THERMAL SOLUTIONS

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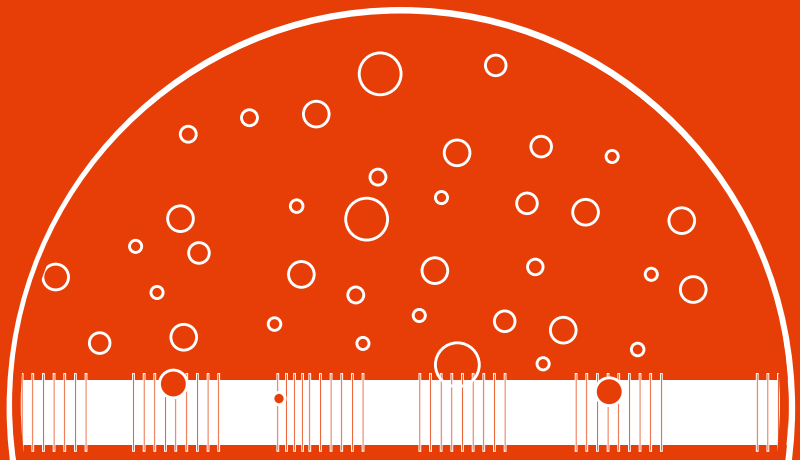

THE HEAT TRANSFER DATA BOOK II

Fundamental heat transfer and application know-how

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Acknowledgement is made in the text for material which has been used from other sources.

PREFACE

For many years now, the Heat Transfer Data Book II has been a valuable reference in all work and research in heat transfer engineering. The manual provides information and discussions on basic heat transfer, sensible heat transfer, condensing heat transfer, Trufin tubes in air-cool heat exchangers, and Trufin tubes in boiling heat transfer.

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CHAPTER 1 – BASIC HEAT TRANSFER

1.1 BASIC MECHANISMS OF HEAT TRANSFER

The basic mechanisms of heat transfer are generally considered to be conduction, convection, boiling, condensation, and radiation. Of these, radiation is usually significant only at temperatures higher than those ordinarily encountered in tubular process heat transfer equipment; therefore, radiation will not be considered in any great detail in this Manual. All of the others play a vital role in equipment design and will frequently appear in the discussion. In this section, the emphasis will be upon a qualitative description of the processes and a few very basic equations.

1.1.1 CONDUCTION

Mechanism

Conduction in a metallic solid is largely due to the random movement of electrons through the metal. The electrons in the hot part of the solid have a higher kinetic energy than those in the cold part and give up some of this kinetic energy to the cold atoms, thus resulting in a transfer of heat from the hot surface to the cold. Since the free electrons are also responsible for the conduction of an electrical current throwugh a metal, there is a qualitative similarity between the ability of a metal to conduct heat and to conduct electricity. In addition, some heat is transferred by interatomic vibrations.

Fourier Equation

The details of conduction are quite complicated but for engineering purposes may be handled by a simple equation, usually called Fourier's equation. For the steady flow of heat across a plane wall (Fig. 1.1) with the surfaces at temperatures of T_1, and T_2 where T_1 is greater than T_2 the heat flow Q per unit area of surface A (the heat flux) is:

1.1) $$\frac{Q}{A} = q = k\left(\frac{T_1 - T_2}{X_1 - X_2}\right) = k\frac{\Delta T}{\Delta X}$$

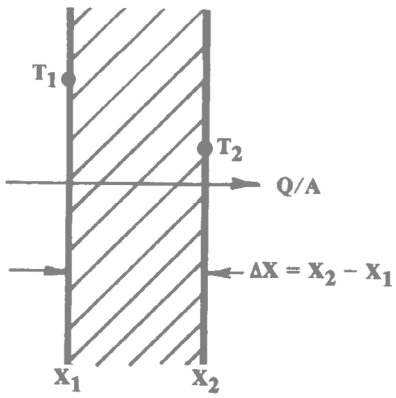

Fig. 1.1 Diagram of Conduction Through a Plane Wall

The quantity k is called the thermal conductivity and is an experimentally measured value for any material. Eq. (1.1) can be written in a more general form if the temperature gradient term is written as a differential:

(1.2) $$\frac{Q}{A} = -k\frac{dT}{dx}$$

The negative sign in the equation is introduced to account for the fact that heat is conducted from a high temperature to a low temperature, so that (dT/dx) inherently negative; therefore the double negative indicates a positive flow of heat in the direction of decreasing temperature.

