Chapter 5
Plate-and-Fin Extended Surfaces

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1. Introduction

This chapter discusses enhanced extended surface geometries for the plate-and-fin heat exchanger geometry, which is shown in Figure 5.1.

![Figure 5.1 A cross-flow plate-and-fin heat exchanger geometry](image)

Normally, at least one of the fluids used in the plate-and-fin geometry is a gas. In forced convection heat transfer between a gas and a liquid, the heat transfer coefficient of the gas is typically 5 to 20% that of the liquid. The use of extended surfaces will reduce the gas-side thermal resistance. However, the resulting gas-side resistance may still exceed that of the liquid. In this case, it will be advantageous to use specially configured extended surfaces, which provide increased heat transfer coefficients 50-150% higher than those given by plain extended surfaces.
For heat transfer between gases, such enhanced surfaces will provide a substantial heat exchanger size reduction. There is a trend toward using enhanced surface geometries with liquids for cooling electronic equipment. Data taken for gases may be applied to liquids if the Prandtl number dependency is known. In the absence of specific data on Prandtl number dependency, one may assume \( \text{St} \propto \text{Pr}^{-2/3} \).

Figure 5.2 Plate-fin exchanger surface geometries
Figure 5.2 shows six commonly used enhanced surface geometries. Typical fin spacings are 300 to 800 fins/m. Vortex generators (shown later) can also provide significant enhancement. Because of the small hydraulic diameter and low density of gases, these surfaces are usually operated with $500 < \text{Re}_{Dh} < 1500$. Although increased performance will exist at higher Reynolds numbers (turbulent regime), fan-power limitations generally limit operation to the above low Reynolds number range.

To be effective, the enhancement technique must be applicable to low-Reynolds-number flows. The use of surface roughness will not provide appreciable enhancement for such low-Reynolds-number flows. Two basic concepts have been extensively employed to provide enhancement:

(1) Repeated growth and wake destruction of boundary layers. This concept is employed in the offset-strip fin, the louvered fin, and to some extent, in the perforated fin.
(2) Special channel or surface shapes that promote fluid mixing by secondary flows (Taylor-Goertler vortices) and boundary layer separation within the channel. Vortex generators are used to generate longitudinal vortices, which wash the downstream surface.

2. Offset-strip Fin

The offset-strip fin (OSF) is possibly the most important enhancement concept that has been developed to enhance heat transfer with gases. It is effective at both high and low Reynolds numbers. The enhancement mechanism of the louver fin discussed in Section 5.3 is basically the same as that of the OSF. The OSF (and louver fin) geometries are used in both plate-and-fin and finned tube heat exchangers.
2.1 Enhancement Principle

Figure 5.3 illustrates the enhancement principle of the Fig. 5.2d OSF. A laminar boundary layer is developed on the short strip length, followed by its dissipation in the wake region between strips. Typical strip lengths are 3 to 6 mm, and the Reynolds number is well within the laminar region.

![Figure 5.3 Plate-fin exchanger surface geometries](image)

The enhancement provided by an OSF of $L_p/D_h=1.88$ is shown by Figure 5.4. The table in Fig. 5.4 compares the dimensions with the OSF scaled to the same hydraulic diameter as that of the plain fin. The scaled dimensions (fin height and fin thickness) of the OSF are approximately equal to those of the plain fin surface. Therefore, the performance difference is due to the enhancement provided by the short strip lengths (6.6 mm) of the OSF.
The local flow and heat transfer characteristics in an OSF array were investigated by DeJong and Jacobi [1997]. Figure 5.5 shows the flow visualization results. The test sample consists of 8 rows of offset strips with $L_p=2.54$ cm, $s=1.28$ cm, and $t=0.32$ cm.

