4.1. Heat Exchangers with High-Finned Trufin Tubes

4.1.1. Areas of Application

It is frequently the case that one fluid in a heat exchange process has a much higher film heat transfer coefficient than the other, under the conditions of the given problem. Thus, water very commonly gives a value of 1000 to 1500 Btu/hr ft²°F, whereas air at atmospheric pressure usually gives a value of about 10.

The consequence of this imbalance is that the size of the heat exchanger is almost completely controlled by the necessity of providing a large area in contact with the poor heat transfer medium. Often, the best way to provide this area without unduly increasing the overall size of the heat exchanger is to use banks of high-finned tubes such as shown in Fig. 4.1, with the poor heat transfer medium flowing across the finned surface and the other fluid inside the tube.

High-finned Trufin is used in a wide variety of services, but the large majority of applications are for transferring heat to atmospheric air. Air has become increasingly important as the ultimate medium for rejecting waste heat, for a variety of reasons. In some areas, water is altogether lacking. Even when available, water may be too costly, or require too much treatment to minimize fouling or corrosion, or require too much reprocessing before it can be discharged to the environment. And in some situations, air is preferable to even readily available water as a cooling medium. The final result is a rapidly increasing need for heat exchangers specifically designed to handle, and transfer heat to, large quantities of air.

In these exchangers, air is blown across banks of finned tubes, picking up heat from the stream on the tube-side, which is correspondingly cooled. The hot air is then usually dispersed into the atmosphere and the heat dissipated by mixing. Equipment designed for this purpose is commonly called an air-cooled heat exchanger or air cooler, and to some extent this term has become a generic term for most high-finned Trufin apparatus. However, it is important to remember that high-finned Trufin has a variety of other applications, as this section will indicate.

Typically, in an air-cooled exchanger, the fluid on the tube-side may be a process liquid that needs to be sensibly cooled before going to storage or to the next step in the process, or the tube-side fluid may be a vapor that must be condensed. Further, the vapor may be originally superheated above its saturation temperature so that it needs to be de-superheated before condensation takes place. The vapor may be essentially a single component, or it may be a mixture of several components, not all of which are necessarily condensable. Subcooling of the condensate may also be performed in air-cooled equipment.
In the cases cited above, the cooling process was direct, in the sense that the heat was transferred directly from the process fluid, through the tube wall and fins, to the air. Indirect air cooling is also used. In this arrangement, the process streams are actually cooled in shell and tube heat exchangers with a closed water loop as the intermediate coolant. The water is then cooled in air-cooled heat exchangers. This arrangement allows the use of generally more compact water-cooled equipment in the immediate vicinity of the process units with ultimate heat rejection accomplished to the atmosphere with air-cooled equipment on the periphery of the plant. The quality of the intermediate coolant can be controlled to minimize corrosion and fouling problems. The equipment and piping arrangement is more extensive and complicated, and the temperature of the process fluids cannot be reduced as low as for the direct cycle.

The indirect cycle can be very useful if air temperatures get so low as to cause the process fluids to freeze up or get exceedingly viscous upon direct cooling. The intermediate fluid in the indirect cycle can be a water-glycol mixture so chosen that it will not freeze under the most extreme conditions encountered. The temperature of the intermediate fluid can be controlled by bypassing a portion of it around the coolers, or by shutting down some or all of the fans.

In addition to its use in rejecting heat from process plants, high-finned Trufin may also be used to reject heat from power plants (both direct and indirect cooling cycles have been proposed and constructed), and refrigeration and air-conditioning systems. In another application - space heating systems - it is the warm air off of the tubes that is the desired product; the heat source may be either condensing steam or a hot liquid inside the tubes.

There are, finally, two applications areas in which the atmospheric air is the heat source rather than the heat sink. In the first of these, air is used to supply heat to a process fluid, either to sensibly warm it or to vaporize it. There is only a limited temperature range over which atmospheric air may be cooled, since if the fin surface temperature drops below the dew point of the air and below 32°F (0°C), ice will form on the fins and can rapidly lead to a restricted air flow and possible mechanical damage to the tube. In the other application, it is the cooled air (for air-conditioning or refrigeration) which is of interest, the cold sink being a boiling refrigerant or possibly a sensibly heating brine.

In principle, the air-side calculations (i.e., on the tube bank finned surface) are the same for all of the above applications, though the specific ranges of parameters vary widely. However, the tube-side calculations are quite different, and are covered in other sections of this manual.

4.1.2. High-Finned Trufin

All high-finned Trufin made by Wolverine has integral fins. That is, the fins are raised from the base tube metal in a fabricating operation so that the final tube and its fins are one piece of metal, except for the Trufin Type L/C which has an internal liner of a different metal. But here also, the outer tube and the fins are a single piece of metal.

Integral firming ensures the maximum thermal efficiency of the tube since there is no possibility of the fins becoming partially or totally separated from the tube metal by environmental corrosion at the base of the fin or by repeated expansion and contraction in operation or by mechanical damage in handling.

There are several different types of Wolverine high-finned tubes manufactured. The descriptions given below are intended only to indicate the major features of each type and certain general classes of applications that each lends itself to. Certain limitations of material availability and construction features are also indicated. For a detailed listing of sizes available, tolerances, material specifications, ordering information, etc., see Section 6.
Fig. 4.2 defines the major geometrical parameters common to Wolverine high-finned tubes. Type H/F Trufin (Fig. 4.3) is normally produced in alloy 122 (DHP copper) but is also available in some sizes of Alloy 706 (90/10 copper nickel). Standard size inside diameters range from 5/16 in. to 1 1/4 in. with corresponding fin diameters from 0.9 in. to 2.2 in. and fin counts of 7 and 9 fins per inch.

Type H/R Trufin (Fig. 4.4) is produced in 3003 aluminum. Standard root diameters range from 3/8 in. to 1 in. with corresponding fin diameters from 0.9 in. to 1.9 in. and standard fin counts of 5, 7, 9, and 11 fins per inch. Type I/L Trufin (Fig. 4.5) has internal longitudinal fins as well as helical fins on the outside. It is available only in 3003 aluminum and has the same fin configurations as Type H/R aluminum.

Type L/C Trufin (Fig. 4.6) is a duplex finned tube in which the outer tube is integrally finned 3003 aluminum with the same outside fin configuration similar to Type H/R aluminum. The inner tube may be of any material including copper, admiralty, copper nickels, low carbon and stainless steels. The dimensions of the inner tube are standard dimensions for heat exchanger tubes. Type L/C Trufin is used when corrosion or pressure considerations require the use of a special material in contact with the process fluid. The aluminum assures good heat conductance through the fins into the air, as well as excellent resistance to atmospheric corrosion.

Fig. 4.2 Geometrical Parameters for High-Finned Trufin Tubes.

