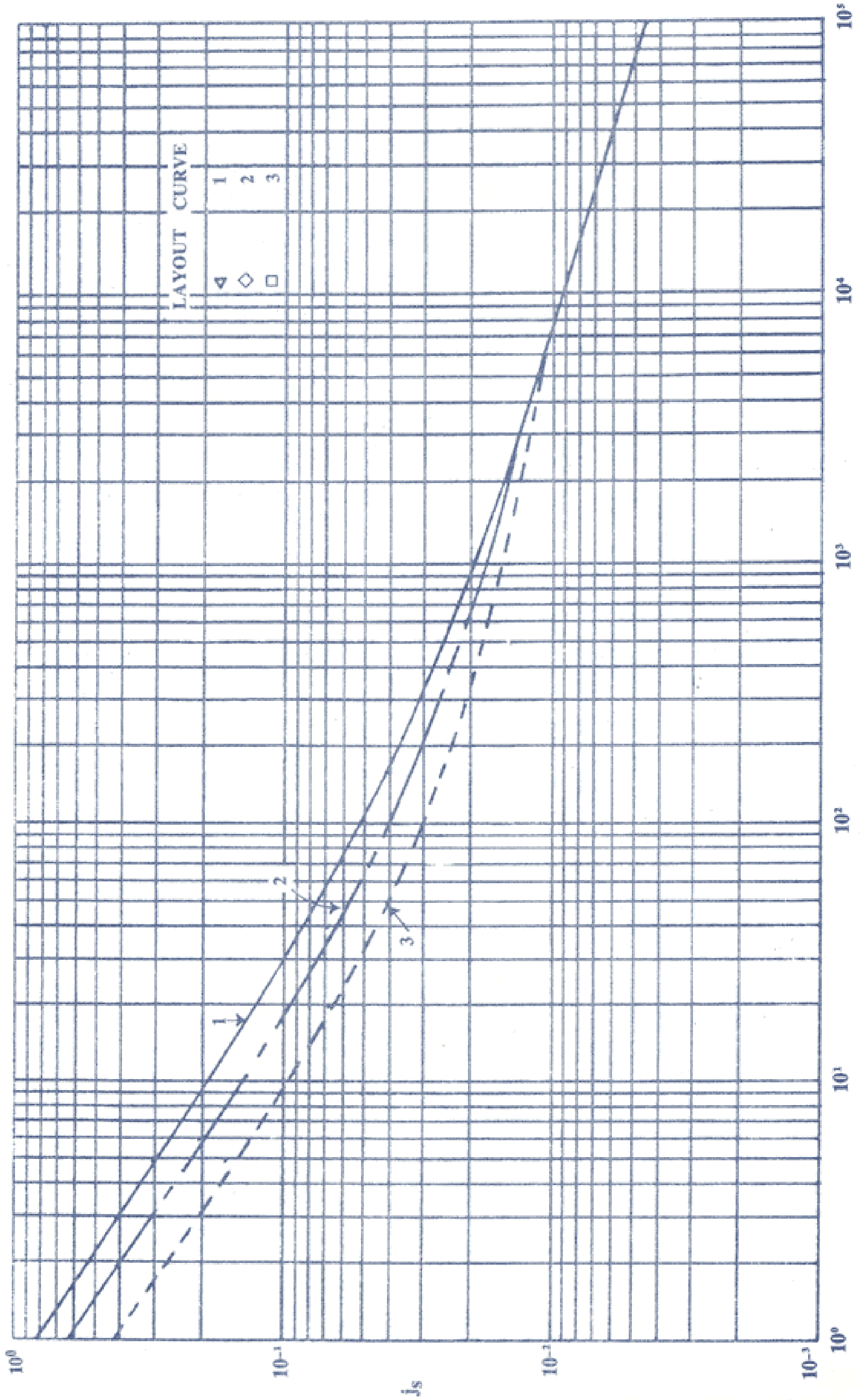

Fig. 2.15 Correlation of j_s for Ideal 19 fin/inch Trufin Tube Banks

The explanation seems to lie in the fact that, in the reduction of the shell and tube exchanger pressure drop data to ideal tube bank data, all of the errors implicit in the correction factors are eventually concentrated in the final result, the f_s curves. In fact, pressure drop prediction is a more uncertain and inaccurate art than heat transfer prediction, and this result should not be surprising. However, it does indicate that care should be taken in developing and using pressure drop correlations and in the interpretation of the result. The correlation that is proposed therefore, is to double the f_s values (at any given Reynolds number) reported from the original Delaware work. These values are shown in Fig. 2.17, when these are used in the Delaware method, the predicted pressure drop should be either about right or on the conservative side, possibly by as much as a factor of 2. In this figure, curves are shown for two different pitch ratios (based upon the outside tube diameter) but there is no dependence upon tube layout. This is speculative, but appears to be consistent with the little information available.

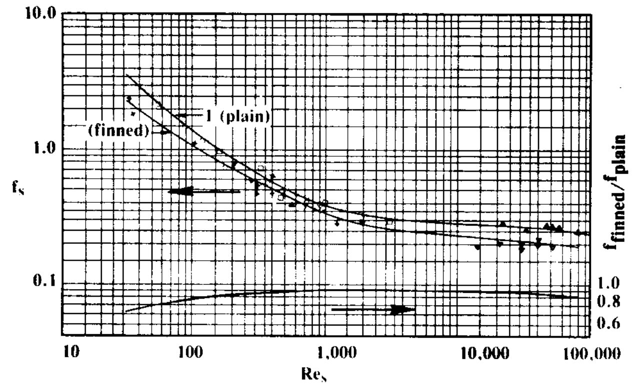

Fig. 2.16 Correlation of the Crossflow Pressure Drop Data for Three Fluids on Shell-Side of Bundles Numbers 1 and 2 (Ref 2)