At $Re_{Dh}=380$, the flow is steady and laminar. At $Re_{Dh}=550$, a periodic secondary structure is formed in the latter region of the test section, and the wake exhibits a feathery appearance. At $Re_{Dh}=630$, the wake takes a roughly sinusoidal appearance.
In Figure 5.6, the heat transfer coefficients (converted from the mass transfer data) are compared with theoretical predictions from Shah and London [1978]. The predictions are for thermal entry length laminar flow between parallel plates at constant temperature. The theoretical values are provided for continuous plates, and interrupted plates.
At low Reynolds numbers ($R_e_{Dh} < 430, v < 0.4 \text{ m/s}$), where the flow is steady and laminar, the data are in reasonable agreement with theoretical values. At higher Reynolds numbers, the experimental data for the interrupted surface are significantly higher than the theoretical values, because of the vortex shedding.

The theoretical Shah and London predictions were developed based on simplified approximation to full momentum and energy equations, and do not account for the complex flow characteristics such as vortex shedding.
2.2 PEC Example 5.1

A more realistic comparison of the OSF and plain fin performance is obtained using Case VG-1 of Table 3.1. Assume that the plain fin (subscript \( p \)) operates at \( Re_{Dh,p} = 834 \). Equation 3.7, which applies to Case VG-1 of Table 3.1, defines the value of \( G/G_p \) at which both surfaces satisfy

\[
\frac{1}{2\sqrt{\frac{h}{h_p}}} = \frac{1}{\sqrt{\frac{P}{P_p}}} = \frac{W}{W_p}.
\]

Hence, it must satisfy

\[
\frac{G}{G_p} = \left( \frac{j/j_p}{f/f_p} \right)^{1/2} \tag{5.1}
\]

It is necessary to iteratively solve Eq. 5.1 for \( G/G_p \), reading the \( j \) and \( f \) values for the OSF surface from Figure 5.4. The final solution of Eq. 5.1 yields \( G/G_p = 0.923 \). So, \( Re_{Dh} = 0.923 \times 834 = 770 \). The \( j \) and \( f \) values of the OSF at \( Re_{Dh} = 770 \) are now known.

The surface area ratio \( (A/A_p) \) is obtained using Eq. 3.2, repeated here as Eq. 5.2

\[
\frac{hA}{h_p A_p} = j \frac{A}{j_p A_p} \frac{G}{G_p} \tag{5.2}
\]
Substituting the known values of $j/j_p$ and $G/G_p$ in Eq. 5.2, one calculates $A/A_p=0.446$. The OSF geometry requires only 44.6% as much surface area to provide the same $hA$ as the plain fin geometry. Using Eq. 3.3, one may show that the flow frontal area must be increased 10% to maintain $P/P_p=1$.

2.3 Analytically Based Models for $j$ and $f$ vs. Re

Kays [1972] proposed a simple, approximate model to predict the $j$ and $f$ vs. Re curves for the OSF. The model assumes (1) laminar boundary layers on the fins, (2) the boundary layers developed on the fin are totally dissipated in the wake region between fins.

Using the equations for laminar flow over a flat plate in a free stream, Kays’ analysis gives

$$j = 0.664 \left( \frac{R_e}{L} \right)^{0.5}$$  \hspace{1cm} (5.3)

$$f = \frac{C_D \delta}{2 L_p} + 1.328 \left( \frac{R_e}{L} \right)^{0.5}$$  \hspace{1cm} (5.4)

The first term in Eq. 5.4 accounts for the form drag on the plate. The form drag contribution is proportional to the fin thickness ($\delta$), and it has a negligible effect on the heat transfer coefficient. Kays suggests use of $C_D = 0.88$ based on potential flow normal to a thin plate.
Note that the Kays model neglects heat transfer and friction on the confining walls of the OSF channel. Although this model is only approximate, it will allow designer to predict the effect of strip length and thickness. Joshi and Webb [1987] have developed an analytically based model to predict the $j$ and $f$ versus $Re_{Db}$ characteristics of the OSF array. This model properly accounts for all geometric factors of the array, heat transfer to the base surface area to which the fins are attached, and is able to account for the nonlaminar region. (The Eqs. are not shown here)

2.4 Transition from laminar to turbulent region

Transition from the laminar region occurs at the Reynolds number where the $j$ factor departs from log-linear form, as the Reynolds number is increased. This transition point is defined in Figure 5.8.

