SELECTION CRITERIA FOR PLAIN AND SEGMENTED FINNED TUBES FOR HEAT RECOVERY SYSTEMS

Don R. Reid  
FINTUBE Corp.,  
Tulsa, Oklahoma.

Jerry Taborek  
Consultant  
Virginia Beach, Virginia

ABSTRACT

Heat recovery heat exchangers with gas as one of the streams depend on the use of finned tubes to compensate for the inherently low gas heat transfer coefficient. Standard frequency welded "plain" fins were generally used in the past, until the high frequency resistance welding technology permitted a cost-effective manufacture of "segmented" fins. The main advantage of this fin design (Fig.1) is that it permits higher heat flux and hence smaller, lighter weight units for most operating conditions. While the criteria which dictate optimum design, such as compactness, weight and cost per unit area favor the segmented fin design, a few other considerations such as fouling, ease of cleaning and availability of dependable design methods have to be considered. This paper analyzes the performance parameters which affect the selection of either fin type.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Units</th>
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</thead>
<tbody>
<tr>
<td>$A_f$</td>
<td>heat transfer area of fins</td>
<td>m², ft²</td>
</tr>
<tr>
<td>$A_0$</td>
<td>heat transfer area total outside</td>
<td>m², ft²</td>
</tr>
<tr>
<td>$H$</td>
<td>height, m, ft</td>
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<tr>
<td>$h$</td>
<td>heat transfer coeff., W/m²·K, Btu/hr·ft²·F</td>
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<td>$k$</td>
<td>thermal conductivity, W/m·K, Btu/hr·ft°F</td>
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</tr>
<tr>
<td>$N_f$</td>
<td>no. of fins/m or fins/inch</td>
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</tr>
<tr>
<td>$Re$</td>
<td>Reynolds no., based on minimum flow area</td>
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</tr>
<tr>
<td>$R_{ft}$</td>
<td>fouling resist. inside, m²·K/W, hr·ft²·F/Btu</td>
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<tr>
<td>$S_f$</td>
<td>fin clearance, (1/$N_f$ - 0.5), m, ft</td>
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</tr>
<tr>
<td>$t_f$</td>
<td>fin thickness, m, ft</td>
<td></td>
</tr>
<tr>
<td>$w$</td>
<td>width of fin segment, m, ft</td>
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</tr>
<tr>
<td>$U$</td>
<td>overall ht. trans. coeff., W/m²·K, Btu/hr·ft²·F</td>
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Greek Symbols

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<thead>
<tr>
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<tbody>
<tr>
<td>$\alpha$</td>
<td>fin efficiency</td>
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<tr>
<td>$\psi$</td>
<td>gas velocity gradient correction factor</td>
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Subscripts

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<tr>
<td>av</td>
<td>average</td>
</tr>
<tr>
<td>e</td>
<td>effective</td>
</tr>
<tr>
<td>f</td>
<td>fin-side</td>
</tr>
<tr>
<td>g</td>
<td>referred to gas-side</td>
</tr>
<tr>
<td>i</td>
<td>referred to inside tube</td>
</tr>
<tr>
<td>o</td>
<td>referred to outside tube</td>
</tr>
<tr>
<td>p, pl</td>
<td>referred to plain tube surface</td>
</tr>
<tr>
<td>w</td>
<td>referred to tube wall</td>
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HISTORICAL BACKGROUND AND OVERVIEW

Whenever the stream to be used on the outside of tube banks is a gas with inherently low heat transfer coefficient, the use of external high finned tubes is indicated. To obtain the resulting large surfaces, it is more economical to "extend" the plain tube surface with fins than to increase the number of plain tubes. While this was recognized in the 1900's, it was not until the 1920's that manufacturing techniques...
permitted production of light duty aluminum helically
wound radial fins for air cooled heat exchangers. For
high temperatures, welded high longitudinal fins and
studs were also introduced at that time. Later, im-
proved resistance welding techniques permitted at-
tachment of helically wound heavy duty radial fins,
with area enlargement factors of about 7 to 15 com-
pared to bare tube.

Until the 1960's, fins with an "L" - shaped foot were
used, as the "standard" low frequency (50 or 60 Hz)
electric current required substantial area to form a
solid bond with the base tube. The height of a plain
fin that can be formed from a metal strip around the
bare tube is limited by the drawing process and thus,
in the early stages of the technology, the "cut" or "se-
gmented" fins were introduced. The fin material is cut
close to the base in intervals of typically 4 to 8 mm
(5/32 to 5/16 in.), depending on tube diameter, so
that the fin perimeter can expand freely. Typical ex-
amples of the two types, plain and segmented fins,
are shown in Fig.1.