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**Fig. 4.3** Wolverine Trufin Type H/F  
**Fig. 4.4** Wolverine Trufin Type H/R
4.1.3. Description of Equipment

1. Basic Arrangements. Most large air-cooled heat exchangers are essentially composed of a shallow (3 to 8 rows) tube bank across which a large quantity of air is blown or drawn at relatively low velocities by large fans. Two different configurations are shown in Figs. 4.7 and 4.8.

Fig. 4.7 shows a horizontal tube, forced draft arrangement, in which the fan is mounted below the tube bank and blows air upwards past the tubes. This configuration is commonly used for both cooling liquids and condensing vapors; especially in the latter case, the tubes may be slanted 2° or 3° downwards in the direction of flow to facilitate drainage of the condensate from the tube. The forced draft arrangement is mechanically attractive: the fan and drive may be supported directly on the ground with a fairly short shaft, easing stress and vibration problems and simplifying maintenance. However, the hot air leaves the top of the unit at a fairly low velocity and may tend to recirculate through this or nearby units and raise the inlet temperature, reducing the unit capacity.

Fig. 4.8 shows the horizontal tube, induced draft arrangement. Induced draft produces generally more uniform airflow across the bundle and projects the hot air plume more positively into the atmosphere, reducing recirculation problems. However, in the arrangement shown, fan and driver are more difficult to secure and maintain, and the fan loses efficiency in handling the less-dense hot air. An alternative arrangement places the driver below the bundle, connected by a long shaft to the fan above the bundle; however, tubes must be left out of the bundle to allow the shaft through, with consequent bypassing problems and loss of surface, not to mention potential shaft vibration problems.
2. Tube Bundle Construction. The tubes are always arranged in a triangular layout or (less commonly) a rotated square layout, as shown in Fig. 4.9. Inline arrangements are never used because a major portion of the air can flow through the bundle in the clear channel between the tips of fins on adjacent tubes and mix only very poorly with the heated air flowing through the fin field. The effect is to reduce the apparent heat transfer coefficient to approximately half of that of a triangular array. (Refs. 1, 2) Even in the staggered layouts the tubes are put as close together as possible without having the tubes vibrate against each other in normal operation. Typically, the tip to-tip clearance is 1/4 in.

![Image of triangular and rotated square layouts]

**Fig. 4.9** Finned Tube Unit Cell Geometries: a) Equilateral Triangular Layout  b) Rotated Square Layout

Usually the tubes are inserted in box-type headers, Fig. 4.10. The tubes may be expanded into and/or welded to the tube sheet. In the simple box header, the flanged cover plate may be easily removed, exposing the ends of all of the tubes for inspection, leak testing, cleaning, replacement, rerolling or rewelding, or plugging. The integrally welded box header is designed for higher pressure operation; any operations on the tubes are carried out by removing the inspection plugs and working through the inspection hole.

![Image of simple box header, box header with tube inspection plugs and pass divider, and manifold header]

**Fig. 4.10** Typical Headers for Air-Cooled Exchangers; a) Simple Box Header with Flanged Cover Plate, b) Integrally-Welded Box Header with Tube Inspection Plugs and Pass Divider, c) Manifold for High Pressure on Tube-Side.

A minimum of three rows of tubes is used in tube banks; the usual maximum is eight rows, though occasionally up to 12 are employed. The greater number of rows of tubes is used where a relatively small
Airflow is required. Generally in this case the tube-side fluid is at high temperature and the air temperature can increase over a correspondingly greater range.

In certain applications (e.g., condensation, especially multi-component mixtures) the tube bank is arranged for single pass; that is, the process fluid is introduced at one end and flows in parallel through all of the tubes in the bundle to the other end, where it exits from the exchanger. For other applications (e.g., cooling of a liquid over a wide temperature range), it is often better to let the tube side fluid flow back and forth through the tube rows making two, three, or occasionally more passes through the exchanger and across the air stream. In this case, the tube side fluid always starts at the top tube row and with successive passes, moves to lower tube rows in order to approximate as closely as possible countercurrent flow. The flow path is controlled by pass dividers or partition plates in the headers, as illustrated for a two pass unit in Fig. 4.10b. Usually an integral number of tube rows (one or two) is taken for each pass, but by using vertical as well as horizontal dividers it is possible to split rows into fractional rows per pass, as shown in Fig. 4.11. However, in order to make room for the plates it may be necessary to omit tubes or distort the layout, which may result in air bypassing. Also, tube-side fluid distribution may be non-uniform. Since the assumptions may not be satisfied, such arrangements should not be used when close temperature approaches are called for.

If the distance between headers is greater than about 4 to 6 feet, it is necessary to provide periodic tube support plates, through which the tubes pass with only enough clearance over the fin diameter to allow the tubes to be inserted easily during assembly. To provide a bearing surface for support, the space between fins may be filled (for the distance of a few fins) with a low melting alloy such as a solder cast in place or by a wraparound shroud, or the support plate may be made thick enough to hold several fins.

The bundle is held together and stiffened by side members of appropriate size and shape (commonly U-channels) bolted or welded to the headers and the support plates and to the legs or tower support structure. The clearance between the outermost tubes and the side members should be as small as possible to minimize air bypassing. Auxiliary features such as plenums, shrouds, louver systems, and screens are fastened to the main structural members for support.

3. Fans and Drivers. Next to the tube bundle itself, the fan and its driver are the most important elements of the air-cooled exchanger. The fans are invariably axial flow propeller type, with four or six blades, up to 24 feet in diameter (larger ones are available). Especially in the larger sizes, adjustable pitch blades are used. Maximum static pressure is limited to one inch of water, with one half inch of water pressure drop a common design specification.

The fan blades are often made of plastic if the air temperature does not exceed 175°F, which includes many induced draft applications. Aluminum blades are usable up to around 300°F and steel at higher temperatures. Tip speed should not exceed about 12,000 ft/min in the larger sizes; apart from strength considerations, fan noise increase rapidly at higher speeds.
Both electric motor and steam turbines are used as drivers, either direct or indirect. In isolated locations, engine drivers can be used. For indirect drive, V-belts or reduction gears are used; temperature limitations must be observed in induced draft positioning. Hydraulic drives are also used.

In estimating fan power requirements, a combined fan and driver efficiency of 60 percent is reasonable, with a range from 50 to 75 percent. In view of the many options open for fans and drives and the specialized nature of the field, detailed information and recommendations should be sought from the manufacturer for each application.

4. Other Components. Fan rings, shrouds and plenums are always used on air-cooled exchangers to guide the airflow and minimize peripheral escape, bypassing, and recirculation of air. There seems to be no general body of information available on design practices in these areas, and reasonable attention to the mechanics of airflow is probably sufficient to avoid catastrophic problems.

A course screen underneath the fan is commonly used as a safety precaution in forced draft units where there is any possibility that people or animals may get close to the fan and also to protect the fan and tube bundle from ingesting flying debris.