Figure 5.8 Illustration of transition of the j factor from the laminar region
Joshi and Webb [1987] determined this point from the $j$ vs. $Re_{Dh}$ data for 21 core geometries, and developed a semiempirical correlation to define the transition point. The Reynolds number ($GD_h/\mu$) at which the wake departs from laminar flow is defined as $Re_{Dh, tr}$ is given by

$$Re_{Dh, tr} = 257\left(\frac{L_p}{S}\right)^{1.23} \left(\frac{\delta}{L_p}\right)^{0.58} \left(\frac{D_h}{t_{mom}}\right)$$

(5.5)

Where $t_{mom} = \delta + 1.328L_p/Re_{L}^{0.5}$ is fin thickness, plus twice the momentum thickness of the boundary layer at the end of the strip length. The flow visualization measurements show that this transition point occurs before eddies are shed in the wake region.

2.5 Correlations for $j$ and $f$ vs. $Re$

Multiple regression, power law correlations have been developed by Wieting [1975] and by Joshi and Webb [1987] to predict the $j$ and $f$ vs. $Re_{Dh}$ characteristics for the OSF array.

Manglik and Bergles [1990] used an asymptotic correlation method developed by Churchill and Usagi [1972] to provide an alternate correlation for the OSF. This method requires that one know the asymptotic values for small and high Reynolds number values.

$$f = (f_{lam}^{n} + f_{turb}^{n})^{1/n} \text{ and } j = (j_{lam}^{n} + j_{turb}^{n})^{1/n}$$

(5.6)
2.6 Use of OSF with Liquids
The previously discussed correlations or models were developed based on air data (Pr=0.7). Hu and Herold [1995] obtained heat transfer and pressure drop data on seven OSF geometries using water and polyalphaolefin (聚α烯烃), for which Prandtl number ranged from 3 to 150. Comparison with the Wieting correlation [1975] and the Joshi and Webb analytical model [1987] revealed that the Hu and Harold data are highly overpredicted by the correlations. Hu and Harold [1995] extended the Joshi and Webb model by incorporating the Montgomery and Wilbulswas [1967] thermal developing solution to account for the Prandtl number effect.

Brazed aluminum automotive oil coolers frequently use oil channels containing the OSF to provide enhancement of the low-Reynolds-number oil. Muzychka and Yovanovivh [2001] developed a variant of the correlation applicable to oil coolers. They also took $j$ and $f$ vs. Re data for a 50/50 glycol/water mixture on 10 turbulator insert geometries, on which the correlation is based. Their data spanned $20<\text{Re}_{Dh}<200$. Data were taken at 2 fluid temperatures, which yielded $Pr=85$ and 150. It is interesting to note that their correlation shows $\text{Nu} \propto Pr^{1/3}$. 
2.7 Effect of percent fin offset

As conceived, the strips of the OSF fin are offset ½ fin spacing, or 50% offset. In this configuration, the wake length is equal to the strip width in the flow direction. However, this may not be the optimum displacement. It is typically assumed that the thermal and velocity layers are fully dissipated in the wake region. However, there is no rational basis for assuming that full dissipation will occur for a wake length equal to the strip width. There are some researchers investigated the effect of percent fin offset, typical results are shown here.

The Nu of row 1 equals that of row 2, and the Nu of row 3 equals that of row 4. Row 5 has a smaller Nu than that of row 3 or 4.

Row 3 offset 30% of the fin pitch. The Nu for row 5 is 10 to 20% higher than for row 5 of the Figure 5.9a array.