In the 1970's the modern high frequency (450 kHz)
resistance welding technique was perfected and is
now exclusively used (with exception of small tube
diameters and some special cases). Because the high
frequency weld forms a strong metallurgical bond, the
"L" foot is no longer required and the fins are formed
from a simple flat metal strip, sometime referred to as
"I" shape. Compared to the low frequency welding
technique, much higher welding speeds are possible
and less material is used than for the "L" foot type.
Both, the plain and segmented fins are manufactured
by the high frequency technique, but the "segmented"
fins have several important advantages:

a. easier to manufacture, as no drawing of the
material is required;
b. the heat transfer coefficient is higher because of
increased turbulence;
c. higher fins are possible because of a), and fin
efficiency is also improved;
d. as a consequence of all above, the heat ex-
changer is lighter and cost is reduced.

During the last two decades the modern "se-
gmented" fin design has been rapidly replacing the
plain fins, in all suitable applications. For understan-
dable reasons, the power and heat recovery industry
is very conservative in the selection of dependable
components; however, segmented tubes are fast
gaining acceptance even for the most severe duties.
This paper clarifies some of the questions commonly
asked regarding selection of the fin type for optimum
performance application and cost-effective design.

For finned tubes application, it is convenient to write
the equation for the overall heat transfer coefficient as
U*, based on a "reference area" A* :
The term \( \Omega \) represents the fin-side heat transfer coefficient corrected by a weighted "effective" fin efficiency \( \Omega_e \); \( A_o \) is the total outside tube area (fins and exposed root); \( R_{fo} \) and \( R_{fi} \) are the outside and inside tube fouling resistances; \( A_{w,av} \) is the area at average wall thickness and \( R_{wall} \) is wall resistance.

Here the advantage of writing the heat transfer equation in this form is that the designer/user can select the most convenient reference area for his application. Three logical selections are possible:

1. \( A^* = A_o \), that is the reference area is the total outside area. This is often the usual way of presentation, but the inside tube resistances will appear exaggerated because the area multiplier \( A_o \) is between 7 to 15. The U* - value appears low, but the area \( A^* \) is large.
2. \( A^* = A_{pl} \), that is referring the U coefficient to the outside plain tube surface; the fin tube coefficient will appear high, directly representing the advantage compared to plain tubes.
3. \( A^* = A_i \), refers the U coefficient to the inside tube surface. The advantage of this presentation is for cases where \( h_i \) is important for comparison.

Analysis of Eq.1 teaches us several important rules of finned tube application:

1. Justification for use of finned tubes occurs when the outside tube resistance is substantially larger than the total inside tube resistance, usually 3 : 1 or higher, to compensate for the cost of fins.
2. For most effective use of externally finned tubes, the inside tube resistance must be low, i.e. high heat transfer coefficient and low fouling resistance. High inside tube resistance may render the economics of external fins ineffective. Designers should be especially careful in selection of inside tube fouling resistances, which are often exaggerated. For example, the inside tube resistance based on a coefficient of 5700 W/m²·K (1000 Btu/hr·ft²·F), will increased by 100 percent using \( R_{fi} = 0.00018 \) m²·K/W, [0.001 (Btu/hr·ft²·F)] by an area enlargement factor of 15.
3. In contrast to item 2 above, the fouling resistance on the finned surface has often only a small effect on U and hence the size of the unit. This is because the low gas coefficient will be only marginally affected by even a large fouling resistance. For example, a typical gas coefficient of 33 (W/m²·K) or 6 (Btu/hr·ft²·F), combined with a rather large fouling resistance of 0.0035 (K·m²/W) or 0.02 (Btu/h·ft²·F), would result in a decrease of U by only 11 percent. While this correctly represents the thermal equations, the selection of high fouling fin-side resistances creates often a false sense of safety factor. Proper design analysis should compare performance under clean and fouled conditions and establish the true safety factor.

Fin Efficiency

The heat transfer from a fin is subject to a penalty called fin efficiency, \( \Omega \), which accounts for the temperature gradient between fin root and fin tip due to heat conduction within the fin. Only such principles of fin efficiency are presented here which illustrate the differences between plain and segmented fins. For detailed treatment see Kern and Kraus (1972). Fin efficiency is defined from analysis in terms of a group "m" and the fin height \( H_f \):

\[
\Omega = \frac{\tanh (m_e H_{fe})}{m_e H_{fe}} \quad m_e = \sqrt{\frac{2 \left( h_f + 1/R_{fo} \right)}{k_f t_{fe}}}
\]

where \( H_{fe} \) is "effective" fin height and \( t_{fe} \) is "effective" fin thickness. The term "effective" indicates that the actual value is modified from analysis for any specific fin geometry, such as plain radial fins, segmented fins or stud fins. Thus radial plain fins are penalized because of the much larger fin-tip circumference compared to the root, as analyzed by Schmidt (1966) and expressed as \( H_{fe} \):

\[
H_{fe} = H_f \left[ \left( 1 + \frac{t}{2 H_f} \right) \left( 1 + 0.35 \ln \frac{D_f}{D_o} \right) \right]
\]

where the term in [ ] increases \( H_f \) and hence decreases \( \Omega \).
The rectangular cross-section of the segmented fins is calculated from analysis of round stud fins with a modified expression for fin thickness $t_{fe}$:

$$t_{fe} = \frac{(t \cdot w)}{(t + w)}$$

(4)

The factor $\psi$ is a correction for non-uniform heat transfer coefficient in plain radial fins, accounting for a flow velocity gradient between fin tip and fin root, as analyzed by Lymer and Ridal (1961). Applicable to fin densities of 4.2 mm/fin or 6 fins/inch and higher and fin heights of 20 mm or 0.75 in. and higher, the following correction is suggested:

$$\psi = 0.7 + 0.3 \Omega$$

(5)

The segmented fin structure permits gas penetration to the fin root, and therefore $\psi = 1$.