A variety of techniques and equipment is used to ensure that the process fluid does not freeze in the tubes under conditions of low air temperature. First, the fans can be shut off: as a rough rule of thumb, an air-cooled heat exchanger will transfer about one-third as much heat with the fans off. Second, louvers may be installed under the tube bank to further restrict airflow. Third, it is possible to arrange for some of the fans to be reversible and deliberately recirculate some of the warm air that has passed through the bundle back through the bundle into the fresh air supply. Fourth, some bundles have a separate row of tubes underneath the main bundle into which steam can be bled to warm the incoming air.
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.

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 Chapter I, and 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, \( \Phi \), 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:

\[
\Phi = \frac{1}{1 + \frac{m^2}{3} \frac{d_o}{d_r}} \quad (4.1)
\]

where

Fig. 4.12  Typical Temperature Distribution in a Fin
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 \( h_0 \) 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 \( \Phi \) in a typical problem, consider the following example:

Type H/R tube, 3003 aluminum:

\[
\begin{align*}
d_r &= 1.00 \text{ in.} \\
d_o &= 1.875 \text{ in.} \\
H &= 0.437 \text{ in.} \\
s &= 0.076 \text{ in.} \\
Y &= 0.015 \text{ in.} \\
k_w &= 110 \text{ Btu/hr ft}^2\text{°F} \\
h_o &= 10 \text{ Btu/hr ft}^2\text{°F} \\
R_{fo} &= 0.0
\end{align*}
\]

Then:

\[
m = \frac{2}{\left(\frac{1}{h_o} + R_{fo}\right) k_w Y} = 0.439
\]

\[
\Phi = \frac{1}{1 + \frac{(0.439)^2}{3} \left(\frac{1.875}{1.000}\right)} = 0.919, \text{ i.e., } 91.9\% \text{ fin efficiency}
\]

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. Copper fins have a higher thermal conductivity and would give a higher \( \Phi \). (Copper nickel (90/10) would give \( \Phi = 0.730 \) 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\(^2\)°F would give \( \Phi = 0.862 \).

A quantity somewhat more directly useful in design calculations is the "Fin Resistance", \( R_{fin} \), defined as:
$A_{\text{fin}} = \frac{1 - \Phi}{A_{\text{root}} / \Phi + \Phi} \left( \frac{1}{h_o} + R_{fo} \right)$ \hspace{1cm} (4.3)

where $A_{\text{root}}$ is the surface area of a unit length of plain(unfinned) tube between the fins and $A_{\text{fin}}$ 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 $R_{\text{fin}}$ as follows:

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

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

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

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 $h_i$, 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 value of $R_{\text{fin}}$ 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 the 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 surface. 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 hr ft²/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/aluminum case), the bond resistance has been measured to be about 0.0002 hr ft²/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 hr ft²/F/Btu (based on contact area) at air side coefficients of 5 Btu/hr ft²/F (based on fin area) and 0.002 hr ft²/F/Btu for air side coefficients of 10 Btu/hr ft²/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:

\[
U_0 = \frac{1}{h_o + R_{fo} + R_{fin} + \left( \frac{\Delta x_w A_o}{k_w A_m} \right)_1 + R_b \left( \frac{A_o}{A_b} \right) + \left( \frac{\Delta x_w A_o}{k_w A_m} \right)_2 + R_f \left( \frac{A_o}{A_f} \right) + \frac{1}{h_i} \left( \frac{A_o}{A_i} \right)}
\]  

\[ (4.5) \]

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.
4.3. Heat Transfer and Pressure Drop in High-Finned Trufin Tube Banks

4.3.1. Heat Transfer Coefficients in Crossflow

1. Briggs and Young Correlation. A number of correlations have been published for heat transfer during flow across banks of finned tubes. The number and range of variables are so large that it would be surprising if a relatively simple correlation would be generally applicable. More complex correlations require correspondingly greater data sets with particular emphasis upon wide ranges of variables and multivariate interactions over these ranges, and the published data generally do not meet these criteria. Therefore, the published correlations must be used with great care to ensure that they are applicable in the range of interest.

Within the above caution, one of the best published correlations for high finned Trufin tubes is due to Briggs and Young (5).

\[
\frac{h_{\text{air}} d_r}{k_{\text{air}}} = 0.134 \left( \frac{d_r \rho_{\text{air}} V_{\text{max}}}{\mu_{\text{air}}} \right)^{0.68} \Pr_{\text{air}}^{1/3} \left( \frac{H}{s} \right)^{-0.2} \left( \frac{Y}{s} \right)^{-0.12}
\]  \hspace{1cm} (4.6)

This correlation represents data for root diameters from 0.44 in. to 1.61 in. and fin heights from 0.056 in. to 0.652 in. Fin spacings ranged from 0.035 in. to 0.17 in. The tubes were in equilateral triangular pitch tube banks with pitches up to 4.5 in.

\( V_{\text{max}} \) is the maximum air-side velocity going through the finned tube bank and the other quantities have their usual meaning. Fin spacing, \( s \), is related to the number of fins per inch \( N_f \) by the equation:

\[
s = \left( \frac{1}{N_f} \right) - Y \]  \hspace{1cm} (4.7)

It will be useful here to demonstrate the use of this correlation and definitions of the terms by carrying out a typical calculation.

2. Example: Calculate the air side heat transfer coefficient for the case of an aluminum Trufin Type H/R tube with a 3/4 in. root diameter with 9 fins/in. with a mean fin thickness of 0.019. The tubes are in an equilateral triangular layout with a 1 7/8 in. pitch. Air is flowing at 70°F with a face velocity of 600 ft/min.

Solution:

The geometrical parameters are as follows:

\[
d_r = 0.750 \text{ in.}
\]

\[
H = \frac{1}{2} (1.625 - 0.75) \text{ in.} = 0.438 \text{ in.}
\]

\[
Y = 0.019 \text{ in.}
\]
s = (1/9 – 0.019) in. = 0.092 in.