Figure 5.9 Local Nusselt numbers in OSF louver array studied by Kurosaki et al. [1998]
2.8 Effect of burried edges
There is a possibility that burrs may be formed on the upstream and downstream fin edges during the shearing operation to make the OSF geometry. This is especially true as the tooling used to shear the strips wears. The existence of such burrs was not established by the original experimenters for the 21 cores in the database. To investigate this possibility, Webb and Joshi [1983] used their asymptotic correlation for the 8 scaled-up, “burrr-free” geometries to predict the friction factor of the 21 actual core database. The friction factors of the actual cores were under predicted 10 to 20%. Hence, it appears that the some degree of burrs existed in the 21 actual cores.

3. Louver Fin
The louvered fin geometry of Figure 5.2f bears a close similarity to the OSF. Rather than offsetting the slit strips, the entire slit fin is rotated 20 to 45°, relative to the airflow direction. The louvered surface is the standard geometry for air-cooled automotive heat exchangers. Currently used louver fin geometries have a louver strip width (in the airflow direction) of 0.9 to 1.5 mm. For the same strip width, the louver fin geometry provides heat transfer coefficients comparable to the OSF.
3.1 Heat Transfer and Friction Correlations

The state-of-the-art correlation is Chang and Wang [1997] for heat transfer and Chang et al. [2000] for friction. They developed heat transfer and friction correlation based on 91 samples of louver fin heat exchanger. Their correlations are given below.

Only to plate-fin geometries having inline tube arrangements
3.2 Flow structure in the louver fin array

Although louvered surfaces have been in existence since the 1950s, it has only been within the past 20 years that serious attempts have been made to understand the flow phenomena and performance characteristics of the louvered fin.

Until the flow visualization studies of Davenport [1980], it was assumed that the flow is parallel to the louvers. At very low values of $Re_L$, Davenport observed that the main flow stream did not pass through the louvers. But, at high values of $Re_L$, the flow became nearly parallel to the louvers.

Figure 5.11 Heat transfer characteristics of several louvered plate fin geometries tested by Achiachia and Cowell [1988]
Since the heat transfer performance is closely related to the flow structure, we may infer that two types of flow structure exist within the louvered, plate fin array:

1. **Duct flow**, in which the fluid travels axially through the array, essentially bypassing the louvers.
2. **Boundary layer flow**, in which the fluid travels parallel to the louvers.

Kajino and Hiramatsu [1987] conducted a flow visualization study of the louvered fin array using dye injection and hydrogen bubble techniques. Figure 5.12 shows two louver arrays, which have the same louver pitch, but different fin pitches.

Figure 5.12 Steaklines at ReL=500 in 160 mm deep flow visualization model of louver fin array (l=10 mm, \( \theta = 26^\circ \)) tested by Kajino and Hiramatsu [1987]
In the left-hand portion of Figure 5.12a, a significant fraction of the flow bypasses the louvers. This is because the hydraulic resistance of the duct flow region is substantially smaller than that for boundary layer flow across the louvers. When the fin pitch is reduced, as in the right-hand part of Figure 5.12b, the hydraulic resistance of the duct is increased, so that most of the flow passes through the louvers.

Webb and Trauger [1991] defined the term flow efficiency (\(\eta\)) to quantify their observation. Figure 5.13 provides a visual definition of the flow efficiency. The flow efficiency (\(\eta\)) is defined using Figure 5.13 as

\[
\eta = \frac{N}{D} = \frac{\text{Actual transverse distance}}{\text{Ideal transverse distance}}
\]
The flow efficiency increases with increasing Reₜₐₜₜ Ł

Figure 5.14 Measured flow efficiency vs. Reynolds number for 20° louver angle as reported by Webb and Trauger [1991]

The figure shows that correlations based on numerical simulation (Achaichia and Cowell, and Zhang and Tafti) predict higher flow efficiencies compared with those from the experimentally based correlations. The trends are, however, quite different.

Figure 5.15 The flow efficiency predicted by various correlations. Predictions were made at pᵣ=1.0 and 1.5 mm, Lᵣ=1.0 mm, t=0.1 mm, and θ=30°
3.3 Analytical model for heat transfer and friction

Sahnoun and Webb [1992] developed an analytical model to predict the $j$ and $f$ factor vs. $Re_L$ for the louver fin array. The development is similar to that of the previously discussed Joshi and Webb [1987] model for the OSF array. However, flow in the louver fin is more complex for several reasons.

