



WOLVERINE TUBE HEAT TRANSFER DATA BOOK

1.5. Application of Extended Surfaces to Heat Exchangers

1.5.1. The Concept of the Controlling Resistance

Examination of Eqs. (1.18), (1.19), or (1.20) shows that, if any one of the terms in the denominator on the right hand side is substantially larger than the others, it essentially fixes the value of U . Since each of these terms is a resistance to heat transfer, the term that dominates is called the "controlling resistance," and the designer needs to devote particular attention to its value.

As noted in the previous section, one heat transfer coefficient may be much lower than the other, and this coefficient is the one that leads to a high resistance for heat transfer. Thus, a low coefficient corresponds to a high, and frequently controlling, resistance.

One way to minimize the adverse effects of a low heat transfer coefficient is to place that stream on the shell side of a heat exchanger and use extended surface to increase the ratio of the outside to inside area (A_o/A_i) to offset the low value of h_o .

1.5.2. Types of Extended Surface

"Extended surface" is a term that covers many possibilities but the type that we shall be most concerned with is the round tube with round fins essentially transverse to the tube axis. Typical integral finned tubes are shown in Fig. 1.48.

The low-finned tube commonly used in shell and tube exchangers provides about 3 to 4 times as much outside area as inside; i.e., A_o/A_i is about 3 to 4. In the Wolverine S/T type, fin counts of 16 to 40 per inch are available. The outside diameter of the fins is just slightly less than that of the bare tube at the ends, so the tube can be inserted through the tube sheet holes. The wall under the fins is controlled to a specified thickness, and the wall at the plain ends is about two gauges heavier. For tube support at baffles, unfinned sections ("lands") are available.

A medium-fin height tube with 11 fins per inch is also used in shell and tube exchangers, and has an A_o/A_i ratio typically about 5.

The high-fin tube is used to advantage when gases are to be heated or cooled or when a process stream is to be air-cooled. High-fin tubes come in a wide variety of fin heights, thicknesses, and spacings, giving values of A_o/A_i up to 25. For corrosion protection, a mechanically bonded liner tube may be used inside the finned tube. The liner can be made of corrosion-resistant alloy, while the outer tube and fins are made of a high conductivity metal such as copper or aluminum to improve heat transfer.

All of the finned tubes produced by Wolverine have integral fins, i.e., the fins are formed from the base tube by an extrusion process so that the final tube and its fins are one piece of metal.

For high-finned tubes, other fin types exist in which the fin is a separate piece of metal wrapped on the tube. The fin is usually held in place only by the tension of the deformed fin metal or occasionally by soldering, welding or brazing the fin to the tube. In these finned tubes, contact resistance between the fin and the tube is an important consideration. Even if the fin is originally in good thermal contact, repeated thermal cycling can result in creep and partial separation of the

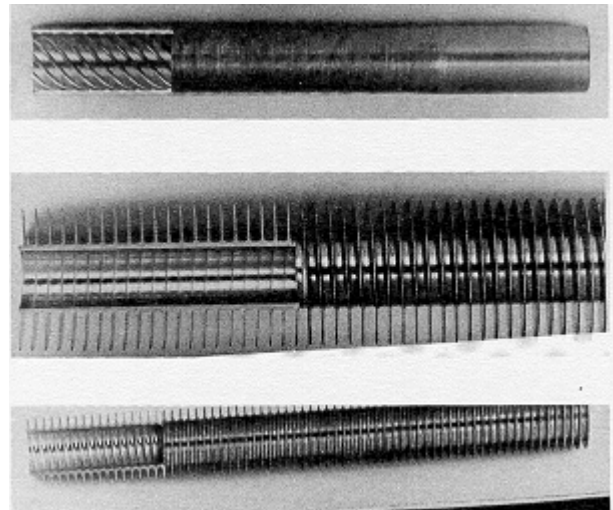


Fig 1.48. Several types of Wolverine Trufin Tubing



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fin. Also, the crevice between fin and tube provides a possible site for corrosion with consequent increase in contact resistance and possibly accelerated failure of the tube.