$V_{\text{max}}$: The problem was stated in terms of the "face velocity", i.e., the average velocity of the air approaching the first row of tubes. To convert this to $V_{\text{max}}$ we must first find the free flow area between two tubes at the point of closest approach, per foot of tube. Since the centers of adjacent tubes are 1.875 in. apart and the root diameter is 0.75 in., there is a 1.125 in. clearance between the root diameters of the tubes giving a clearance area of $12(1.125) = 13.50$ in.$^2$/ft. From this clearance area must be subtracted the area blocked by the fins on each tube, which per foot is

$$2(9 \text{ fin/in.})(12 \text{ in./ft.})(0.019 \text{ in./fin})(0.438) = 1.80 \text{ in.}^2/\text{ft.}$$

Thus the free flow area between tubes per foot of length is 11.70 in.$^2$. The "face area" corresponding to these same two adjacent tubes per foot of length is simply the center-to-center distance between the tubes – the pitch – or

$$(1.875 \text{ in.})(12 \text{ in./ft.}) = 22.50 \text{ in.}^2/\text{ft.}$$

The air flowing at 600 ft/min. approaching the face must accelerate to

$$600 \left( \frac{22.50}{11.70} \right) = 1150 \text{ ft/min}$$

to flow between the tubes and this is the value of $V_{\text{max}}$. The physical properties of air at 70°F are:

- $k = 0.0150 \text{ Btu/hr ft}^\circ\text{F}$
- $\rho = 0.0765 \text{ lb}_m/\text{ft}^3$
- $\mu = 0.439 \text{ lb}_m/\text{ft hr}$
- $c_p = 0.240 \text{ Btu}/\text{lb}_m\circ\text{F}$
- $Pr = \frac{0.240(0.0439)}{0.0150} = 0.702$

Finally we may calculate $h_o$:

$$h_o = \left( \frac{0.134(0.0150)}{0.750/12} \left[ \frac{(0.750/12)(0.0765)(1150)(60)}{0.0439} \right]^{0.68} \times (0.702)^{1/3} \left( \frac{0.438}{0.092} \right)^{-0.2} \left( \frac{0.019}{0.092} \right)^{-0.12} \right)$$

$$= 0.0322(432)(0.889)(0.732)(1.208)$$

$$h_o = 10.9 \text{ Btu/hr ft}^2\circ\text{F}$$

This is a very typical value of the air side coefficient, and it may be compared to the relatively very much higher values for tube-side (~1000 for water, 2000 + for condensing steam, ~300 for a medium organic
liquid, ~100 for a heavy organic liquid). This comparison illustrates immediately the importance of high-finned Trufin in air cooled service.

### 4.3.2. Mean Temperature Difference in Crossflow

1. **The LMTD.** It was pointed out in Chapter 1, the great simplification introduced into heat exchanger design by the Mean Temperature Difference (MTD) concept. In its simplest practical form, assuming countercurrent flow, constant overall heat transfer coefficient, etc., the correct definition of the MTD is the Logarithmic Mean Temperature Difference (LMTD) defined as:

   \[
   \text{LMTD} = \frac{(T_i - t_o) - (T_o - t_i)}{\ln \left( \frac{T_i - t_o}{T_o - t_i} \right)}
   \tag{4.8}
   \]

   where \( T_i \) and \( T_o \) are the hot fluid inlet and outlet temperatures and \( t_i \) and \( t_o \) are the cold fluid inlet and outlet temperatures, respectively.

   With this concept one may write the basic design equation (if all of the conditions are satisfied) as

   \[ A_o = \frac{Q}{U_o \text{LMTD}} \tag{4.9} \]

   where \( A_o \) and \( U_o \) are referred to the same reference area, usually the total outside heat transfer area of the heat exchanger.

2. **MTD for Crossflow.** In air-cooled heat exchangers, the air and the process fluid are in crossflow to one another, not in countercurrent flow as assumed in the LMTD derivation. In this case, it is necessary to apply a correction factor \( F \) which may be obtained by mathematical analysis. The Eq. (4.9) may be written as:

   \[ A_o = \frac{Q}{U_o F \text{LMTD}} \tag{4.10} \]

   where, it is important to note, the LMTD is calculated as in Eq. (4.8). The correction factor \( F \) is a function of the parameters and

   \[ P = \frac{t_o - t_i}{T_i - t_i} \tag{4.11} \]

   and

   \[ R = \frac{T_i - T_o}{t_o - t_i} \tag{4.12} \]

   \( F \) is plotted in Fig. 4.13 for crossflow with one tube-side pass, i.e., the tube side fluid flows in parallel through all the tubes. It is assumed that an equal amount of fluid flows through each tube.

   \( F \) is plotted in Fig. 4.14 for crossflow with two tube-side passes, with an equal number of tubes in each pass. There may be more than one row of tubes in each pass. Note that the tube-side flow goes through
the uppermost tubes first, and that the overall flow pattern is moving towards the tube side fluid being in countercurrent flow to the air flow. In fact, for three or more passes, the overall flow pattern is so close to countercurrent that $F$ can be taken to equal to 1.00 with very small error.

**Fig. 4.13** MTD Configuration Correction Factor for Crossflow with One Tube-Side Pass
4.3.3. Pressure Drop in Crossflow

1. Robinson and Briggs Correlation. Correlations for pressure drop across banks of finned tubes are subject to the same cautions as used for heat transfer, and in fact pressure drop is subject to even greater uncertainty. One of the better correlations in the open literature is due to Robinson and Briggs (6):

\[ f_r = 9.47 \left( \frac{d_r \rho_{air} V_{max}}{\mu_{air}} \right)^{-0.32} \left( \frac{P_t}{d_r} \right)^{-0.93} \]  

(4.13)

where \( f_r \) is the friction factor and \( P_t \) is the transverse pitch between adjacent tubes in the same row. The friction factor is defined as

\[ f_r = \frac{\Delta P_{air} R_c}{2n\rho_{air}V_{max}^2} \]  

(4.14)

where \( \Delta P_{air} \) is the pressure drop across the tube bank (in say lb/ft^2) and \( n \) is the number of tube rows in the bank.

This correlation represents data for tube banks with root diameters from 0.734 to 1.61 in., fin diameters from 1.561 in. to 2.750 in., and pitches from 1.687 to 4.500 in.
Most of the data are for equilateral triangular arrangements. However, two tube banks with isosceles triangular arrangements were tested, and it was found that the data for these two banks could be adequately correlated if the additional factor

\[
\left( \frac{P_t}{P} \right)^{0.52}
\]

were included on the right-hand side of Eq. (4.13). In the above expression, \( P \) is the "longitudinal" pitch of the tube bank, defined as the distance between the centers of adjacent tubes in different rows, measured along the diagonal. For an equilateral arrangement, \( P_t = P \). Since one of the isosceles tube banks tested was close to a staggered square arrangement there is some reason to believe that this correlation will prove adequate for predicting pressure drop in such geometries also.

2. Example. Using the same tube and arrangement of the previous example and the same air flow rate, calculate the pressure drop across a tube bank of five rows of tubes.

