Chapter 6

Externally Finned Tubes

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1. Introduction

Finned-tube heat exchangers have been used for heat exchange between gases and liquids (single or two phase) for many years.

![Figure 6.1](image)

**Figure 6.1** Finned tube geometries used with circular tubes: (a) plate fin-and-tube used for gases, (b) individually finned tube having high fins, used for gases. (From Webb [1987]) (c) Low, integral-fin tube.

A plain surface geometry will increase the air-side $hA$ value by increasing the area ($A$). Use of enhanced fin surface geometries will provide higher heat transfer coefficients than a plain surface. To maintain reasonable friction power with low-density gases, the gas velocity is usually less than 5 m/s.

Important basic enhancement geometries include wavy and interrupted fins. Variants of the interrupted strip fin are also used with finned-tube heat exchangers for heat exchange to a tube-side fluid.
Because the gas-side heat transfer coefficient may be 5 to 20% that of the tube-side fluid, the use of closely spaced, high fins is desirable. High fin efficiency can be obtained, if the fin material has high thermal conductivity, e.g., aluminum or copper. If steel fins are required, fin efficiency considerations will dictate shorter or thicker fins.

Operational constraints, such as gas-side fouling may limit the fin density. Air-conditioning applications use 500 to 800 fins/m while process air coolers are usually limited to 400 fins/m.
Dirty, soot-laden gases may limit the fin density to 200 fins/m. Different correlations are required for the Figure 6.1a and b geometries. The fin material used also depends on the operating temperature and the corrosion potential. Listed below are the fin and tube materials used in a variety of applications:

1. **Residential air conditioning**: Aluminum fins and copper or aluminum tubes
2. **Automotive air conditioning**: Aluminum fins and aluminum tubes
3. **Automotive radiators**: Aluminum fins brazed to aluminum tubes, or copper fins soldered to brass tubes.
4. **Process industry heat exchangers**: Air-cooled condensers that may use aluminum fins on copper or steel tubes.
5. **Boiler economizers (锅炉省煤器) and heat-recovery exchangers**: Steel fins on steel tubes required by the higher operating temperature.
In addition to describing the various fin geometries and their performance characteristics, this chapter compares the performance of alternative heat exchanger and fin configurations. Then heat exchanger and enhanced fin geometries that will yield the highest performance per unit heat exchanger core weight are identified. Finally, possible improvements in the air-side surface geometry are considered.

2. The geometric parameters and the Reynolds number

2.1 Dimensionless Variables

The flow pattern in finned-tube heat exchangers is very complex, due to its three-dimensional nature and flow separations. The use of enhanced fin geometries introduces further complications. The geometric and flow variables that affect the heat transfer coefficient and friction factor are the following.
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(1) **Flow variables**: Air velocity \((u)\), viscosity \((\mu)\), density \((\rho)\), thermal conductivity \((k)\), and specific heat \((c_p)\).

(2) **Tube bank variables**: Tube root diameter \((d_o)\), transverse tube pitch \((S_t)\), row pitch \((S_r)\), tube layout (staggered or inline), and the number of rows \((N)\).

(3) **Fin geometry variables**: For a plain fin, these are the fin pitch \((p_f)\), fin height \((e)\), fin thickness \((t)\). If, for example, an enhanced wavy fin geometry is used the added variables are the wave height \((e_w)\), the wave pitch \((p_w)\), and the wave shape.

Thus, there are seven geometry variables for a plain fin (excluding the tube layout) and five flow variables. Two additional variables are introduced to account for the wavy fin geometry. **Dimensional analysis** specifies that the number of possibly important dimensionless groups is the number of variables minus the number of dimensions.

The dimensionless flow variables typically used in correlations are the **Reynolds number** and the **Prandtl number**. For heat transfer, one has the **Nusselt number** or the **Stanton number**. For pressure drop, one uses the **friction factor**.
2.2 Definition of Reynolds Number

The basic definition of the Reynolds number is \( \frac{L_c G}{\eta} \), where \( L_c \) is a characteristic dimension and \( G \) is usually defined as the mass velocity in the minimum flow area.