1.5.3. Fin Efficiencies and Related Concepts

Consideration of the heat flow path from the tube-side fluid through the tube wall, and fins, and into the fin side fluid reveals that the distance which the heat must flow' is longer than in the corresponding plain tube case. It is true that the additional distance is through the usually highly conductive fin metal, but there is still an additional resistance to the flow of heat, which partially offsets the advantage of the extended surface. In order to take account of this resistance in our calculations, we define a "fin efficiency," ϕ , in the following manner: ϕ is the ratio of the total heat transferred from the actual fin to the total heat that would be transferred if the fin were isothermal at its base temperature. This definition is illustrated by Fig. 1.49. In this figure, the fin temperature T_f is higher than the bulk outer fluid temperature, T_o , so heat flows from the fin to the outer fluid. All the heat thus transferred however must flow into the fin through the base or root of the fin. Therefore, the temperature at the base of the fin, T_r , is the highest temperature in the fin. The temperature decreases continuously from that point as one proceeds outward in the fin in order to cause the heat to flow from the base to the fin. The rate of decrease depends upon the shape and thermal conductivity of the fin and the film heat transfer coefficient from the fin to the outer fluid.

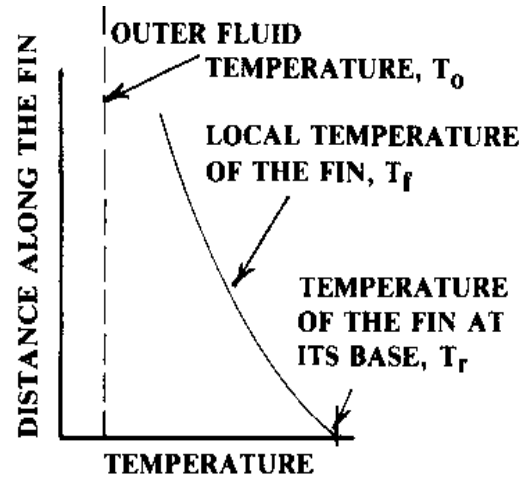


Fig. 1.49 Diagram of Fin Temperature Profile.

It is clear that the maximum amount of heat would be transferred from the fin if the fin were everywhere at T_r , and that could only happen if the fin metal had infinite thermal conductivity. No metal has infinite thermal conductivity, of course, but this makes a convenient reference point against which real fins can be compared, as in the above definition of fin efficiency.

A complex mathematical analysis is necessary to obtain a solution for the fin efficiency, and this will not be carried out here. A very complete discussion of the problem and its solutions is given in the book, "Extended Surface Heat Transfer", by Kern and Kraus (12).

For the kinds of fins that are considered here, a good equation to use over most of the range of interest is:

$$\Phi = \frac{1}{1 + \frac{m^2}{3} \sqrt{\frac{d_o}{d_r}}} \quad (1.33)$$

where

$$m = H \sqrt{\frac{2}{\left(\frac{1}{h_o} + R_{fo}\right)ky}} \quad (1.34)$$



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The geometrical variables are defined in Fig. 1.50, 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.

An inspection of Eqs. (1.33) and (1.34) indicates that the fin efficiency is higher for the better-conducting fin materials, and higher as the outside heat transfer coefficient is lower. Finned tubes are not generally used in applications where the fin efficiency is less than 0.65.

The fin efficiency is applicable only to the fin area of the tube. It is assumed that heat is transferred from the root area of the tube (between the fins) with an efficiency of 1.00. Thus, we may write the equation for the total heat transfer from the tube as:

$$Q = h_o A_{root} (T_r - T_o) + h_o A_{fin} (T_r - T_o) \Phi \quad (1.35)$$

where A_{root} refers to the heat transfer area of the root portions between the fins and A_{fin} to the heat transfer area of all the fins on the tube. This equation may be written as:

$$Q = h_o (T_r - T_o) (A_{root} + \Phi A_{fin}) \quad (1.36)$$

or

$$Q = h_o (T_r - T_o) A_{eq} \quad (1.37)$$

where A_{eq} , the equivalent or effective heat transfer area, is

$$A_{eq} = A_{root} + \Phi A_{fin} \quad (1.38)$$

If the outside of the tube (including the fins) has a fouling film on it, Eqs. (1.35), (1.36) and (1.37) must be written:

$$Q = \left(\frac{1}{\frac{1}{h_o} + R_{fo}} \right) A_{root} (T_r - T_o) + \left(\frac{1}{\frac{1}{h_o} + R_{fo}} \right) A_{fin} (T_r - T_o) \Phi \quad (1.39)$$

or

$$Q = \left(\frac{1}{\frac{1}{h_o} + R_{fo}} \right) (T_r - T_o) (A_{root} + \Phi A_{fin}) \quad (1.40)$$

or

$$Q = \left(\frac{1}{\frac{1}{h_o} + R_{fo}} \right) (T_r - T_o) A_{eq} \quad (1.41)$$

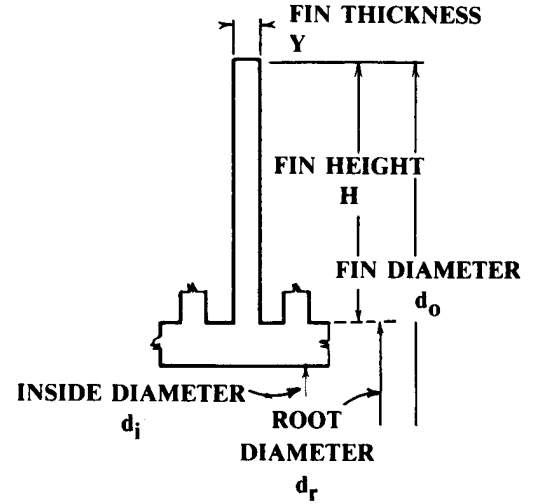


Fig. 1.50 Geometrical Variables for Fin



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and Eq. (1.38) is unchanged.

A_{eq} may be divided by the total outside heat transfer area of the tube A_o to give a weighted fin efficiency.

$$E_w = \frac{A_{root} + \Phi A_{fin}}{A_{root} + A_{fin}} \quad (1.42)$$

and values of this ratio can be calculated for a given tube as a function of material and $\left(\frac{1}{h_o} + R_{fo}\right)$ once and for all. This has been done in Figures 1.51 a, b, c for the Wolverine Trufin Products.

1.5.4. The Fin Resistance Method

Another way of representing the heat transfer effectiveness of a finned tube is by the "fin resistance method," as first developed by Carrier and Anderson (13) and further modified by Young and Ward (14). This is the method that will be generally used in the remainder of this Manual. The line of argument that led to Eq. (1.16) was based upon a plain tube in which all of the outer heat transfer was taken to be equivalent in its ability to transfer heat. The validity of that equation may be extended to finned tubes if we rewrite the equation introducing an additional resistance due to heat conduction through the fin, R_{fin} :

$$Q = \frac{T_i - T_o}{\frac{1}{h_i A_i} + \frac{R_{fi}}{A_i} + \frac{\Delta x_w}{k_w A_m} + \frac{R_{fin}}{A_o} + \frac{R_{fo}}{A_o} + \frac{1}{h_o A_o}} \quad (1.43)$$

The third term in the denominator, referring to the wall resistance, is evaluated for that portion of the tube wall that lies between the inside diameter and the root diameter, i.e., exclusive of the fins. Thus

$$\Delta x = 1/2(d_r - d_i) \quad (1.44)$$

and

$$A_m \approx \pi(d_i + \Delta x)L \quad (1.45)$$

We now define an overall heat transfer coefficient U_o based on the entire outer surface of the finned tube, A_o :