2.3.3. Effect of Fouling on Trufin

There are few data available on fouling on finned surfaces and even these data are contradictory. On the one hand there is a general belief that Trufin should not be used in severely fouling services because the fouling deposits may find a firmer foothold between the fins than on a plain tube and may close off the surface entirely. However, it is also true that a given weight of fouling material (in moderate quantity) must spread over a larger area and therefore have a proportionately smaller effect on Trufin than on a plain one. Also, it has been found that some brittle fouling deposits are cracked off from Trufin during thermal cycling as the tube expands and contracts.

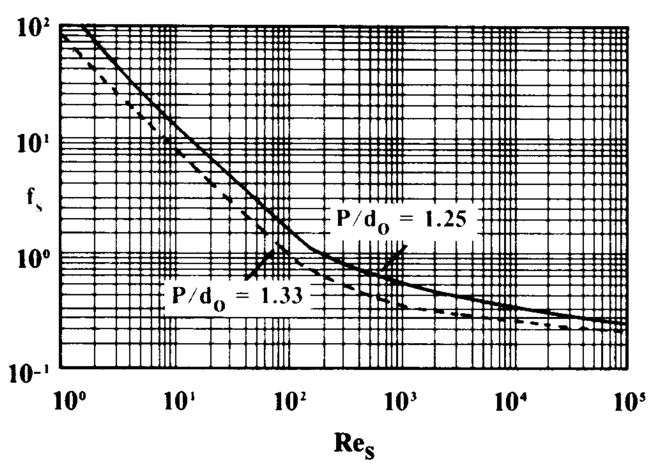

Fig. 2.17 Correlation of f_s vs. Re_s for Ideal 19 fin/inch Trufin Tube Banks.

In sum, the final decision as to whether or not Trufin should be used in a given fouling service must be left to the designer's judgment. In general, the decision would be not to use Trufin in severely fouling service (but in that case the stream in question should probably go inside the tube anyway, to simplify cleaning.) For most services, however, Trufin will likely prove no worse than a plain tube in total deposit accumulated and has the advantage that the penalty against the area required is reduced by the increased ratio of outside (finned) area to inside area, compared to a plain tube.