As previously stated, there are seven dimensions associated with a plain fin geometry, and nine for the wavy fin geometry. Hence, there is no unique characteristic dimension. Hence, there are eight possible values of \( L_c \) for definition of the Reynolds number of the plain fin geometry.

Two different characteristic dimensions have been used to define \( L_c \) in the Reynolds number. They are the tube diameter \( (d_o) \) or the hydraulic diameter \( (D_h) \).

Kays and London [1984] choose to use the hydraulic diameter for the characteristic dimension for all situations, including bare and finned-tube banks. There is no evidence to suggest that hydraulic diameter is a better choice. In fact, there is evidence that the tube diameter may be a better choice for finned-tube banks.
For fully developed flow in tubes, one defines **laminar and turbulent** regimes. Do such regimes also exist for finned-tube banks? To evaluate this, consider the case of the Figure 6.1a geometry.

![Figure 6.1 (a) plate fin-and-tube used for gases](image)

Assume that the geometry uses $d_o=19$ mm with an equilateral triangular pitch of $S_t=44.45$ mm, and 472 fins/m with 0.2 mm thickness. Assume air enters the exchanger at 3 m/s and 20° C. The mass velocity in the minimum flow area ($G$) is 7.34 kg/m²s and the hydraulic diameter is 3.68 mm. The Reynolds numbers based on $d_o$ and $D_h$ are **8710 and 1640**, respectively. Is the flow laminar or turbulent? Based on $Re_{Dh}$, one would say it is laminar. But, based on tube diameter ($Re_d$), one would say it is turbulent. In reality, it **exhibits some of both characteristics**.
If the tubes were not present, the flow geometry would be a parallel plate channel, for which $D_h = 3.82$ mm. The Reynolds number is 1588, which is clearly laminar. However, the tubes shed eddies, which wash over the fin surface and provide mixing of the flow.

If the Reynolds number based on hydraulic diameter were dominant over the Reynolds number based on tube diameter, one would expect that the Nu and/for different fin pitches would tend to fall on one line. It is shown below that this is not the case.

2.3 Definition of the Friction Factor

This course strives to use only the Fanning friction factor ($f$), defined as follow:

$$f = \frac{\Delta \rho D_h}{2L G^2} \quad (6.1)$$

However, other friction factor definitions are frequently used for tube banks (bare and finned). A common definition for tube banks is given the symbol $f_{tb}$. It is related to the Fanning friction factor by the equation:

$$f_{tb} = \frac{fL}{D_h} \quad (6.2)$$

where $N$ is the number of tube rows in the flow direction and $L$ is the flow depth. For bare or finned tubes, $L = S_f (N-1) + d_e$, where $d_e$ is diameter over the fins. For a bare tube bank, $d_e = d_o$. 

3. Plain Plate-Fins on Round Tubes

Figure 6.1a shows the finned-tube geometry with continuous, plain plate fins in a staggered tube layout. An inline tube geometry is seldom used because it provides substantially lower performance than the staggered tube geometry.

3.1 Effect of Fin Spacing

Rich [1973] measured heat transfer and friction data for the Figure 6.1a geometry having plain fins, four rows deep, on 12.7-mm-diameter tubes equilaterally spaced on 32-mm centers. The tubes and fins were made of copper, and the fins were solder-bonded to min.

Figure 6.3 Heat transfer and friction characteristics of a four-row plain plate fin heat exchanger for different fin spacings.
Rich proposed that the friction drag force is the sum of the drag force on a bare tube bank ($\Delta p_t$) and the drag caused by the fins ($\Delta p_f$). The difference between the total drag force and the drag force associated with the corresponding bare tube bank is the drag force on the fins. Thus, the friction component resulting from the fins is given by

$$f_f = (\Delta p - \Delta p_t)2A_c \rho/(G^2 A_f)$$  \hfill (6.3)

The term $\Delta p_t$ is that measured for a bare tube bank of the same geometry, without fins. Both $\Delta p$ drop contributions are evaluated at the same minimum area mass velocity.

