

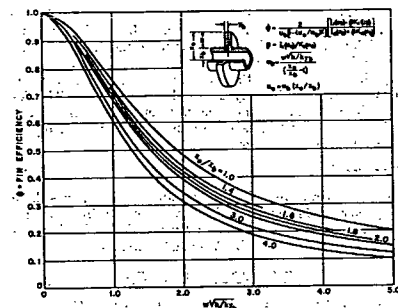


Fig. 15 Efficiency of Annular Fins of Constant Thickness

Assuming that the heat transfer coefficients for the finned surface and prime surface are equal, a surface efficiency, ϕ_s , can be derived for use in Equation 21.

$$\phi_s = 1 - \frac{A_f}{A} (1 - \eta) \quad (20)$$

$$q = \phi_s A (t_s - t_f) \quad (21)$$

where A is the total surface area, equal to the sum of the finned and prime areas.

The temperature distribution and fin efficiencies for various fin shapes have been derived by Harper and Brown,¹¹ Schmidt,¹² Gardner,¹³ and others. A survey of optimum fin dimensions is given by Jakob.¹⁴ Figs. 15 to 18 present the curves and equations of Gardner for annular fins, straight fins, and spines. For constant thickness square fins, it is recommended that the efficiency of a constant thickness annular fin of the same area be used.¹⁵ More accurate results, particularly in the case of rectangular fins of large aspect ratio,

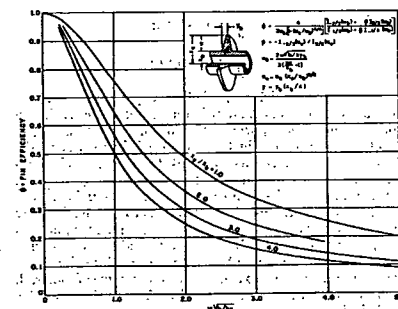


Fig. 16 Efficiency of Annular Fins of Constant Heat Flow Area

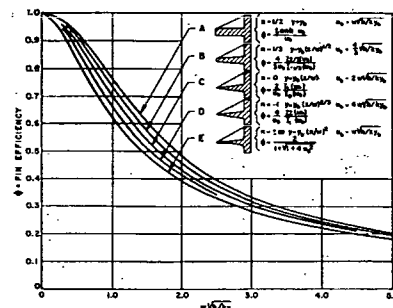


Fig. 17 Efficiency of Several Types of Straight Fin

can be obtained by dividing the fin into circular sectors as described by Carrier and Anderson.¹⁶

Various methods are used to fabricate finned surface. In some cases fins are extruded from the prime surface—for example, the short fins found on the tubes in flooded evaporators or water-cooled condensers. In other cases the fins are fabricated separately, sometimes of a different material, and bonded to the prime surface. Metallurgical bonds are achieved by furnace brazing, dip-brazing, or soldering. Mechanical bonds are obtained by tension winding fins around tubes (spiral fins) or expanding the tubes into the fins (plate fin). Metallurgical bonding, if properly done, leaves negligible thermal resistance at the joint. It is not always economical, however. The thermal resistance of a mechanical bond may or may not be negligible depending upon the application, the care taken in the manufacture and the materials and temperatures involved.¹⁷ For plate fin coils having expanded tubes, tests by Dart¹⁸ showed that substantial losses in performance can be obtained with fins having cracked collars.

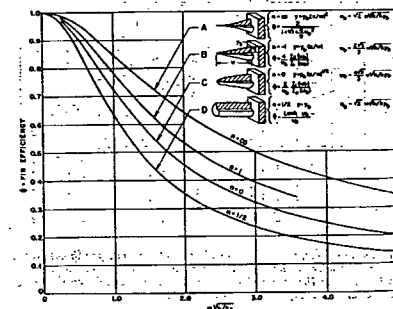


Fig. 18 Efficiency of Four Types of Spine

On the other hand, negligible thermal resistance was found in coils with continuous collars and properly expanded tubes.

Finned-Tube Heat Transfer

The heat transfer coefficients for finned coils follow the basic equations of convection, condensation and evaporation. However, the particular arrangement of the fins used appears to affect to some extent the values of constants and the exponential powers in the equations. It is generally necessary to refer to test data for the exact coefficients.

For natural convection finned coils (sometimes called gravity coils) approximate coefficients can be obtained by considering the coil to be made up of tubular and vertical fin surfaces at different temperatures, and then applying the natural convection equations to each. This is made difficult by the fact that the natural convection coefficient depends upon the temperature difference, which varies at different points of the fin.

