4.2. Heat Transfer with High-Finned Trufin Tubes

4.2.1. Fin Temperature Distribution and Fin Efficiency

1. Temperature Distribution in Fins. The temperature in a fin is not constant, due to the resistance to conductive heat transfer in the fin metal. A typical temperature profile in a fin is shown in Fig. 4.12.

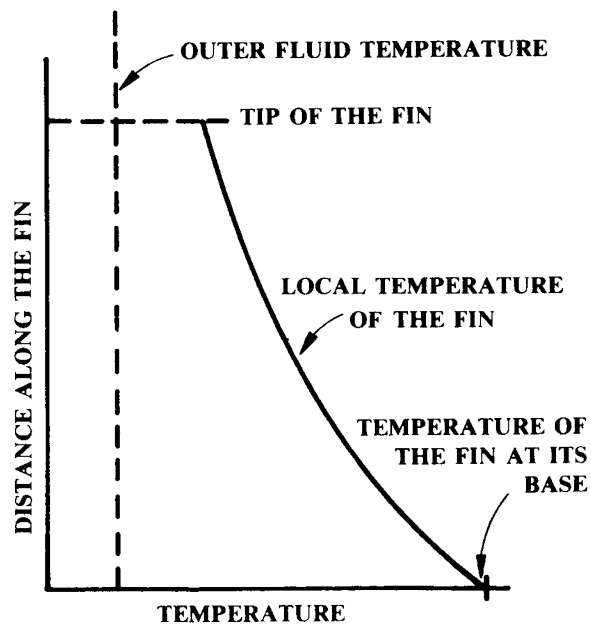

Fig. 4.12 Typical Temperature Distribution in a Fin

The details of calculating the temperature distribution are quite complex and will not be given here; the most comprehensive reference on this subject is the book, "Extended Surface Heat Transfer" by Kern and Kraus (3). The results depend upon a number of parameters, including fin geometry (shape, height, and thickness), fin material, and outside fluid temperature and heat transfer coefficient. It is also necessary to make a number of assumptions; for example, most analyses assume that the outside fluid has a constant bulk temperature and a constant heat transfer coefficient at all points on the fin surface. This is known not to be true, but the real state of affairs is not well understood and would introduce great complexity into the analysis if one tried to be completely rigorous. As a practical matter, the results obtained from the simplified analysis seem to be consistent with experience and lead to acceptable designs.

The subject of fin efficiency was discussed in Section Iand curves for fin efficiency and fin resistance were given for low-finned Trufin. Since the values for high fin were not given, the method of obtaining the values will be repeated.

2. *Fin Efficiency and Resistance*. The fin efficiency, Φ, is the ratio of the total heat transferred from the real fin in a given situation to the total heat that would be transferred if the fin were isothermal at its base temperature. For the kinds of fins that are considered here, a good equation to use over the range of interest is:

$$\Phi = \frac{1}{1 + \frac{m^2}{3}\sqrt{\frac{d_o}{d_r}}} \tag{4.1}$$

where

$$m = H\sqrt{\frac{2}{\left(\frac{1}{h_o} + R_{fo}\right)k_w Y}} \qquad (4.2)$$

Equation (4.1) is actually based upon fins of uniform thickness, whereas the fins on Wolverine high-finned trufin are actually slightly thicker at the base and thinner at the tips. The error is small and in fact the Wolverine fins are slightly more efficient than this equation indicates.

The geometrical variables are defined in Fig. 4.2 and h_o and R_{fo} are respectively the actual convective heat transfer coefficient and the actual fouling resistance on the fin side, based on the fin area. To gain an appreciation of the probable magnitude of Φ in a typical problem, consider the following example:

Type H/R tube, 3003 aluminum:

d_r	=	1.00 in.
d_o	=	1.875 in.
H	=	0.437 in.
s	=	0.076 in.
Y	=	0.015 in.
k_w	=	110 Btu/hr ft^2°F
h_o	=	10 Btu/hr ft^2°F
R	=	0.0

Then:

$$m = \left(\frac{0.437}{12}ft\right)\sqrt{\frac{2}{\left(\frac{1}{10}\frac{hrft^2\,°F}{Btu}\right)\left(110\frac{Btu}{hrft^2\,°F}\right)\left(\frac{0.015}{12}ft\right)}} = 0.439$$

$$\Phi = \frac{1}{1 + \frac{(0.439)^2}{3}\sqrt{\frac{1.875}{1.000}}} = 0.919$$, i.e., 91.9% fin efficiency

There are small differences between the nominal dimensions and the actual dimensions , and some variation from lot to lot in the latter. See Section 6 for details. Nominal dimensions will be used in the examples in this Section.

As we will observe later, this efficiency is, if anything, biased towards the low side of most applications. Cop per fins have a higher thermal conductivity and would give a higher Φ. (Coppernickel (90/10) would give Φ = 0.730 under otherwise identical conditions, but is not commonly used for high-finned tubes.) Thicker fins (our example used the thinnest available) would give higher efficiencies. The film heat transfer coefficient was typical of atmospheric air under nominal operating conditions; an extreme value of 20 Btu/hr ft°F would give Φ = 0.862.