Figure 6.4 Plot of the $j$ factor and the fin friction vs. $Re_{St}$
Figure 6.4 shows that the $j$ factor is a function of velocity in the minimum flow area ($G_\phi$), and is essentially independent of fin spacing. At the same mass velocity ($G_\phi$), the bare tube bank heat transfer coefficient is 40% larger than that of the finned-tube bank.

Normally, the $j/f$ ratio will increase as the fin spacing is reduced, because the fractional parasitic drag associated with the tube is reduced. Use of the Reynolds number based on $S_f$ has no real significance, as all geometries tested had the same $S_f$.

Figure 6.4 may be regarded as evidence that the Reynolds number based on hydraulic diameter will not correlate the effect of fin pitch.

In a later study, Rich [1975] used the same heat exchanger geometry with 551 fins/m to determine the effect of the number of tube rows on the $j$ factor. Figure 6.5 shows the average $j$ factor (smoothed data fit) for each exchanger as a function of $Re_{SI}$.

The row effect is greatest at low Reynolds numbers and becomes negligible at $Re_{SI}>15,000$. 
3.2 Correlations for Staggered Tube Geometries
Correlations to predict the $j$ and $f$ factors vs. Reynolds number for plain fins on staggered tube arrangements were developed by McQuiston [1978], Gray and Webb [1986], Kim et al. [1999], and Wang et al. [2000a].

The McQuiston and the Gray and Webb heat transfer correlations are comparable in accuracy. However, the Gray and Webb friction factor correlation is much more accurate than that of McQuiston.
The Gray and Webb [1986] heat transfer correlation (for four or more tube rows of a staggered tube geometry) is

\[ j_4 = 0.14 \text{Re}_d^{-0.328} \left( \frac{S_j}{S} \right)^{-0.302} \left( \frac{S}{d_o} \right)^{0.031} \]  

(6.3)

Equation 6.3 assumes that the heat transfer coefficient is stabilized by the fourth tube row, hence the \( j \) factor for more than four tube rows is the same as that for a four-row exchanger. The correction for rows less than four is based on correlation of the Figure 6.5 data, and is given by

\[ \frac{j_N}{j_4} = 0.99 \left[ 2.24 \text{Re}_d^{-0.092} \left( \frac{N}{4} \right)^{0.031} \right]^{0.607(4-N)} \]  

(6.4)

The Gray and Webb [1986] friction correlation assumes that the pressure drop is composed of two terms. The first term accounts for the drag force on the fins, and the second term accounts for the drag force on the tubes. The friction factor of the heat exchanger is given by

\[ f = f_f \frac{A_f}{A} + f_j (1 - \frac{A_f}{A})(1 - \frac{t}{\rho_f}) \]  

(6.5)

where

\[ f_f = 0.508 \text{Re}_d^{-0.521} \left( \frac{S_f}{d_o} \right)^{1.318} \]  

(6.6)
3.3 Correlations for Inline Tube Geometries

Schmidt [1963] reports data and a correlation for the inline geometry. However, little use exists for an inline tube arrangement. This is because tube bypass effects substantially degrade the performance of an inline tube arrangement.

3.4 Plain Individually Finned Tubes

3.4.1 Plain Individually Finned Tubes

Extruded fins or helically wrapped fins on circular tubes, as shown by Figure 6.1b, are frequently used in the process industries and in combustion heat-recovery equipment.

Figure 6.1 (b) individually finned tube having high fins, used for gases

Both plain and enhanced fin geometries are used. A staggered tube layout is used, especially for high fins ($e/d_o>0.2$).