$$Q = U_o A_o (T_i - T_o) \quad (1.46)$$

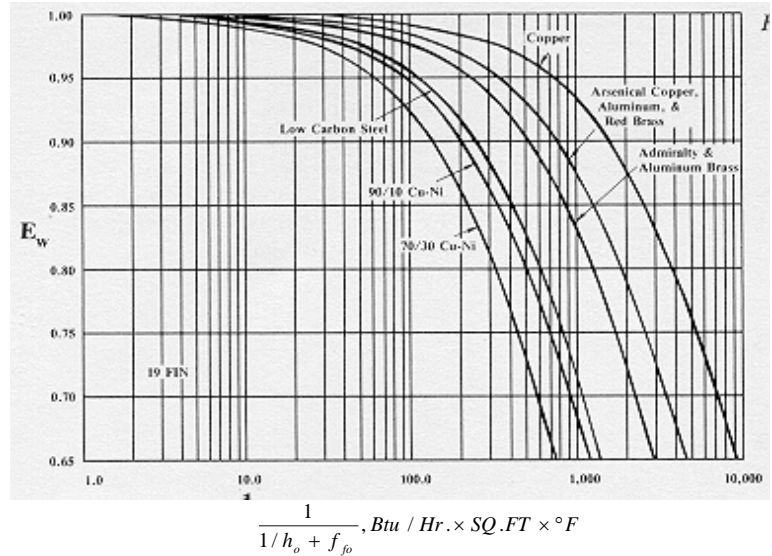


Fig. 1.51a Weighted Fin Efficiency of 19 Fin Type S/T Trufin

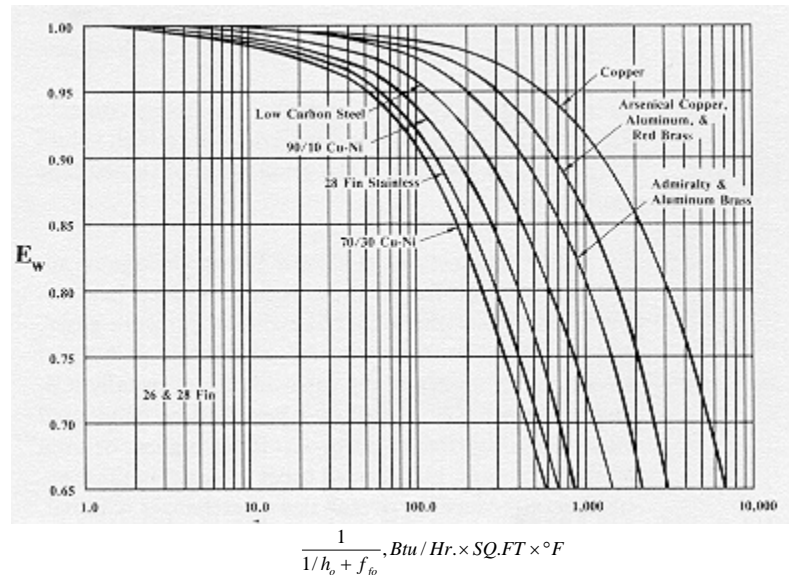


Fig 1.51b. Weighted Fin Efficiencies of 26 and 28 Fin Type S/T Trufin



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Eliminating Q and $(T_i - T_o)$ between Eqs. (1.43) and (1.46) and rearranging, we have:

$$\frac{1}{U_o} = \frac{1}{h_o} + R_{fo} + R_{fin} + \frac{\Delta x A_o}{k_w A_m} + R_{fi} \left(\frac{A_o}{A_i} \right) + \frac{1}{h_i} \left(\frac{A_o}{A_i} \right) \quad (1.47)$$

As written, each term on the right-hand-side is a resistance to heat transfer for the particular process involved; the temperature drop across each part of the process is proportional to the magnitude of the corresponding resistance.

If we take the first three terms on the right side of Eq. (1.47), the corresponding temperature drop is that from the root to the external fluid, $(T_r - T_o)$, and the corresponding heat flow is

$$Q = \frac{(T_r - T_o) A_o}{\left(\frac{1}{h_o} + R_{fo} + R_{fin} \right)} \quad (1.48)$$

This may be equated to Eq. (1.40) which uses the fin efficiency Φ , but which must give the same heat flow. With some rearrangement

$$R_{fin} = \left[\frac{1 - \Phi}{\frac{A_{root}}{A_{fin}} + \Phi} \right] \left[\frac{1}{h_o} + R_{fo} \right] \quad (1.49)$$

Using Eqs. (1.33, 1.34) to evaluate Φ , R_{fin}

may be calculated as a function of $\left(\frac{1}{h_o} + R_{fo} \right)$

for various fin materials and geometries: results are given in Figures 1.52a, b, c for the standard Wolverine Trufin Products.