Conduction Through a Tube Wall

The main advantage of Eq. (1.2) is that it can be integrated for those cases in which the cross sectional area for heat transfer changes along the conduction path. A section of tube wall is shown in Fig. 1.2. Q is the total heat conducted through the tube wall per unit time. At the radial position r in the tube wall ($r_i \leq r \leq r_Q$) , the area for heat transfer for a tube of length L is A = 2 rL. Putting these into Eq. (1.2) gives

(1.3) $$\frac{Q}{2\pi rL} = -k\frac{dT}{dx}$$

which may be integrated to

(1.4) $$Q = \frac{2\pi L\ k(T_i - T_o)}{\ln(r_o/r_i)}$$

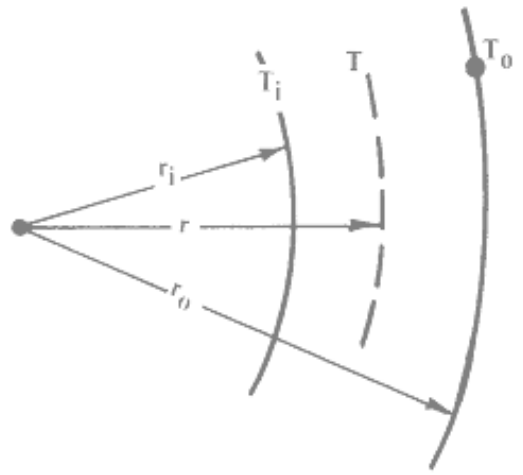

Fig. 1.2 Diagram of Conduction Through a Cylindrical Wall

If $T_i < T_o$, Q comes out negative; this just means that the heat flow is inward, reversed from the sense in which we took it. For thin walled tubes, the ratio of the outer to the inner radius is close to unity, and we can use the simpler equation,

(1.5) $$Q = \frac{2\pi r_o L\ k(T_i - T_o)}{r_o - r_i}$$

with very small error.

Conduction Through a Bimetallic Wall

Sometimes, for reasons of corrosion, strength and/or economy, a tube is actually constructed out of two tightly fitting concentric cylinders of different metals as shown in Fig. 1.3. Note that r' is the outside radius of the inner tube and the inside radius of the outer tube, and T' is the corresponding temperature. From Eq. 1.4 we may write directly for the inner tube:

(1.6) $$Q = \frac{2\pi L\ k_i(T_i - T')}{\ln(r'/r_i)}$$

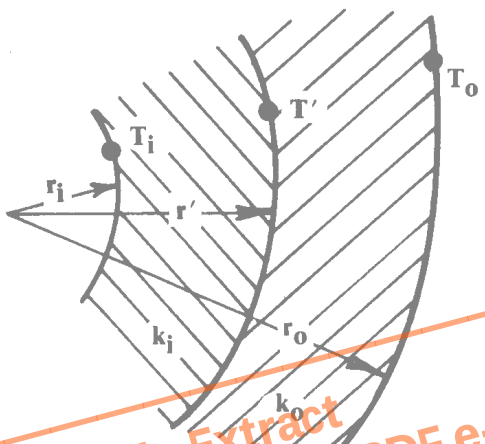

Fig. 1.3 Diagram of Conduction Through Two Cylindrical Walls in Contact

and for the outer tube:

(1.7) $$Q = \frac{2\pi L\, k_o (T' - T_o)}{\ln(r_o / r')}$$

Since the same amount of heat must flow through both tubes, the Q's are equal. Then the equations can be combined to eliminate the unknown temperature T', and Q is given by:

(1.8) $$Q = \frac{T_i - T_o}{\dfrac{\ln(r'/r_i)}{2\pi L\, k_i} + \dfrac{\ln(r_o / r')}{2\pi L\, k_o}}$$

Contact Resistance

In the previous section, the assumption was made that the outer surface of the inner cylinder and the inner surface of the outer cylinder were at the same temperature, implying that there was no resistance to heat transfer between the two. This assumption is essentially correct if the two surfaces are metallurgically bonded to one another, as can be achieved when one material is fused to the other or when they are bonded by a detonation wave. The assumption can be seriously in error if the two surfaces are merely in close physical contact, even at the very high pressures that can be exerted by shrink-fitting.

Practical metal surfaces generally have roughnesses ranging from 10 to 180 microinches, the degree and the form of the roughness depending upon the metal and the method of forming the surface. When two such surfaces are placed in contact, the "hills" are touching while the "valleys" are filled with the ambient atmosphere, usually air. Because of the low thermal conductivity of gases, practically all of the heat is conducted through the points in metal-to-metal contact. At low pressures, this will be only a small portion of the surface perhaps less than one percent and the resulting constriction of the heat flow lines can lead to an interface resistance several times greater than in the metal slabs themselves.

At higher contact pressures between the surfaces, the hills are flattened to give a greater surface area in contact in order to sustain the load, and the interface resistance decreases. Various methods are used to ensure good thermal contact including co-extrusions and shrink-fitting, but repeated thermal cycling in the normal operation of process equipment, together with creep, can cause long-term serious loss of efficiency.

Unfortunately it is almost impossible to predict contact resistance in process equipment applications. Here, as in all process equipment design, the engineer must assess the consequences of being wrong and the possible alternatives.

1.1.2 SINGLE PHASE CONVECTION

Fluid Motion Near a Surface

Convective heat transfer is closely connected to the mechanism of fluid flow near a surface, so the first matter of importance is to describe this flow.

CHAPTER 2 – SENSIBLE HEAT TRANSFER

2.1 HEAT EXCHANGERS WITH LOW AND MEDIUM FINNED TRUFIN

2.1.1 AREAS OF APPLICATION

In Chapter 1, we found that it is usually advantageous to use Trufin Tubes when one of the film heat transfer coefficients is significantly smaller than the other. The lower coefficient tends to dominate or control the magnitude of U, the overall heat transfer coefficient, resulting in a large required heat transfer area and a correspondingly large heat exchanger. We also showed that one can reduce the total length of tubing required and therefore the size of the heat exchanger if finned surface is used in contact with the fluid having the low film heat transfer coefficient.

An approximately optimum design can be obtained under these conditions if the resistances of the two sensible heat transfer processes are approximately equal. This requirement may be stated as

$$\frac{1}{h_i A_i} \approx \frac{1}{h_o A_o} \quad \text{or} \quad \frac{A_o}{A_i} \approx \frac{h_i}{h_o}$$

A large number of applications result in values of (h_i/h_o) ranging from 2 to 10, and it is under these conditions that types S/T and W/H Trufin are most applicable, for these tubes have (A_o/A_i) values ranging from just under 3 up to over 6.

This section is devoted to applications where single-phase heat transfer is taking place on the finned surface of the tubes. Typical applications include (but are not limited to) the following:

1. **Cooling of liquids and gaseous product with cooling water.** It is frequently necessary to cool gas or liquid products for storage, using cooling tower or naturally available water. Unless the product is very corrosive, the water will usually be in the tubes. The water coefficient will usually be about 1000 *Btu/hr ft² °F*, whereas a typical shell-side coefficient will be from 50 for a moderate pressure gas to 300 to 350 for a non aqueous, low-viscosity liquid. The use of high-finned tube (Type H/F) might be considered for the moderate pressure gas, but construction requirements will usually indicate a shell and tube exchanger with medium-finned type W/H or type S/T Trufin. For the liquids, one of the low-finned type S/T Trufin tubes is usually indicated.