The recommended correlations for a staggered tube layout are made by Briggs and Young [1963] for heat transfer and Robinson and Briggs [1966] for pressure drop.
Equation 6.7 is based on airflow over 14 equilateral triangular tube banks and covers the following ranges: \(1100 \leq \text{Re}_d \leq 18,000\), \(0.13 \leq s/e \leq 0.63\), \(1.0 \leq s/t \leq 6.6\), \(0.09 \leq e/d_o \leq 0.69\), \(0.01 \leq t/d_o \leq 0.15\), \(1.5 \leq S_f/d_o \leq 8.2\). The standard deviation was 5.1%.

The isothermal friction correlation of Robinson and Briggs [1966], which we have rewritten in terms of the tube bank friction factor, is

\[
f_{ib} = 9.47 \text{Re}_d^{-0.316} \left(\frac{S_f}{d_o}\right)^{-0.515}
\]

\[
(6.8)
\]

### 3.4.2 Low Integral-Fin Tubes

Rabas et al. [1981] developed more accurate \(j\) and \(f\) correlations for low fin heights and small fin spacings. The correlations are given below with the exponents rounded off to two significant digits.

\[
j = 0.292 \left(\frac{d_f G_f}{\mu}\right)^{0.234} \left(\frac{s}{d_o}\right)^{0.25} \left(\frac{e}{s}\right)^{0.76} \left(\frac{d_f}{d_o}\right)^{0.73} \left(\frac{d_f}{S_f}\right)^{0.71} \left(\frac{S_f}{S_o}\right)^{0.38}
\]

\[
(6.9)
\]

Where \(n = -0.415 + 0.0346(d_f/s)\). The friction correlation is

\[
f = 3.805 \left(\frac{d_f G_f}{\mu}\right)^{-0.234} \left(\frac{s}{d_o}\right)^{0.25} \left(\frac{e}{s}\right)^{0.76} \left(\frac{d_f}{d_o}\right)^{0.73} \left(\frac{d_f}{S_f}\right)^{0.71} \left(\frac{S_f}{S_o}\right)^{0.38}
\]

\[
(6.10)
\]

The equations are valid for staggered tubes with \(N \geq 6\), \(5000 \leq \text{Re}_f \leq 25,000\), \(1.3 \leq s/e \leq 1.5\), \(0.01 \leq s/t \leq 0.06\), \(e/d_o \leq 0.10\), \(0.01 \leq t/d_o \leq 0.02\), and \(1.3 \leq S_f/d_o \leq 1.5\).
3.5 Enhanced Plate Fin Geometries with Round Tubes

The wavy (or herringbone) fin and the offset strip fin (also referred to as parallel louver) geometries are the major enhanced surface geometries used on circular tubes.

The combination of tubes plus a special surface geometry establishes a very complex flow geometry. The heat transfer coefficient of the wavy fin is typically 50 to 70% greater than that of a plain (flat) fin.

3.5.1 Wavy Fin

There are two basic variants of the wavy fin geometry as illustrated in Figure 6.6. They are the smooth wave and the herringbone configurations.

Figure 6.6 Two basic geometries of the wavy fin; (a) herringbone wave, (b) smooth wave.
Much work has been done on the herringbone wave geometry. However, very limited work has been done on the smooth wave geometry. Goldstein and Sparrow [1977] used a mass transfer technique to measure local and average mass transfer coefficients on a model having herringbone configuration. At Re=1000 (based on fin spacing), the wave configuration yielded a 45% higher mass transfer coefficient compared with the plain fin counterpart. They proposed that the enhancement results from Goetler vortices that form on concave wave surfaces.

3.5.2 Offset Strip Fins
The OSF concept (also known as “slit fins”) has been applied to finned-tube heat exchangers with plain fins for dry cooling towers and for refrigerant condensers.

Figure 6.7 Comparison of the heat transfer coefficient for the OSF and plain fin geometries for 9.5-mm-diameter tubes, 525 fins/m, and 0.2-mm fin thickness as reported by Nakayama and Xu [1983].
At 3 m/s air velocity, the OSF provides a 78% higher heat transfer coefficient than the plain fin. For the same louver geometry, the OSF will provide a higher heat transfer coefficient, when used in the plate-and-fin-type heat exchanger.