Dillen and Webb [1994] modified the Sahnoun and Webb [1992] model by empirically accounting for the bypass effect. In the Shanoun and Webb model, the bypass effect was modeled using the flow efficiency equation.
3.4 PEC Example 5.2

Automotive radiators typically use the louver fin geometry, rather than the OSF geometry. Compare the performance of the OSF with the louver fin. Assume the geometric design parameters shown in Table 5.1.

Table 5.1 Geometric Design Parameters for PEC Example 5.2

<table>
<thead>
<tr>
<th>Item</th>
<th>OSF</th>
<th>Louver</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fins/m</td>
<td>472</td>
<td>Same</td>
</tr>
<tr>
<td>Fin thickness ((\delta)), mm</td>
<td>0.10</td>
<td>Same</td>
</tr>
<tr>
<td>Height of fin array ((H)), mm</td>
<td>7.62</td>
<td>Same</td>
</tr>
<tr>
<td>Width of louver ((L_p)), mm</td>
<td>2.03</td>
<td>Same</td>
</tr>
<tr>
<td>Louver angle (for louver fin only)</td>
<td>No</td>
<td>26°</td>
</tr>
<tr>
<td>Ratio of louver length to fin height</td>
<td>1.0</td>
<td>0.8</td>
</tr>
<tr>
<td>Hydraulic diameter ((D_h)), mm</td>
<td>3.18</td>
<td>Same</td>
</tr>
<tr>
<td>Contraction ratio ((\sigma))</td>
<td>0.940</td>
<td>Same</td>
</tr>
</tbody>
</table>

The radiator is designed to operate at 27°C with 8.94 m/s air frontal velocity. Compare the \(j\) and \(f\) factors for the two fin designs at the same air velocity. This corresponds to the simple PEC Case FN-2 of Table 3.1.

The \(j\) and \(f\) factors are calculated using correlations of Davenport for the louver fin, and Joshi and Webb for the OSF geometries. Use of the correlations gives for the OSF \((j=0.0124, f=0.0574)\) and for the louver fin \((j=0.0123, f=0.045)\).
Using PEC Case FN-2, $A_{osf}/A_{louv} = j_{louv}/j_{osf} = 0.0132/0.0124 = 1.065$. The friction power ratio is proportional to the product, $fA$ for $G=constant$. Hence, $P_{osf}/P_{louv} = (f_{osf}/f_{louv})/(A_{osf}/A_{louv}) = (0.0574/0.045)/(1.065) = 1.35$.

This example shows that the louver fin has marginally higher $j$ factor (6.5%, which provides a 6.5% surface area reduction), assuming negligible tube-side thermal resistance. However, the 28% higher friction factor of the OSF, combined with its 6.5% lower $j$ factor, results in 35% higher friction power (and pressure drop).

### 4. Convex Louver Fin

<table>
<thead>
<tr>
<th>No.</th>
<th>$\theta$ (deg)</th>
<th>% Offset</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>2</td>
<td>12.8</td>
<td>23</td>
</tr>
<tr>
<td>3</td>
<td>17.4</td>
<td>33</td>
</tr>
<tr>
<td>4</td>
<td>24.6</td>
<td>53</td>
</tr>
<tr>
<td>5</td>
<td>9.7</td>
<td>20</td>
</tr>
<tr>
<td>6</td>
<td>17.4</td>
<td>33</td>
</tr>
<tr>
<td>7</td>
<td>20.7</td>
<td>42</td>
</tr>
<tr>
<td>8</td>
<td>24.6</td>
<td>53</td>
</tr>
</tbody>
</table>
Figure 5.18a shows the effect of the angle $\theta$ on the $j$ and $f$ factors for the OSF geometry. This figure shows that the 50% offset ($\theta=24.6^\circ$) does not provide the highest $j$ factor. At $Re_{Dh}=500$, the $\theta=17.4^\circ$ (33% offset) provides a 22% higher $j$ factor than the $\theta=24.6^\circ$ (50% offset) array. This agrees well with the findings of Kurosaki et al. [1988].