2. **Cooling of compressed gases** (either between the stages or when the compression is complete.) These gas coefficients can vary from 25 to 100; the values are lower than in the previous case because pressure drops are often limited to low velocities through the cooler. Again, medium finned type S/T or W/H Trufin is probably indicated because of its more favorable area ratios, but low finned tubing is also often used.

3. **Feed-effluent exchangers and similar arrangements for heat recovery.** There is increasingly a need to recover heat by using a hot effluent stream from a reactor or a distillation column to heat an incoming stream. One of these streams usually has intrinsically a higher heat transfer coefficient (for example, a hot liquid effluent stream) than the other, and exchanger design advantages often result if Trufin is used.

The above are only typical applications for S/T and W/H Trufin. As a general statement, Trufin should be used wherever the resulting exchanger is less expensive or more operationally convenient than the plain tube exchanger required for the same service. Often, only a comparison of the final designs of both the finned and the plain tube exchangers will reveal the advantages of the Trufin tube design.

2.1.2 DESCRIPTION OF LOW AND MEDIUM FINNED TRUFIN

1. **Type S/T Trufin Low-Finned Tube.** An example of Type S/T Trufin tube is shown in Fig. 2. 1. The tube shown has 19 fins per in., but similar tubes are produced with 16, 26, 32, and 40 fins per in. The fin height for these tubes is approximately 1/16 inch, and these are the tubes commonly referred to as low-finned tubes. The 40-fin product is also supplied with a .035 in. fin height. The 32-fin product has a fin height of .032 and is generally supplied in titanium.

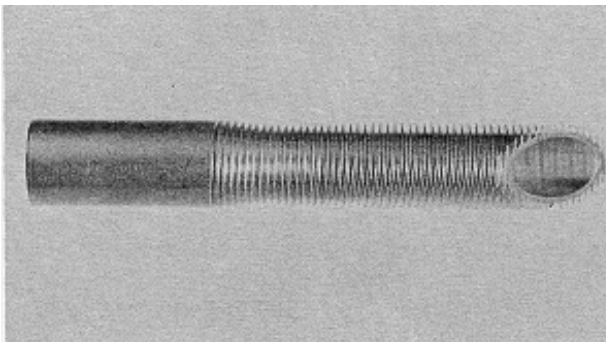

Fig. 2.1 Wieland Type S/T Trufin Low-Finned Tube With 19 Fins per In

2. **Type S/T Trufin Medium-Finn Tube.** S.T Trufin medium finned tube is characterized by having 11 fins per in. and fins 1/8 in. high, resulting in outside to inside surface area ratios of about 5. A typical tube is shown in Fig. 2.2. These tubes are supplied either with belled ends suitable for rolling into tube sheets or plain ends up to 3 in. in length.

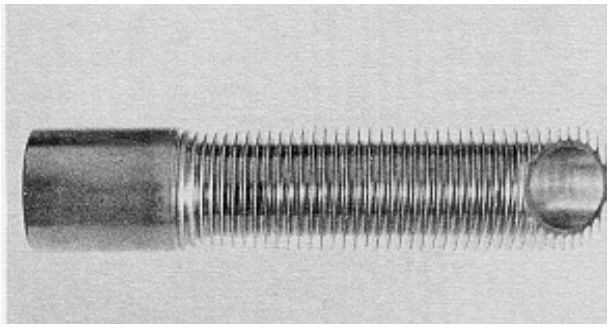

Fig. 2.2 Wieland Type S/T Trufin Medium-Finned Tube With 11 Fins per In

3. **Type S/T Turbo-Chil Finned Tube.** Type S/T Turbo-Chil finned tube is illustrated in Fig. 2.3. This tube configuration combines the 19, 26, or 40 fins per in. external surface enhancement of conventional S/T Trufin with the enhancement of the inside heat transfer coefficient afforded by the spiral ridges. The turbulence level of the fluid in the tube is increased by the spiral ridges.
   Because Turbo-Chil enhances both the inside and outside heat transfer, it is mainly useful in applications where the heat transfer coefficients on either side of a plain tube would be comparable in magnitude. Turbo-Chil then allows a sharply increased heat transfer rate per unit length of tube and can considerably reduce the volume of heat exchanger required for a particular service.
   Special correlations are required for the in tube heat transfer and pressure drop for Turbo-Chil.