The OSF shown in Figure 5.4 provides 150% higher heat transfer coefficient than the plain fin at the same velocity.

Generalized empirical correlations for $j$ and $f$ vs. Re have not been developed for OSF geometry on round tubes.

### 3.5.3 Louvered Fin

The louver geometry discussed in Section 5.3 has been applied to finned-tube heat exchangers. Louver patterns are formed on the fin area between tubes. Care must be exercised in louvering the fin surface, because the louvers can cut the conduction path from the tube.
3.5.4 Vortex Generators

For circular finned tubes, a low heat transfer coefficient exists in the wake region behind the tubes, especially at low Reynolds numbers. Vortex generators on the fin surface help to reduce the width of the wake zone, and thus improve heat transfer in the wake region. Studies of vortex generators on circular finned tubes have shown that the performance improvement is not as great (relative to a plain fin surface) as for flow in channels without any tubes.
An explanation for this is that in the circular finned tube geometry, horseshoe vortices generated in front of the tube cause longitudinal horseshoe vortices, which increase the heat transfer over the fins along the path. These horseshoe vortices provide significant enhancement on the fins, relative to that provided by vortex generators.

Figure 6.8 Fin configurations tested by Lozza and Melo [2001]: (a) louver fin A, (b) louver fin B, (c) louver fin with vortex generator.

3.6 Enhanced Circular Fin Geometries

Figure 6.9 Enhanced circular fin geometries: (a) plain circular fin; (b) slotted fin; (c) punched and bent triangular projections; (d) segmented fin; (e) wire loop extended surface.

Figure 6.10 Segmented or spine fin geometries used in air conditioning applications, (a) (From LaPorte et al. [1979].) (b) Described by Abbott et al. [1980] and tested by Eckels and Rabas [1985]
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Figure 6.11 $j$ and $f$ vs. $Re_d$ characteristics of the Figure 6.10 enhanced fin geometries compared with a plain, circular fin geometry as reported by Eckels and Rabas [1985].

<table>
<thead>
<tr>
<th>Curve</th>
<th>Geometry</th>
<th>$S/d_o$</th>
<th>$d_o/d_L$</th>
<th>Fins/m</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Plain fin</td>
<td>1.0</td>
<td>3.47</td>
<td>787</td>
</tr>
<tr>
<td>2</td>
<td>Figure 6.10a</td>
<td>0.91</td>
<td>3.47</td>
<td>787</td>
</tr>
<tr>
<td>3</td>
<td>Figure 6.10b</td>
<td>0.95</td>
<td>2.45</td>
<td>787</td>
</tr>
</tbody>
</table>

Figure 6.12 Comparison of segmented fins (staggered and inline tube layouts) with plain, staggered fin tube geometry as reported by Weierman et al. [1978]. $S/d_o=2.25$, $e/d_o=0.51$, $s/e=0.12$, $w_f/e=0.17$. 43/72

Figure 6.12 Comparison of segmented fins (staggered and inline tube layouts) with plain, staggered fin tube geometry as reported by Weierman et al. [1978]. $S/d_o=2.25$, $e/d_o=0.51$, $s/e=0.12$, $w_f/e=0.17$. 44/72
### 3.7 Oval and Flat Tube Geometries

#### 3.7.1 Oval vs. Circular Individually Finned Tubes

Oval and flat cross-sectional tube shapes are also applied to individually finned tubes.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Circular</th>
<th>Oval</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tube dia. (d)</td>
<td>29</td>
<td>199.9/35.2</td>
</tr>
<tr>
<td>Fin. ht. (e)</td>
<td>9.8</td>
<td>10/9.3</td>
</tr>
<tr>
<td>Fin. htk. (t)</td>
<td>0.4</td>
<td>0.4</td>
</tr>
<tr>
<td>Face pitch (S/d)</td>
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<td>1.05</td>
</tr>
<tr>
<td>Row pitch (S/d)</td>
<td>1.15</td>
<td>1.04</td>
</tr>
<tr>
<td>Fins/m</td>
<td>312</td>
<td>312</td>
</tr>
</tbody>
</table>

![Figure 6.13](image)

*Figure 6.13* Heat transfer and friction characteristics of circular and oval finned tubes in a staggered tube layout as reported by Brauer [1964]

The oval tubes gave 15% higher heat transfer coefficient and 25% less pressure drop than the circular tubes.