Solution: The geometrical and flow parameters (including Reynolds number) are unchanged. Then the friction factor \( f_r \) is, by Eq. (4.13):

\[
f_r = 9.47 \left[ \frac{\left( \frac{0.750}{12} \right) (0.0765)(1150)(60)}{0.439} \right]^{-0.32} \left( \frac{1.875}{0.750} \right)^{-0.93} = 9.47(0.0575)(0.426) = 0.232
\]

and the pressure drop is, by Eq. (4.14):

\[
\Delta P_{air} = \frac{0.232(2.5)(0.0765 lb_m / ft^3)(1150 ft/min)^2}{32.2 \left( \frac{lb_m ft}{lb_f sec^2} \right) \left( \frac{60 sec}{min} \right)^2} = 2.03 \left( \frac{lb_f}{ft} \right)^2 = 0.0141 \left( \frac{lb_f}{in.} \right)^2
\]

It is customary in stating the air-side pressure drop in air-cooled heat transfer equipment to quote it in "inches of water", that is, as the number of inches of height of a column of water at the earth's surface that would be supported by the pressure difference. The pressure drop is converted from \( lb_f/ft^2 \) to inches of water by multiplying the reciprocal of the density of water (commonly taken as 62.4 \( lb_m/ft^3 \)), by (12 in./ft), and by \( g_c/g \) which has a numerical value of unity but converts the units properly. So, for this case:

\[
\Delta P_{in.} \text{ of } H_2O = \left( \Delta P_{air} \frac{lb_f}{ft^2} \right) \left( \frac{1}{\rho_{H_2O}} \right) \left( \frac{g_c \text{ to } ft \text{ of water}}{g} \right)
\]

\[
\Delta P_{in.} \text{ of } H_2O = 2.03 \left( \frac{lb_f}{ft^2} \right) \left( \frac{1}{62.4 lb_m / ft^3} \right) \left( \frac{12 \text{ in.}}{ft} \right) \left( \frac{lb_m}{lb_f} \right) = 0.39 \text{ in. } H_2O
\]
This value is well within the capability of fans that would be used in this service. A pressure drop of 1/2 in. of water is a common design value.

### 4.3.4. Other Air-Side Pressure Effects

There can be other sources of pressure loss on the air-side of an air-cooled exchanger, such as the louver system (even when it is open), fan guards, structural elements, plenums and shrouds. Few data or correlations are generally available in the literature to permit accurate evaluations of these effects. Fortunately, the losses should be very small and a rough estimate of their magnitude is usually sufficient, mainly as reassurance that they will not be major features in limiting the performance of the units.

The following is a useful rule of thumb procedure for any flow situation in which the major pressure effect is from a sudden acceleration of the flow followed by a sudden deceleration.

Calculate the increase in air velocity that must be achieved in order for the air to flow through the restriction (say the open louver system, for example). Then allow twice the velocity head calculated from this velocity increase as the irrecoverable pressure loss. Thus, suppose the flow in the previous example had to accelerate from 800 feet/minute to 900 feet/minute to pass through a fan guard. The velocity head represented by that velocity change is:

\[
\Delta P = \frac{\rho_{\text{air}} \left( \frac{V_2^2}{2} - \frac{V_1^2}{2} \right)}{g_c}
\]

\[
= \left( 0.0765 \ \text{lb} \text{m} / \text{ft}^3 \right) \left( (900 \ \text{ft/min})^2 - (800 \ / \text{min})^2 \right)
\]

\[
= \frac{2 \left( 32.2 \ \frac{\text{lb} \text{m}}{\text{lb}_f \ \text{sec}^2} \right)(60 \ \text{sec} \ / \text{min})^2}{\text{min}}
\]

\[
= 0.056 \ \frac{\text{lb}_f}{\text{ft}^2} = 3.9 \times 10^{-4} \ \frac{\text{lb}_f}{\text{in}^2} = 0.011 \ \text{in. H}_2\text{O}
\]

Doubling this loss gives 0.02 in. H$_2$O as the probable maximum effect. This is a sufficiently small loss (compared to 0.39 in. of H$_2$O across the tube bank) that it would not cause a major problem in the design or operation.

If the pressure drop thus calculated turns out to be substantial by this estimate, manufacturers' data or a much more detailed calculation of the effect is required.

Finally, if the discharge air plume is to be accelerated strongly in a smoothly converging duct in order to cause it to travel far above the unit before it mixes with the surrounding air, the additional pressure drop required for acceleration can be conservatively taken to be one velocity head based upon the desired discharge velocity.
4.4. Preliminary Design Procedures

4.4.1. Principles of the Design Process

The essential feature of most design problems involving air coolers is that a certain thermal change must be made on the process stream, using air which can only undergo limited temperature and pressure changes.

The air-side heat transfer process is usually controlling in the heat removal process, and the limited pressure drop possible with the air restricts the values of velocities (and, therefore, heat transfer coefficients) to a very narrow range.

The first problem of design is to select the general features of the heat exchanger configuration. The second problem is to calculate whether or not the configuration selected will transfer the required amount of heat within the pressure drop limitations.

There are basically three possible outcomes to the second problem:

1. The configuration selected will transfer the heat, using (but not exceeding) the available pressure drops. This then represents the desired design.

2. The configuration will not transfer the heat specified unless the air-side pressure drop is exceeded. In this case, it is necessary to select a different (larger) design and return to the heat transfer and pressure drop calculations, until the first outcome above is achieved.

3. The configuration will transfer the heat specified, but does not use the pressure drop available. In this case, the exchanger is too large and can be reduced in size, saving money, until the first outcome is realized.

The above is a gross over-simplification of the design problem, but it illustrates the essentials of the process and the criteria by which success is measured.

This section will concentrate upon the first problem – the selection of a design whose major features are fairly close to the final design. In fact, for many purposes, such as plant capital cost estimates or preliminary plant layout, the first-cut design may be sufficient.

4.4.2. Selection of Preliminary Design Parameters

1. Selection of Tube. Finned surface tubes are mainly of interest when the fluid on the fin side has a much lower coefficient than on the tube-side. To a very rough approximation, the tube should be chosen so that

\[ h_i A_i \approx h_o A_o \]  

or in terms of the area ratios available,

\[ \frac{A_o}{A_i} \approx \frac{h_i}{h_o} \]  

(4.16)  

(4.17)
Since, as we have seen, ho for typical air-cooler applications is about 10 Btu/hr ft²°F, we may make a rough tube selection by choosing one with an $A_o/A_i$ ratio approximately one-tenth the numerical value of $h_i$ when the latter is given in Btu/hr ft²°F.

Thus, if experience had taught that $h_i$ was about 100 Btu/hr ft²°F (typical of cooling a medium weight organic liquid in the lower turbulent flow regime), then one would consider choosing a tube with $A_o/A_i \approx 10$, of which there are many available.

However, many streams to be cooled (such as water or condensing steam) give an $h_i \geq 1000$ Btu/hr ft²°F, and there simply are no tubes available that have the corresponding area ratios. In those cases, one simply selects one of the higher ratio tube configurations, taking into account material, availability, prices, and other less tangible considerations.

Process stream considerations do affect the choice of tube also. For low flow rates, and for liquids generally, the smaller diameter tubes are generally preferable so that tube-side velocities can be kept up to ensure that the flow is in the turbulent regime and that fouling is minimized. Turbulent flow is generally preferred, partly because of the better heat transfer coefficient, but also to reduce the possibility of a flow maldistribution among the tubes.