For the same angle $\theta$, Figure 5.18b shows that the OCLF provides higher $j$ factor than the OSF geometry. For $\theta=24.6^\circ$, the $j$ factor of the OCLF array is 48% higher than for the OSF array at $Re_{Dh}=500$. However, the friction factor increase is approximately 70%. 
Figure 5.19 is a plot of $j$ and $f$ vs. $\theta$ for the OSF and OCLF geometries for $Re_{Dh}=500$. It shows that the maximum values of $j$ and $f$ occur at for the OSF geometry. The $j$ factor for the OCLF array increases with, at least up to the highest $\theta$ tested (24.6°). Figure 5.19 shows that the higher $j$ factor of the OCLF is accompanied by a higher friction factor than in the OSF array.

Hatada and Senshu also performed flow visualization experiments in scaled-up OSF and OCLF arrays, without tubes present.

Examination of Figure 5.20c suggests that flow separation on the convex louvers causes significant profile drag for the $\theta=24.6^\circ$ louver angle.
Hitachi [1984] uses the convex louver fin geometry in its commercial plate fin-and-tube heat exchangers. Figure 5.21 is taken from its product brochure.

![Figure 5.21 The OCLF geometry in the Hitachi fin-and-tube heat exchanger. (Courtesy of Hitachi Cable.)](image)

5. Wavy Fin

The term wavy or corrugated is used to describe the Figure 5.2c geometry.

![Wavy](image)

**Figure 5.2**

For a corrugated geometry having constant corrugation angles and sharp wave tips, the key parameters that affect the performance are the wave pitch ($p_w$), the corrugation angle ($\theta$), and channel spacing ($s$). Whether the wave geometry has smooth or sharp corners will affect the performance.
At $Re_{th}=2000$, the Nusselt number enhancement ratio is 2.3 and 3.2 for the narrow and wide channels, respectively. The corresponding friction increases are 2.3 and 3.8, respectively.

**Figure** 5.23 Wavy channel data of All and Ramadhyani [1992].
(a) Friction factor, (b) Nusselt number for $Pr=7$. Curve 1, $\theta=30^\circ$, $b/p_w=0.29$, Curve 2, $\theta=20^\circ$, $b/p_w=0.23$, Curve 3, $\theta=20^\circ$, $b/p_w=0.15$, Curve 4, parallel plate.

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6. Three-Dimensional Corrugated Fins

Three-dimensional corrugated channels are used in plate-type heat exchangers and rotary regenerators. Figure shows a cutaway view of a corrugated channel.

Corrugated Channels
Both $j$ and $f$ increase monotonically up to $\theta = 80^\circ$. Beyond $80^\circ$, they decrease slightly, and approach the values of a two-dimensional corrugated channel. The increase is much more pronounced for the friction factors than for the $j$ factors. For $\theta$ increasing from $30^\circ$ to $80^\circ$, the friction factors increases 20 to 30 times, while the $j$ factors increase only 2 to 3 times.

7. Perforated (穿孔的) Fins

This surface geometry (Figure 5.2e) is made by forming a pattern of spaced holes in the fin material before it is folded to form the U-shaped flow channels.

If the porosity of the resulting surface is sufficiently high, enhancement can occur due to boundary layer dissipation in the wake region formed by the holes.
Shah [1975] provides a detailed evaluation of the perforated fin based on his study of test data on 68 perforated fin geometries. Shah concludes that little enhancement occurs for $Re_{Dh} < 2000$, if the heat transfer coefficient is based on the plate area before the holes were punched. Moderate enhancement may occur in the transition and turbulent flow regimes, $Re_{Dh} > 2000$, depending on the hole size and the plate porosity.