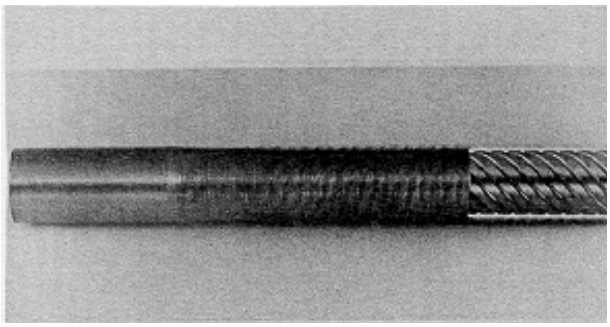

Fig. 2.3 Wieland Type S/T Turbo-Chil Finned Tube Note the 10-start internal ridge

2.2 BASIC EQUATIONS FOR HEAT EXCHANGER DESIGN

2.2.1 THE BASIC DESIGN EQUATION AND OVERALL HEAT TRANSFER COEFFICIENT

The basic heat exchanger equations applicable to shell and tube exchangers were developed in Chapter 1. Here, we will cite only those that are immediately useful for design in shell and tube heat exchangers with sensible heat transfer on the shell-side. Specifically, in this case, we will limit ourselves to the case when the overall heat transfer coefficient is constant and the other assumptions of the mean temperature difference concept apply. Then the basic design equation becomes:

(2.1) $$Q_T = U^* A^* F(LMTD)$$

where Q_T is the total heat load to be transferred, U* is the overall heat transfer coefficient referred to the area A*, A* is any convenient heat transfer area, LMTD is the logarithmic mean temperature difference for the purely countercurrent flow configuration, and F is the configuration correction factor for multiple tube-side and/or shell-side passes. Charts of F for the common shell and tube exchanger configuration are discussed later.

U* is most commonly referred to A_o the total outside tube heat transfer area, including fins, in which case it is written as U_o and is related to the individual film coefficients, wall resistance, etc. by

(2.2) $$U_o = \frac{1}{\frac{1}{h_o} + R_{fo} + R_{fin} + \frac{\Delta x_w A_o}{k_w A_m} + R_{fi}\frac{A_o}{A_i} + \frac{1}{h_i}\frac{A_o}{A_i}}$$

where h_o and h_i are the outside and inside film heat transfer coefficients, respectively, R_{fo} and R_{fi} are the outside and inside fouling resistances, Δx_w, and K_w are the wall thickness (in the finned section) and wall thermal conductivity, and R_{fin} is the resistance to heat transfer due to the presence of the fin. Since all of the low and medium-finned tubes manufactured by Wieland Thermal Solutions are integral (i.e., tube and fins are all one piece of metal), there is no need to include a contact resistance term.

Suitable correlations for h_o and h_i will be developed later in this section. The fouling resistances are ordinarily specified by the customer based upon experience with the streams in question, but typical values may be found in Chapter 1, Table 1.2.

The mean wall heat transfer area A_m is given with sufficient precision as

(2.3) $$A_m = \frac{\pi L}{2}(d_i + d_r)$$

If it is preferred to use an overall heat transfer coefficient based upon the inside heat transfer area A_i the following relationship holds:

(2.4) $$U_o A_o = U_i A_i$$

It is of the greatest importance to always identify the reference area when quoting the value of a film or overall heat transfer coefficient.