For high tube-side flow rates, or for gases and condensing vapors, or where tube-side pressure drop is limited, larger diameter tubes are generally preferable. No absolute rules can be written - each case must be considered on its own merits in terms of operability and cost.

2. Selection of a Tube Layout. It has been emphasized above that staggered tube layouts - usually equilateral triangular, less commonly other triangular or rotated square - must be used to minimize bypassing in tube banks. Within this limit, however, and for a given tube, there is the question of what pitch to use. There must be some minimum clearance - on the order of 3/16 to ¼ inch - between fin tips to prevent fin-to-fin impact (with noise and mechanical damage resulting) during operation. Larger clearances are possible and perhaps desirable, since pressure drop decreases more rapidly than heat transfer coefficient as the pitch increases and the velocity decreases. Inevitably, however, the result is to increase the size and cost of the exchanger. Therefore, the usual practice is to put the tubes as close together as possible.

3. Selection of Design Air Temperatures. The selection of the inlet and exit air temperatures are both matters of concern to the designer, though the considerations in their respective selection are quite different.

The inlet air temperature at any given moment at a given location is set by nature, but there is usually a substantial range of air temperatures over the course of a year or even a day. For most plant locations, the data are available to plot the percentage of hours in the year when the air temperature will exceed a given value. The designer (or usually, the process engineer) must then decide which temperature to choose in terms of the fraction of time that the heat exchanger will be nominally under-designed and incapable of handling the design heat load at the process stream temperatures specified.

Thus, the designer may elect the 5 percent level – that air temperature that is exceeded only 5 percent of the hours of the year. In principle, this means the exchanger will fail to meet demand 5 percent of the year and will be over-designed the other 95 percent. In practice, it is not nearly that simple. Uncertainties at many other points in the design process, including heat transfer coefficients, process stream conditions, changes in the operation of the plant, fouling transients, etc., plus the provision of process
control and flexibility in other parts of the process greatly affect even the criterion of what constitutes failure to meet process requirements.

Therefore the selection of an inlet temperature loses much of the central importance that has been assigned to it in some past discussions, and the major concern is to select one near but below the maximum air temperature likely to be encountered. Then the exchanger is designed with an eye upon flexibility to application and operation in the particular circumstances of the given problem.

It should be noted that problems are as likely to arise from the fact that the air temperature is colder than the design value most of the time. Thus the process stream may be overcooled, leading to freeze-up or, in a reflux condenser, overloading a column or its reboiler. These problems should be anticipated by the process engineer, rather than the exchanger designer, but someone needs to make provision for controlling the air flow rate or other variables to avoid the worst consequences.

More often than not, choice of the design inlet air temperature fixes the minimum process fluid exit temperature. This is because it is not generally economically justified to cool the tube-side fluid to a temperature lower than about 20°F hotter than the inlet air; stated another way, the temperature approach at the cold end is generally chosen to be at least 20°F. However, approaches as low as 10°F have been specified, leading to a substantial increase in area. Larger approaches can be used, of course, if there is no need to cool the process fluid so far.

The exit air temperature must also be chosen by the designer, but the considerations limiting this choice are set by the process rather than the climate. That is, the exit air temperature must always be less than the inlet temperature of the process stream. If there are several tube-side passes and if the other assumptions underlying the validity and the logarithmic mean temperature difference derivation are satisfied, then in theory one can obtain a workable design for any exit air temperature less than the inlet process fluid temperature. In practice, this is never pushed to the limit, first, because there are too many things that can cause the theoretical model to be violated, and second, because it simply leads to an excessively large and costly exchanger.

The approach temperature limit sometimes occurs between the exit air and the inlet process fluid. In that case, the approach is rarely less than 20°F and more commonly 40°F.

Where only one or two tube-side passes are involved, it is important to calculate $F$, the MTD correction factor, for the temperatures chosen and make sure that it has a good value ($F > 0.8$) and that it is not on or near the steep portion of the curve. If these conditions are not satisfied, it is necessary to back off on the temperatures specified and pay the penalty in process efficiency.

If the assumptions underlying the $F$-LMTD derivation are not satisfied, then a much more careful analysis of the approach temperature selection and the MTD evaluation are required.

### 4.4.3. Fundamental Limitations Controlling Air-Cooled Heat Exchanger Design

1. **The Nature of the Problem.** There are two or three aspects of air-cooled heat exchanger design which are in some sense in competition. On the one hand, there is the limited capacity of the air to absorb heat; this we may term the "thermodynamic limitation." On the other hand, there is the limited rate at which heat can be transferred to the air; this we may term the "rate limitation."

The thermodynamic limitation calls for moving very large quantities of air across the exchanger, accepting the maximum possible change in the air temperature. Given the low pressure drops acceptable, however, this large air flow must be accommodated by using a large face area and shallow depth in the exchanger.
The "rate limitation" requires that the temperature difference between the streams be kept as great as possible (which implies small temperature changes in the air) and that the velocities be kept as high as possible, again within the fan capabilities.

Both limitations have in common the desire to use high air velocities to overcome them, and the mutual limit upon this we could call the "pumping limitation."

In every problem, these three limitations must be balanced, but the point of balance is very dependent upon the particular features of each problem. In this section, we present the common design standards that have evolved and deduce from them a procedure for selecting a preliminary design.

2. The "Pumping Limitation. " The fans in use on air cooled heat exchangers give a maximum practical pressure drop of one inch of water (5.20 lb/ft² = 0.0361 lb/in²). However, the usual design range is from 0.3 inches of water to 0.7 inches, with 0.5 inches being a convenient design target. This converts very approximately into two somewhat more convenient design quantities.

The first is in terms of the mass velocity passing through the minimum flow area, defined in terms already used

\[ \rho \text{air} V_{\text{max}} \]

Typical values of \( \rho \text{air} V_{\text{max}} \) as a function of the number of rows of tubes are shown in Table 4.1.

The second design quantity commonly cited is the face velocity, the average air velocity approaching the face of the tube bank, which as a function of the number of rows is given in Table 4.2.

<table>
<thead>
<tr>
<th>n, No. of Rows of Tubes</th>
<th>( \rho \text{air} V_{\text{max}} ) lbm/hr ft²</th>
</tr>
</thead>
<tbody>
<tr>
<td>3</td>
<td>5000-6000</td>
</tr>
<tr>
<td>4</td>
<td>5000</td>
</tr>
<tr>
<td>5</td>
<td>4500</td>
</tr>
<tr>
<td>6</td>
<td>4000</td>
</tr>
<tr>
<td>8</td>
<td>3500</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>n, No. of Rows of Tubes</th>
<th>( V_{\text{face}} ) ft/min</th>
</tr>
</thead>
<tbody>
<tr>
<td>3</td>
<td>900</td>
</tr>
<tr>
<td>4</td>
<td>800</td>
</tr>
<tr>
<td>5</td>
<td>700</td>
</tr>
<tr>
<td>6</td>
<td>600</td>
</tr>
<tr>
<td>8</td>
<td>500</td>
</tr>
</tbody>
</table>

These are, of course, only representative values, and the comparability of these values between the two tables is only approximate. They are, however, very useful for preliminary estimates.