The performance advantage of the oval tubes results from lower form drag on the tubes and the smaller wake region on the fin behind the tube.

The use of oval tubes may not be practical unless the tube-side design pressure is sufficiently low.
3.7.2 Flat Extruded Aluminum Tubes with Internal Membranes

Higher design pressures are possible using flattened aluminum tubes made by an extrusion process. Figure 6.2f shows a patented finned tube concept made from an aluminum extrusion. Such tubes can be made with internal membranes, which strengthen the tube and allow for a high tube-side design pressure.

The Figure 6.2e flat tube geometry offers significant advantages over the Figure 6.2a or Figure 6.2b strip fin geometries on round tubes.

(1) The airflow is normal to all of the narrow strips on the Figure 6.2e geometry, which is not the case for the Figure 6.2a spine fin design. Further, the wake dissipation length decreases in the direction of the fin base in the Figure 6.2a geometry.
(2) A low-velocity wake region does not occur behind the tubes of the Figure 6.2e flat tube geometry. The low wake velocity of the Figure 6.2a and b geometries causes a substantial reduction of the heat transfer coefficient, as documented by Webb [1980].

(3) The fraction of the Figure 6.2e surface that is louvered is substantially greater than in the Figure 6.2b geometry. If a greater area distribution of louvers were provided in the Figure 6.2b geometry, the fin efficiency would substantially decrease. This is because the slits would cut the heat conduction path from the base tube.

(4) Further, the low projected area (投影面积) of the Figure 6.2e flat tube will result in lower profile drag.
### 3.8 Row Effects—Staggered and Inline Layouts

The published tube bank correlations are generally for deep tube banks and do not account for row effects. The heat transfer coefficient will decrease with rows in an inline bank due to the bypass effects. However, the coefficient increases with number of tube rows in a staggered bank. This is because the turbulent eddies shed from the tubes cause good mixing in the downstream fin region.

As an approximate rule, one may assume that the heat transfer coefficient for a staggered tube bank of the Figure 6.2a and b finned tubes has attained its asymptotic value at the fourth tube row.

Inline tube banks generally have a smaller heat transfer coefficient than staggered tube banks. At low $Re_d$ ($Re_d<1000$) with deep tube banks ($N\geq8$), Raba and Huber [1989] show that the heat transfer coefficient may be as small as 60% of the staggered tube value.

The heat transfer coefficient of the inline bank increases as the $Re_d$ is increased; at $Re_d=50,000$ with $N\geq8$, the inline to staggered ratio may approach 0.80.
There is a basic difference in the flow phenomena in staggered and inline finned-tube banks.

**Figure 6.13** $\text{NuPr}^{-1/3}$ vs. $\text{Re}_d$ for inline and staggered banks of circular finned tubes with plain fins, as reported by Brauer [1964].

**Figure 6.14** Flow patterns observed by Brauer [1964] for (a) staggered and (b) inline finned tube banks.
The bypass effect reduces the performance the inline geometry as the number of rows increases!!

Figure 6.15 Row effect of inline tube banks for the Figure 6.20 segmented fin tubes ($S/d_0=S_i/d_0=2.25$).

3.9 Heat Transfer Coefficient Distribution

Although the standard fin efficiency calculation assumes the heat transfer coefficient is constant over the fin surface area, local measurements have shown this is not the case. The flow accelerates around the tube and forms a wake region behind the tube. This causes local variations of the heat transfer coefficient.