A quantity somewhat more directly useful in design calculations is the "Fin Resistance", R_{fin}, defined as:

$$A_{fin} = \left[\frac{1-\Phi}{\frac{A_{root}}{A_{fin}} + \Phi}\right]\left[\frac{1}{h_o} + R_{fo}\right] \tag{4.3}$$

where A_{root} is the surface area of a unit length of plain(unfinned) tube between the fins and A_{fin} is the heat transfer area of all of the fins on a unit length of tube. Continuing with the example above, we can compute the value of R_{fin} as follows:

$$A_{root} = \pi\left(\frac{1}{12}\ ft\right)(11\ \ fins\ \ per\ inch)(12\ \ inches/ft)\left(\frac{0.076}{12}\right) = 0.219\ \ ft^2/ft\ \ of\ \ length$$

$$A_{fin} = \left\{\frac{\pi}{4}\left[(1.875)^2 - (1)^2\right]\left(\frac{1}{144}\right)ft^2\right\} \times \left\{\left(11\frac{fins}{in.}\right)\left(\frac{2\ \ sides}{fin}\right)\left(\frac{12in}{ft}\right)\right\} = 3.62\ \ ft^2/ft\ \ of\ \ length$$

$$R_{fin} = \left[\frac{1-0.919}{\frac{0.219}{3.62} + 0.919}\right]\left[\frac{1}{10Btu/hr\ \ ft^2\,°F} + 0\right] = 0.00827\frac{hr\ \ ft^2\,°F}{Btu}$$

which corresponds to an effective heat transfer coefficient *for the fins only* of 121 Btu/hr ft^2°F. This may be compared to a typical value of h,, for air-cooled exchangers of 10 Btu/hr ft^2°F, which indicates that the fin resistance is only a small part of the total resistance to heat transfer.

The fin resistance then can be directly incorporated into the equation for the overall heat transfer coefficient as follows:

$$\frac{1}{U_o} = \frac{1}{h_o} + R_{fo} + R_{fin} + \frac{\Delta x}{k_w}\frac{A_o}{A_m} + R_{fi}\left(\frac{A_o}{A_i}\right) + \frac{1}{h_i}\left(\frac{A_o}{A_i}\right) \tag{4.4}$$

The value of R_{fin} may be calculated for any desired case by using Eqns. 4.1, 4.2 and 4.3.

4.2.2. Effect of Fouling on High-Finned Trufin

As a matter of consistency and principle, the analysis to this point has steadfastly incorporated the term R_{fo}, the resistance due to fouling on the finned surface. As a matter of fact, fouling on high-finned Trufin with air on tlie fins is seldom a serious problem, unless there is extensive deposition of material as from massive corrosion (indicating a poor material choice) or a heavy dust storm or ingestion of debris. In the latter cases, continued operation is out of the question, and there is no alternative but to shut down and remove the obstructions. Under normal conditions, the continuous movement of air past the surface tends to minimize deposition of sand and dust, and such deposits as may form can usually be removed by occasionally running a compressed air jet over the sur face. Accordingly, R_{fo} is usually taken as zero for high finned Trufin applications.

4.2.3. Contact Resistance in Bimetallic Tubes

In Type L/C Trufin, there is an internal liner of a metal other than the 3003 aluminum of the outer tube and fins. The two metals will sometimes be in imperfect contact with one another, leading to an additional resistance to the flow of heat. Generally at low temperatures of the metal-to-metal interface, the liner is exerting a positive pressure upon the aluminum finned tube. But as the tube temperature rises, the aluminum expands more rapidly than the liner and a definite gap develops. The gap is filled with air, introducing a substantial additional resistance to the flow of heat.

There have been several studies, both experimental and analytical, made of this problem and the results have been surveyed by Kulkarni and Young (4). This paper and its references should be consulted for details and predictive methods, but it is desirable to summarize here the main findings:

1. At the fabrication temperature of approximately 70°F, there is a positive contact pressure of about 400 psi for a stainless steel liner inside aluminum. Presumably a similar value would exist for other liner metals.

2. This results in a contact resistance of about 0.00005 hrft^2°F/Btu, based upon the contact surface. This is negligible for any practical application.

3. At the point of zero contact pressure (which occurs at a bond temperature of about 200-215°F in the steel/aluminurn case), the bond resistance has been measured to be about 0.0002 hrft^2°F/Btu. This is still negligible for most applications.

4. At tube side fluid temperatures of 1000°F and air side temperatures of 200°F, the bond resistance is computed to increase to values as high as 0.003 hrft^2°F/Btu (based on contact area) at air side coefficients of 5 Btu/hrft^2°F (based on fin area) and 0.002 hrft^2°F/Btu for air side coefficients of 10 Btu/hrft^2°F. When the corresponding area ratios (say between 1: 10 and 1:20) are taken into account, bond resistance is seen to be about 10-25 percent of the total resistance to heat transfer and definitely needs to be considered in the design. However, it would not seem that a very detailed calculation of the effect is in order unless many such high temperature cases are to be handled.

   The complete formulation of the overall heat transfer coefficient calculation for the bimetallic tube with contact resistance is then:

$$U_0 = \frac{1}{\frac{1}{h_o} + R_{fo} + R_{fin} + \left(\frac{\Delta x_w}{k_w}\frac{A_o}{A_m}\right) + R_b\left(\frac{A_o}{A_b}\right) + \left(\frac{\Delta x_w}{k_w}\frac{A_o}{A_m}\right)_2 + R_{fi}\left(\frac{A_o}{A_i}\right) + \frac{1}{h_i}\left(\frac{A_o}{A_i}\right)} \quad (4.5)$$

   where $\left(\frac{\Delta x_w A_o}{k_w A_m}\right)_1$ is the wall resistance for the fin metal root, R_b is the bond resistance based upon the bond contact area A_b, $\left(\frac{\Delta x_w A_o}{k_w A_m}\right)_2$ is the wall resistance for the liner tube, and the other terms have their usual meaning.