3. The "Thermodynamic Limitation. " The thermodynamic limitation is nothing more than a heat balance and thus exists in all heat exchangers. But it is particularly critical in air-cooled exchangers because of the low mass rate at which air may be blown across the tube bank. Thus, if the total duty of the exchanger is \( Q \) Btu/hr, and if the air inlet and exit temperatures are \( t_i \) and \( t_o \), respectively, then the mass flow rate of air required is

\[
w_{\text{air}} = \frac{Q}{c_{p,\text{air}}(t_o - t_i)}
\]

(4.18)
For all practical purposes, the value of $c_{p,\text{air}}$ may be taken as constant at 0.24 Btu/lb °F.

The mass flow rate of air may be related to the face velocity $V_{\text{face}}$ and face area $A_{\text{face}}$ by

$$A_{\text{face}} = \frac{w_{\text{air}}}{\rho_{\text{air}}V_{\text{face}}}$$  \hspace{1cm} (4.19)

where $\rho_{\text{air}}$ is evaluated at air inlet temperature. Care must be taken of course to keep the units consistent.

But as noted in the discussion of the pumping limitation, there is a fairly narrow range of values of $V_{\text{face}}$ that can be provided in an air-cooled exchanger, and this is inversely related to the number of tube rows. By combining Eqs. (4.18) and (4.19),

$$\left(\frac{A_{\text{face}}}{Q}\right)_T = \frac{1}{c_{p,\text{air}}(t_o - t_i)\rho_{\text{air}}V_{\text{face}}}$$  \hspace{1cm} (4.20)

where the notation $(A_{\text{face}})_T$ indicates that this is the face area required by the "thermodynamic limitation". From Table 4.2 we may find typical design values for $V_{\text{face}}$ as a function of $n$ from the pumping limitation. Substituting these values into Eq. (4.20), noting the ft/min units used on $V_{\text{face}}$, allows us to calculate the face area required per unit of heat to be transferred as a function of $n$ given the values of $t_i$ and $t_o$.

**Example:** Assume that 100,000 lb/hr of water is to be cooled from 180 °F to 120 °F, using air available at 90 °F ($\rho_{\text{air}} = 0.0737$ lbm/ft³). If the air is heated to 140 °F, what face area is required on a thermodynamic basis as a function of the number of tube rows, using the values from Table 4.2:

Solution: From Eq. (4.20),

$$\left(\frac{A_{\text{face}}}{Q}\right)_T = \frac{1}{0.24 \frac{\text{Btu}}{\text{lbm} \cdot \text{°F}}(140 - 90) \frac{\text{°F}}{\text{F}}(0.0737 \frac{\text{lbm}}{\text{ft}^3})V_{\text{face}}} = \frac{0.01885}{V_{\text{face}}\text{ft/min}} \frac{\text{Btu}}{\text{hr}}$$

Since $Q = (100,000 \text{ lb/hr})(1 \text{ Btu/(lbm °F)})(180 - 120) \text{ °F} = 6 \times 10^6 \text{ Btu/hr}$, we may also calculate the face area required, the results are:

<table>
<thead>
<tr>
<th>$n$</th>
<th>$V_{\text{face}}$, ft/min</th>
<th>$(A_{\text{face}})_T$, ft²</th>
<th>$(A_{\text{face}})_T$, Btu/hr</th>
</tr>
</thead>
<tbody>
<tr>
<td>3</td>
<td>900</td>
<td>2.09 × 10⁻⁵</td>
<td>126</td>
</tr>
<tr>
<td>4</td>
<td>800</td>
<td>2.36 × 10⁻⁵</td>
<td>141</td>
</tr>
<tr>
<td>5</td>
<td>700</td>
<td>2.69 × 10⁻⁵</td>
<td>162</td>
</tr>
<tr>
<td>6</td>
<td>600</td>
<td>3.14 × 10⁻⁵</td>
<td>189</td>
</tr>
<tr>
<td>8</td>
<td>500</td>
<td>3.77 × 10⁻⁵</td>
<td>226</td>
</tr>
</tbody>
</table>

Alternatively, had we specified a tube and tube layout we could have carried out a similar set of calculations based upon Table 4.1. This would probably be a better criterion to use for actual design, but it is not quite so convenient as the procedure given.

Before developing the implications of the thermodynamic limitation further, let us take a look at the heat transfer rate limitation.
4. The "Rate Limitation." The basic equation for the rate of heat transfer is:

\[ A_0 = \frac{Q}{U_o(MTD)} \quad (4.21) \]

where \( A_0 \) and \( U_o \) must be on the same area basis, usually the total outside area of the finned tube. If reasonable estimates of \( U_o \) and MTD can be made quickly, the \( A_0 \) is easily found. This heat transfer area can be put on a comparable basis with the "Thermodynamic Limitation" if it is translated into the face area required by the equation:

\[ \left( A_{\text{face}} \right)_{HT} = \frac{A_0}{n A_{HT}^*} \quad (4.22) \]

where \( n \) is the number of tube rows, and \( A_{HT}^* \) is the finned tube heat transfer area per square foot of face area and per row. \( A_{HT}^* \) must be determined for each choice of tube and layout; for the case calculated in the previous example,

\[
\begin{align*}
A_{\text{fin}} / \text{ft of length} &= \frac{\pi}{4} \left[ (1.625 \text{ in})^2 - (0.75 \text{ in})^2 \right] \left[ \frac{2 \text{ sides/fin}}{144 \text{ in}^2/\text{ft}^2} \right] \left[ \frac{0.144 \text{ in}}{\text{fin}} \right] \left[ \frac{12 \text{ in}}{\text{ft}} \right] = 2.45 \text{ ft}^2/\text{ft} \\
A_{\text{root}} / \text{ft of length} &= \pi \left( \frac{0.75 \text{ in.}}{12 \text{ in.}/\text{ft}} \right) \left[ 1 \text{ ft} - \left( \frac{\text{fin}}{\text{in.}} \right) \left( \frac{0.019 \text{ in.}}{12 \text{ in.}} \right) \left( \frac{1 \text{ ft}}{12 \text{ in.}} \right) \right] = 0.163 \text{ ft}^2/\text{ft} 
\end{align*}
\]

Total heat transfer area per foot of tube = 2.61 ft²

With a 1.875 in. pitch, each foot of width of tube bank has \( \frac{12 \text{ in.}}{1.875 \text{ in.}} = 6.40 \) tubes per row, so

\[ A_{HT}^* = (6.40 \text{ tubes/ft-row})(2.61 \text{ ft}²/\text{ft}) = 16.70 \text{ ft}²/\text{ft}² \] of face, per row of tubes