Shen et al. [1987] tested flat plate-and-fin channels having round holes or rectangular cutouts. They found no benefit in the laminar range, but holes promote earlier transition and they observed moderate increase in the turbulent range. Substantially higher performance was obtained with the rectangular slots.

Figure 5.25a shows an interesting variant of the perforated fin, which was tested by Fujii et al. [1988]. This geometry has 2.0-mm-diameter holes in corrugated plates. The plates are aligned, such that the channels have expanding and contracting flow areas.

As shown in Figure 5.25b, secondary flow through the perforations is speculated to be an important contributor to the enhancement level.
8. Pin Fins (针翅) and Wire Mesh

A pin fin surface geometry may be made of a special weave of “screen wire.” Forming this screening into the shape of U-shaped channels results in the pin fin surface geometry. The wire may have a round, elliptical, or square cross-section shape.

Although such pin fin geometries may have high performance, they are not widely used, because the cost of such surfaces is significantly higher than the cost of the thin sheet used to make geometries such as the OSF and louver fins.
Figure 5.27 Expanded metal (copper) matrix geometry tested by Torikoshi and Kawabata [1989]
9. Vortex Generators
Streamwise vortices will be shed from geometric shapes attached to the wall, which attack the flow at an angle. Figure 5.28 from Fiebig et al. [1993] illustrates several types of vortex generators that have been investigated.

![Figure 5.28 Various types of vortex generators. (a) Wing-type vortex generators: from left, delta wing, rectangular wing, delta winglet, and rectangular winglet pair; (b) illustration of vortices and vortex renewal. (From Fiebig et al. [1993].)]](image)

A system of vortices forms on the protrusion, bends around it, and is carried downstream in a longitudinal vortex pattern. Longitudinal vortices were found by Eibeck and Eaton [1987] to persist for more than 100 protrusion heights downstream.

To be a practical vortex generator, the protrusion height should be comparable to the local boundary layer thickness. If it is higher, it will cause a significantly higher pressure drop increase than the heat transfer enhancement.
As illustrated in Figure 5.28b, vortex generators may be arranged in pairs forming and in an alternate manner. The interaction of adjacent vortices can be divided into two-types: "common flow down" and "common flow up," as described by Mehta et al. [1983].

![Figure 6.16](image)

Figure 6.16 Individual fins having a pair of winglet vortex generators: (a) common-flow-down, (b) common-flow-up configuration. (From O’Brien et al. [2003].)

When the direction of the secondary flow between two counter-rotating vortices is toward the wall, the vortices are called common flow-down, and when the direction is away from the wall, they are called common flow-up. The pairing of vortex generators produces common flow-down vortices and pairing produces common flow-up vortices.

Figure 5.29 shows a sketch of longitudinal vortices behind a vortex generator placed in a laminar boundary layer on a flat plate (Torii et al. [1994]).
The main vortex is formed by the flow separation on the leading edge of the wing, while the corner vortex is formed by deformation of near-wall vortex lines at the pressure-side of the wing. Sometimes, an induced vortex rotating opposite to the main and corner vortices are observed.

Figure 5.29 A sketch of longitudinal vortices generated from a winglet vortex generator.