With natural convection, fins generally need a high fin efficiency, (80 to 95 percent) for optimum heat transfer. A low fin efficiency reduces the temperature near the tip. This not only reduces the Δt near the tip, but also cuts down the coefficient, h , which in natural convection depends on Δt . Likewise, the coefficient of heat transfer decreases as the number of fins per inch increases, due to the increased interference of convection currents from the adjacent fins and reduced free-flow passage. Two to four fins per inch are common. In general, high coefficients result from large temperature differences and small flow restriction.

Very little published information is available for heat transfer rates in natural convection finned coils. Chaddock and Edwards¹⁹ measured coefficients for a number of circular fin-on-tube arrangements. Using the fin spacing, δ , as the characteristic length they were able to correlate their data in the form $Nu_{\delta} = f(N_{Gr} \delta / D_o)$, where D_o is the fin diameter. Data for free convection and radiation from wire and tube heat exchangers is given in References 85, 86, and 87.

Forced convection finned coils are used extensively in a wide variety of equipment. The fin efficiency for optimum performance is smaller than for gravity coils, since the forced-convection coefficient is almost independent of the temperature difference between the surface and the fluid. However, very low fin efficiencies should be avoided, since the inefficient surface gives high ratio of pressure drop to heat transfer. An efficiency of 70 to 90 percent is often used.

As the number of fins per inch of tube length is increased to obtain a large surface area for heat transfer, the coefficient generally increases somewhat. This is due to larger air velocity between the fins for the same face velocity, and also due to reduced equivalent diameter. However, a point is reached when the boundary layer formed on one fin surface (Fig. 2) begins to interfere with the boundary layer formed on the adjacent fin surface. The interference of boundary layers results in a decrease of the heat transfer coefficient which may offset the advantage gained from larger surface area.

The selection of the number of fins per inch for forced convection finned coils usually depends upon economic and practical considerations, such as fouling, frost formation, condensate drainage, cost, weight, and volume. For conventional coils, it is generally not desirable to exceed 13 to 14 fins per inch. Forced-air coils are seldom designed with less than six fins per inch, except in the case of units where other factors, such as frost formation, necessitate fewer fins per inch.

A number of means have been used to obtain higher coefficients

with a given air velocity and surface. The most common consists of creating air turbulence to increase the heat transfer coefficient; although this is usually accompanied by higher pressure drop. Some of the conventional means of causing air turbulence are: (1) staggered tubes instead of in-line tubes for multiple-row coils, (2) artificial additional tubes, or collars or fingers made by suitably forming the fin materials, (3) corrugated fins instead of plane fins, and (4) louvered or interrupted fins.

Heat transfer data for various types of finned coils will be found in the following references: London and Kays,²⁰ Shepherd²¹ for one-row plate fin coils; Ghai²² for straight fins; Kats²³ for disk and spiral fins; Jameson²⁴ for circular fins; Gunter²⁵ for pressure drop with various types of surface; and Belling²⁶ and Clark²⁷ for fins inside tubes.

Some of the data of Shepherd for one-row coils is shown in Fig. 19. The thermal resistances plotted in Fig. 19 include the temperature drop through the fins, and are based on one square foot of total external surface area.

In the absence of exact data, dimensionless heat transfer and pressure drop correlations given by Kern²⁸ for circular fins, double pipe longitudinal finned-surface exchangers, and longitudinal fins inside tubes of shell-and-tube exchangers may be used.

Finned surfaces for condensing or boiling are used in water-cooled shell-and-tube units having circular fins outside the

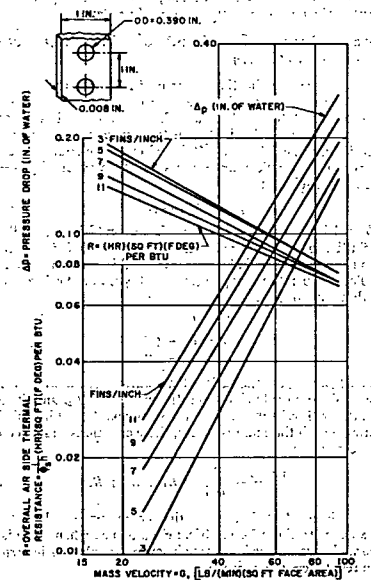


Fig. 19 Overall Air-Side Thermal Resistance and Pressure Drop for 1-Row Coils