Figure 6.16 Two-row finned tube geometry simulated by Saboya and Sparrow [1974].
Figure 6.17 Local Sherwood number distribution measured by Saboya and Sparrow [1974] for a one-row plate fin geometry. The tube is located at $0.27 \leq x/L \leq 0.73$.

Figure 6.18 Distribution of mass transfer coefficients (m$^3$/m$^2$-hr) on a two-row plate finned tube measured by Krückels and Kottke [1970]. (a) $Re_d=1160$, (b) $Re_d=5800$.
Figure 6.19 Distribution of mass transfer coefficients (m³/m²-hr) on a single circular finned tube (212 fins/m) measured by Krückels and Kottke [1970]. (a) Re_d=1940, (b) Re_d=9700

3.10 Performance Comparison of Different Geometries

3.10.1 Geometries Compared

Table 6.1 Heat Exchanger Geometries Compared by Webb and Gupte [1990]

<table>
<thead>
<tr>
<th>Geometries</th>
<th>Spine</th>
<th>Wavy</th>
<th>Slit</th>
<th>CLF</th>
<th>OSF</th>
<th>Louver</th>
</tr>
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<tbody>
<tr>
<td>Figure</td>
<td>6.2a</td>
<td>6.2c</td>
<td>6.2b</td>
<td>6.2d</td>
<td>6.20b</td>
<td>6.20c</td>
</tr>
<tr>
<td>d_a (mm)</td>
<td>9.52</td>
<td>9.52</td>
<td>9.52</td>
<td>9.52</td>
<td></td>
<td></td>
</tr>
<tr>
<td>b (mm)</td>
<td>3.46</td>
<td>3.46</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>a (mm)</td>
<td>7.87</td>
<td>8.38</td>
<td>8.38</td>
<td>8.38</td>
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<td>8.38</td>
</tr>
<tr>
<td>t (mm)</td>
<td>0.76</td>
<td>0.76</td>
<td>0.76</td>
<td>0.76</td>
<td>0.76</td>
<td>0.76</td>
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<tr>
<td>S_t (mm)</td>
<td>25.4</td>
<td>23.62</td>
<td>25.4</td>
<td>25.4</td>
<td>21.84</td>
<td>21.84</td>
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<tr>
<td>S_o (mm)</td>
<td>25.4</td>
<td>20.60</td>
<td>21.60</td>
<td>21.60</td>
<td></td>
<td></td>
</tr>
<tr>
<td>L_p (mm)</td>
<td>1.98</td>
<td>1.59</td>
<td>1.59</td>
<td>1.59</td>
<td></td>
<td></td>
</tr>
<tr>
<td>n_L</td>
<td>4</td>
<td>5</td>
<td>5</td>
<td>5</td>
<td></td>
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<tr>
<td>θ (degrees)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>20</td>
<td>20</td>
</tr>
</tbody>
</table>

a: Major diameter for rectangular tube cross section, m; b: Minor diameter for rectangular tube cross section, m; L_p: Strip flow length of OSF or louver pitch of louver fin, m; n_L: Number of louvers in airflow depth, dimensionless.
3.10.2 Analysis Method

Following are the sources of the various correlations, or data, used to predict the air-side heat transfer and friction characteristics:

1. Figure 6.2a spine fin: Scaled data of Eckels and Rabas [1985]
2. Figure 6.2b slit fin: Mori and Nakayama [1980] data
3. Figure 6.2c wavy fin: Webb [1990] wavy fin correlation
4. Figure 6.2d CLF: Hatada and Senshu [1984] data
5. Figure 6.20b OSF: Wieting [1975] correlation
6. Figure 6.20c louver fin: Davenport [1984] correlation
The Kandlikar [1987] correlation was used to predict the tube-side heat transfer coefficient for vaporization of R-22 in plain tubes and flat tubes (using the hydraulic diameter).

![Figure 6.21](a) $h_o$ vs. air velocity, (b) air $\Delta p$ vs. air velocity.