Figure 5.30 Streamwise velocity contours downstream of two delta winglet vortex generators with (a) common-flow-down and (b) common-flow-up configuration. Two delta winglet pairs have 18° angle of attack and 40-mm spacing. (From Pauley and Eaton [1988]).
10. Metal Foam Fin
Metallic foam has high surface-area-to-volume ratio, and the flow mixing is enhanced due to the tortuous passage. The pressure drop, however, is larger compared with other enhanced fin geometries, because of the typically small hydraulic diameter.
Recently, metal foams find application to electronic heat sinks. Bhattacharya and Mahajan [2000] applied metal foams to finned channel heat sinks. Klett et al. [2000] developed a high-thermal-conductivity carbon foam material. Carbon foam has much higher thermal conductivity than aluminum foam. Hence, it will operate at considerably higher fin efficiency. The carbon foam heat exchanger had a heat transfer coefficient of 280 to 500 W/m²⋅K for 1.5 to 4.5 m/s frontal air velocity. This value is approximately three times that of a louver fin geometry.
The carbon foam had very large heat transfer surface area per unit volume \((\beta = 1.88 \times 10^6 \text{ m}^2/\text{m}^3)\), which compares to \(\beta = 2 \times 10^3 \text{ m}^2\text{m}^3\) for a louver fin geometry with 25 fins/in. However, this carbon foam had a pore size of 60 to 325 \(\mu\text{m}\), which results in a very small hydraulic diameter. The principal problem with this carbon foam is that the pore size is too small to meet practical design requirements. It may be possible to develop a foam structure with a larger pore size that would be more practical.

Figure 5.32 Comparison of fin efficiency of three mesh materials.
11. Plain Fins
If plain fins are used (e.g., Figure 5.2a and b), the flow channel will have a rectangular or triangular cross section.

If the flow is turbulent, standard equations for turbulent flow in circular tubes may be used to calculate $j$ and $f$, provided $Re$ is based on the hydraulic diameter ($D_h$). If $Re_{Dh}<2000$, one may use theoretical laminar flow solutions for $j$ and $f$. Values of $j$ and $f$ for developing and for fully developed laminar flow are given in Section 2.4 for a variety of duct shapes.

PEC Example 5.3
This example is presented using the FN-3 criterion. Two plain fin geometries with the same fin pitch are compared: a triangular geometry ($T$) and a rectangular geometry ($R$). Figure 5.33 shows the flow cross section of the two geometries.

Figure 5.36 Rectangular and triangular fin geometries used for PEC Example 5.3.
The FN-3 criterion constrains the two geometries to operate at the same mass flow rate with equal frontal velocity and $hA$ values. Because the velocities are known, the $j$ and $f$ values are directly calculable. The calculations are made for airflow at 27°C and 4 m/s frontal velocity. The $j$ and $f$ factors are given by the fully developed laminar solution for constant wall temperature and are taken from Table 2.2. By using the calculated $h$ and $f$ values,

$$\frac{A_T}{A_R} = \frac{h_R}{h_T} = \frac{39.80}{18.53} = 2.148$$

$$\frac{V_T}{V_R} = \frac{A_T}{A_R} \frac{\beta_R}{\beta_T} = 2.148 \times \frac{1225}{1143} = 2.30$$

$$\frac{P_T}{P_R} = \frac{f_T}{f_R} \frac{A_T}{A_R} \left( \frac{D_{hT} \text{ Re}_T}{D_{hR} \text{ Re}_R} \right)^3 = \frac{0.0167}{0.024} \times 2.148 \times \left( \frac{2.94 \times 781}{2.96 \times 776} \right)^3 = 1.49$$

For the same frontal area, flow rate, and heat transfer rate, this Case FN-3 example shows that the triangular geometry requires 115% more surface area and 49% greater pumping power.
12. Conclusions

(1) The Plate-and-fin heat exchanger geometry has become an increasingly important design. The high-performance offset strip and louver fins provide quite high heat transfer coefficients for gases and two-phase applications. It offers significant advantages over the traditional fin-and-round-tube geometry. Key advantages are lower gas pressure drop than circular tube designs, and the ability to have the fins normal to the gas flow over the full gas flow depth. Vortex generators or metallic foams are emerging as new enhancement geometries. Significant advances have been made on numerical analysis of the complex plate-fin geometries.

(2) Early variants were applied to gas to gas applications. It is now used for gases, liquids, or two-phase fluids on either side. Designs using extruded aluminum tubes with internal membranes allow quite high tube-side design pressure, e.g., 150 atm. Further innovative designs, applications, and advanced fin geometries are expected. Currently, aluminum, steel, and even ceramics are used.
Thank you for your attention!

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