By combining Eqs. (4.21) and (4.22) we may obtain the equation for the face area required by the rate limitation:

\[ \frac{\left( A_{\text{face}} \right)_{HT}}{Q} = \frac{1}{n A_{HT}^* U_o(MTD)} \quad (4.23) \]

A necessary condition of a final design is that the left hand sides of Eqs. (4.20) and (4.23) be equal. It is immediately observable that, other things being equal, \( \frac{(A_{\text{face}})_T}{Q} \) increases with increasing number of rows of tubes \( n \), whereas \( \frac{(A_{\text{face}})_{HT}}{Q} \) decreases with \( n \) increasing. Therefore, there is some cross-over point for any given design problem, and if this point can be approximately located by rapid calculations, the main features of the preliminary configuration can be set. In order to do that, it is necessary to estimate values of \( U_o \) and MTD.
As noted previously, the overall heat transfer coefficient is largely controlled in air-cooled exchangers by the air side coefficient. In the example given, the air-side coefficient was found to be 10.9 Btu/hr ft² °F at a face velocity of 600 ft/min and a maximum mass velocity of (1150 ft/min)(60 min/hr)(0.0765 lb m/ft) = 5280 lbm/ft² hr. Corresponding values of \( h \) would be 14.4 Btu/hr ft² °F at a face velocity of 900 ft/min and 8.3 Btu/hr ft² °F at a face velocity of 400 ft/min.

These values may be compared to typical values for the other terms given in Eq. (4.5) as follows:

<table>
<thead>
<tr>
<th>Term</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>( 1/R_{fi} ), Based on inside tube area</td>
<td>Heavy fouling: 100, Moderate fouling: 500, Light fouling: 2000</td>
</tr>
<tr>
<td>( k_w/\Delta x_w )</td>
<td>(Stainless steel liner, based on liner area): 2000</td>
</tr>
<tr>
<td>( k_w/\Delta x_w )</td>
<td>(Aluminum tube, based on root area): 15,000</td>
</tr>
<tr>
<td>( 1/R_c )</td>
<td>(Maximum contact resistance, based on contact area): 300</td>
</tr>
<tr>
<td>( 1/R_{fin} )</td>
<td>(Maximum resistance for aluminum fins, based on fin area): 125</td>
</tr>
</tbody>
</table>

From these values, we can see that the air-side resistance can vary from almost 100 percent of the total resistance (for, e.g., condensing steam) to about half of the total (for a viscous liquid, assuming that the tube has been chosen following the \( h_i A_o \approx h_i A_i \) criterion). Thus, for preliminary calculations, it is a reasonable approximation to estimate \( U_o = 10 \) Btu/hr ft² °F, based on total outside finned tube area, shading this to values as low as 5 Btu/hr ft² °F where low air velocities and/or low intube heat transfer coefficients are involved. Alternatively, for high tube-side coefficients and high air velocities (corresponding to shallow tube banks), the overall coefficient may be estimated as high as 12 Btu/hr ft² °F, based on total outside tube area.

To carry out the rate calculations, it is also necessary to have an estimate of the mean temperature difference. For preliminary estimation purposes, in cases where very close temperature approaches are not contemplated, it is often sufficient to use the arithmetic mean temperature difference:
AMTD = \frac{1}{2} [(T_i - t_o) + (T_o - t_i)] \quad (4.24)

The AMTD is always equal to or greater than the LMTD, the difference depending upon the ratio \(\frac{(T_i - t_o)}{(T_o - t_i)}\). When this ratio is close to unity, \(AMTD \approx LMTD\); as the ratio departs further from unity, the discrepancy between AMTD and LMTD becomes greater. Additionally, for one or two tube-side passes, the configuration correction factor \(F\) must be used to convert the LMTD to the MTD. For all practical air-cooler designs, \(1.0 \leq F \leq 0.8\), so a value of \(F = 0.9\) is a good estimate.

*If \(F < 0.8\) for a given problem, using Figs. (4.13) or (4.14), it is probably necessary to change design temperatures or number of tube-side passes to ensure a good design.

Example. Continue using the previous example, cooling water from 180°F to 120°F using air available at 90°F. Assume also that Wolverine H/R Trufin with \(d_i = 3/4\) in. and \(d_o = 1 5/8\) in., on a 1 7/8 in. equilateral triangular pitch is used; thus \(A_{HT} = 16.70 \text{ ft}^2/\text{ft}^2\) of face area per row.

If we start by estimating an exit air temperature of 140°F as before, the AMTD is quickly found to be:

\[ AMTD = \frac{1}{2} [(180 - 140) + (120 - 90)] = 35°F \]

The LMTD is 34.8°F. We may also wish to check \(F\) at this point:

\[ P = \frac{140 - 90}{180 - 90} = 0.556 \]
\[ R = \frac{180 - 120}{140 - 90} = 1.20 \]

and from Fig. 4.13 or 4.14 we get for \(F\) a value of about 0.82 if there is one tube-side pass and about 0.91 for two tube-side passes.

We may now set up a table for \(\left(\frac{A_{face}}{Q}\right)_{HT}\) similar to that for \(\left(\frac{A_{face}}{Q}\right)_T\) in the previous section:

Comparison of this table with the previous one indicates that the two closely correspond at the point of an air cooler with six rows of tubes and a face area of 190 ft² or in round numbers, a unit 20 feet long and 10 feet wide. The expectation that the fan requirements are probably within design range is due to the fact that we have used typical face velocities. The total heat transfer area (including fins) required is (6 rows)(10 feet wide) \(\left(\frac{12 \text{ in.}}{1.875 \text{ in. pitch}}\right)\) (20 feet long)(2.61 ft²/ft of tube) = 20,000 ft², or a total of 7680 ft of tube.

The next step is to verify the heat transfer coefficients and pressure drop in each side, and verify that two tube rows/pass are necessary and sufficient to maintain good tube-side velocity. From the air-side pressure drop calculation, a fan and driver specification may be obtained. There will almost certainly be some modifications in the approximate design obtained here, but this gives the designer a good place to start.
<table>
<thead>
<tr>
<th>n</th>
<th>No. of Tube Side Passes</th>
<th>F</th>
<th>MTD °F</th>
<th>V_{face} ft/min</th>
<th>U_o</th>
<th>(A_{face})_{HT} ft^2</th>
<th>(A_{face})_{HT}</th>
<th>Q Btu/hr</th>
<th>ft^2</th>
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</thead>
<tbody>
<tr>
<td>3</td>
<td>1</td>
<td>0.82</td>
<td>28.5</td>
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<td>12</td>
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<td>2</td>
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<td>800</td>
<td>11</td>
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<td></td>
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<td></td>
</tr>
<tr>
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<td>500</td>
<td>8</td>
<td>2.69x10^{-5}</td>
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<td></td>
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</tbody>
</table>