The calculation methodology used the VG-1 criterion described in Chapter 3. The objective is to reduce the heat exchanger size and weight. The case of an R-22 evaporator is used to compare the performance of the various geometries. The evaporator operates at the following conditions: heat duty (2408 W), airflow rate (0.53 m$^3$/s), refrigeration saturation temperature (2.77°C), inlet air temperature (10°C), fins/m for the reference spine fin (787), air $\Delta p$ (0.74 mm w.g.). The air frontal velocity was allowed to vary from 0.76 to 1.78 m/s, and the fin/m is allowed to vary from 314 to 867 for each air-side surface geometry.
The calculation procedure used to size the heat exchanger for a specified frontal velocity is as follows:

1. The required $kA$ is calculated for the specified airflow rate, heat duty, inlet air temperature, and refrigerant saturation temperature.
2. Set the fins per meter at the lowest value (314 fins/m) and the number of tube rows in the airflow direction ($N$) to one.
3. Calculate the heat exchanger frontal area, and the air-side and tube-side heat transfer coefficients for the specified air frontal velocity.
4. Calculate the available $kA$ from:
   \[
   \frac{A_p N}{kA} = \frac{A_p N}{h_i A_i} + \frac{A_p N}{\eta h_o A_o}
   \]
5. Calculate the air pressure drop.
6. If the available $kA$ is less than the required value, increase the fins/m value by 20 fins/m, and repeat steps 2 through 4. If the available $kA \geq$ required $kA$, then the pressure drop is checked. If this is within the specified 0.74 mm w.g. limit, an acceptable solution exists. If no design is obtained, the number of rows is increased by one and the calculations are repeated from step 2.
This methodology results in selecting the heat exchanger having the minimum number of tube rows, which should yield the minimum material cost.

3.10.3 Calculated Results

Table 6.2 compares the various performance parameters of the six heat exchanger configurations. The Figure 6.18b spine fin is taken as the reference for comparison. Each design listed in Table 6.3 provides the same heat duty and operates at the same air-side pressure drop and airflow rate.

<table>
<thead>
<tr>
<th>Table 6.2 Comparison of All-Aluminum Heat Exchangers (Plain Tubes with Two Refrigerant Circuits)</th>
</tr>
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<tbody>
<tr>
<td><strong>Row</strong></td>
</tr>
<tr>
<td>---------</td>
</tr>
<tr>
<td>Fins/m</td>
</tr>
<tr>
<td>u_in (m/s)</td>
</tr>
<tr>
<td>h_e (W/m²·K)</td>
</tr>
<tr>
<td>η</td>
</tr>
<tr>
<td>G_f (kg/m²·s)</td>
</tr>
<tr>
<td>h_i (W/m²·K)</td>
</tr>
<tr>
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<tr>
<td>w_blue (kg)</td>
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<tr>
<td>w_tot (kg)</td>
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</tbody>
</table>
3.11 Conclusions

Enhanced surfaces are routinely used for application to gases. The dominant enhancement types are wavy or some form of interrupted strip fin. When used on circular tubes, a heat transfer enhancement of 80 to 100% is practically achieved.

Some advances were made on analytical or numerical models to predict the heat transfer performance of banks of high finned tubes. Power-law empirical correlations have been developed for plain fins.

Although some work has been done to develop correlations for enhanced fins on circular tubes, they are not sufficiently general to account for the many geometric variables involved. The problem is complicated by the row effect of inline and staggered tube arrangements.

Both the individually finned-tube and plate fin-and-tube geometries are used. Enhanced surface geometries are available for both geometries. Preference of the geometry type is dependent on application and manufacturing cost considerations, rather than performance.
Higher performance can be obtained from **oval or flat tubes**. Oval tubes may not be practical for high tube-side design pressure. If aluminum is an acceptable material, **extruded aluminum tubes having internal membranes** offer potential for significant performance improvement, relative to round tube designs. High tube-side design pressures can be met. The resulting brazed aluminum heat exchangers offer many advanced technology possibilities. High-performance fin geometries are quite adaptable to this heat exchanger concept.

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**Thank you for your attention!**

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