An important feature of R_{fin} is that it is nearly constant for a given fin geometry and material out to high values of the abscissa (relative to the usual range of finned tube application.)

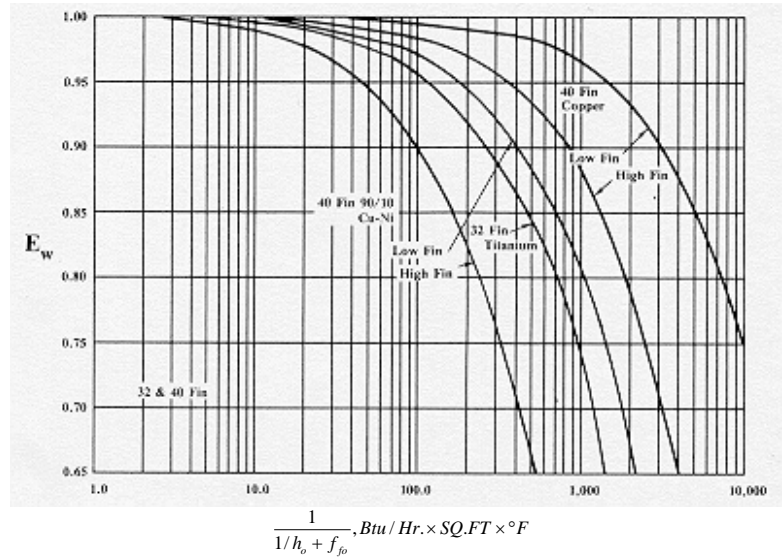


Fig.1.51c Weighted Fin Efficiencies of 32 & 40 Fin Type S/T Trufin

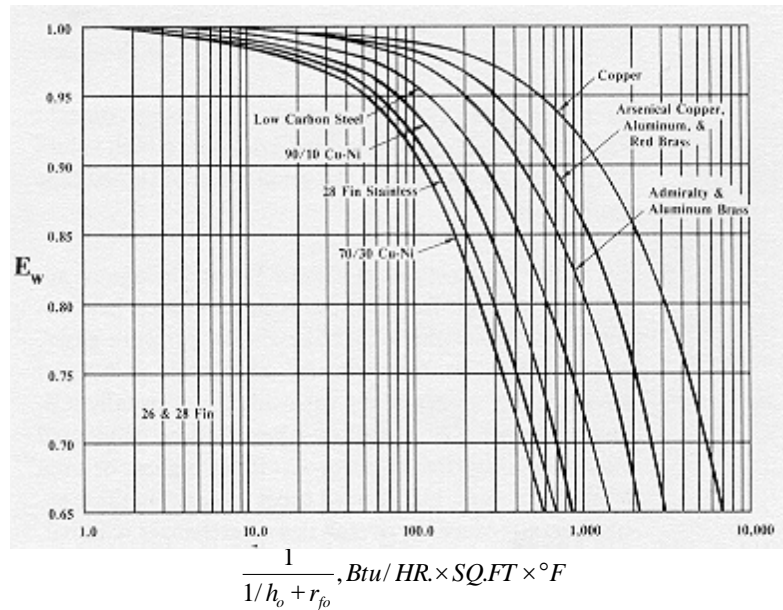


Fig. 1.52a Fin Resistance of 19 Fin Type S/T Trufin



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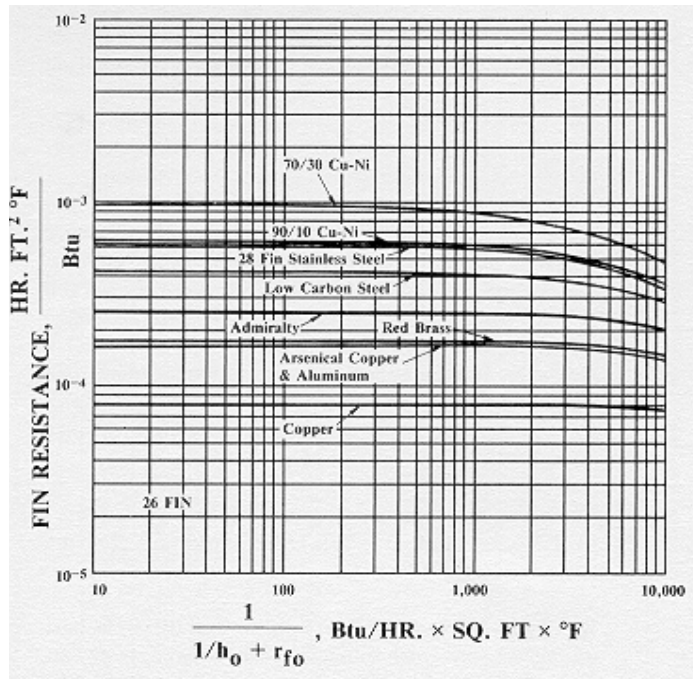


Fig.1.52b Fin Resistance of 26 & 27 Fin Type S/T Trufin

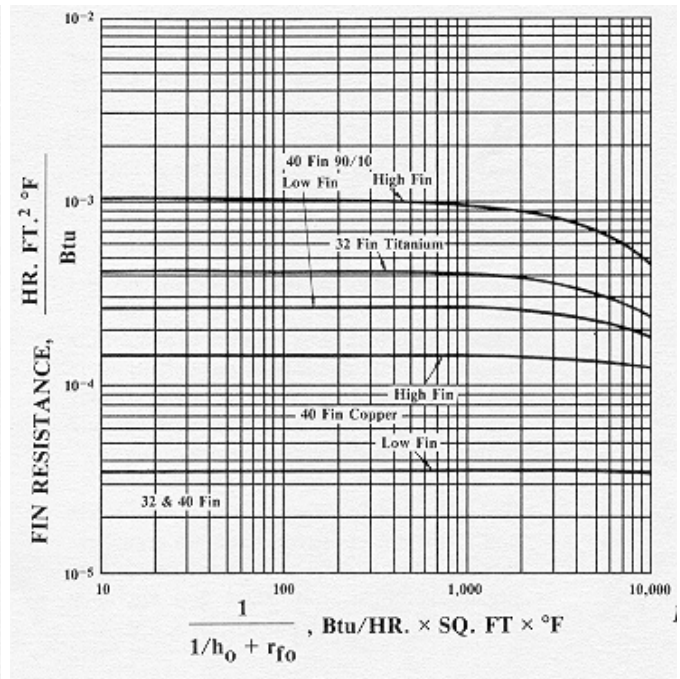


Fig. 1.52c Fin Resistance of 32 & 40 Fin Type S/T Trufin

1.5.5. Some Applications of Finned Tubes.

One major application of high-finned tubes is in air-cooled heat exchangers. Atmospheric air, like all low pressure gases, gives a very low heat transfer coefficient at normal velocities. By contrast, the tube-side fluid, usually a liquid to be sensibly cooled or a vapor to be condensed may have a coefficient up to 100 times higher, or even more. Therefore, high-finned tubes are used in these exchangers to reduce the overall size of exchanger required. Even so, some of these installations cover several acres. Construction details and design methods for this kind of heat exchanger are given in Chapter 4 of this manual.

Low- and medium-finned tubes are used in a variety of sensible, condensing, and boiling services in shell and tube exchangers. A typical sensible heat transfer application would be cooling a compressed gas in a compressor inter-cooler, using cooling water in the tubes.

Low-finned tubes are used for condensing organic vapors, which have condensing coefficients only a third or a quarter of that of the cooling water inside the tubes. In addition to providing additional heat transfer area, the fins provide drip points that facilitate the drainage of the condensate. On the other hand, finned tubes are not used for condensing steam or other high surface tension fluids on the fin side. The high surface tension causes the liquid to hang on the surface, largely insulating it by a static film of liquid.

Finned tubes are also used in boiling services, especially when condensing steam is the heating medium inside the tubes. The condensing coefficient in this case may be 2000 or 3000 $Btu/hr.ft^2.F$, so even the high heat transfer coefficients commonly associated with nucleate boiling may be relatively small by comparison, and the design can benefit by the use of low-finned tubes.