Fin efficiency is usually presented in graphical form as $\Omega$ vs. $(m_e \cdot H_{fe})$, as is shown in Fig.2. There the line with $(D_t/D_o) = 1$ applies to segmented fins of all dimensions, while the parameter $(D_t/D_o) > 1$ is used for plain fins. Finally, the finned tube heat transfer coefficient must be multiplied by a "weighted" or "effective" fin efficiency term $\Omega_e$, which penalizes only the finned part of the surface. Thus $\Omega_e$ is always larger than $\Omega$, and the following equation applies:

$$\Omega_e = 1 - (1 - \Omega) \left( \frac{A_f}{A_o} \right)$$

(6)

**Heat transfer coefficient for high finned tubes**

Only limited data of uncertain quality were available for high fins until the extensive work at the University of Michigan in the 1950's. Systematic sets of data for welded plain and segmented fins, were obtained in the 1960's at a large scale wind tunnel at Heat Transfer Research Inc. (HTRI). Selected literature on the subject includes work by Rabas and Eckels (1975), Rosenman et al., (1976) and Weierman et al., (1974). Based on these and other available data, Weierman (1976, 1979) published a comprehensive set of design equations for wide ranges of tube and fin geometry combinations, which is widely quoted in the literature. While it is not the objective of this paper to dwell on thermo-hydraulics design details, the basic principles of the thermal correlations must be included here as background for the fin selection criteria.

The basis for the heat transfer coefficient of finned tubes is a correlation for crossflow over bare tubes. However, the surface area extension by plain fins is never as effective as the prime tube surface (not considering fin efficiency itself). This is mainly because of incomplete penetration of the gas between the fins. Segmented fins have also a decreased effectiveness, but to much lesser degree, because they promote gas penetration between the fins and increase turbulence of the gas stream between the fins.

The Weierman heat transfer equation is presented here in a somewhat modified form, restricted to equilateral triangular tube layout and tube rows > 4. The base equation for the Nusselt number for plain tubes

$$Nu_p = \frac{0.25 \cdot Re^{0.65} \cdot Pr^{1/3} \cdot \left( \frac{T_b}{T_{wall}} \right)^{0.25}}{k_o}$$

(7)

is modified for use with welded fins by two correction factors $F_{Df}$ and $F_{fd}$, which express the effects of fin height and fin density, as described in principle below.

$$Nu_t = \frac{h_t \cdot D_o}{k_o} = Nu_p \cdot \left[ F_{Df} \cdot F_{fd} \right]$$

(8)

1. The correction factor $F_{fd}$ penalizes the Nu-number for the incomplete gas penetration into the fins. This is shown in Fig.3 as function of the fin density parameter $H_t/S_t$, where $S_t = (1/N_t - t_t)$, the
clear space between fins. Logically, it has a different values for staggered and in-line tube layout. For low fin heights and low fin densities, i.e. for \((H_f/S_f) < 2\), there is no appreciable difference between segmented and plain fins. However, for higher values of \((H_f/S_f)\), the penalty is much more severe for plain than for segmented fins. This is because the fin segmentation allows better penetration of the gas between the fins. This, in turn, permits the use of greater fin height for segmented fins without excessive penalty. For typical fin densities, the correction factor for segmented fins will be between 10 to 25% higher than for plain fins.

2. The correction factor \(F_{DF} = (D_f/D_o)^{0.5}\) is always larger than 1.0 and represents the heat transfer enhancing effects of the fins. It is shown in Fig. 4 as \((D_f/D_o)^{0.5}\) plotted against the base tube diameter \(D_o\), with fin height \(H_f\) as the main parameter. Restrictive limits are drawn in for plain fins \((F_{DF} = 1.3)\) and for segmented fins, with an additional parameter of \(H_f/S_f\) for segmented fins. The limits roughly coincide with the practical manufacturing restrictions.

Space does not permit analysis and comparison of pressure drop between plain and segmented fins. In general, the same reasons which create superior heat transfer performance of segmented finned tubes will also result in increased pressure drop. However, the generally smaller size of segmented fin exchangers (lower number of tube rows) will largely compensate for the increased friction, as confirmed from numerous designs.

**SELECTION OF FIN DESIGN ELEMENTS**

The design of any finned tube heat exchanger starts with the selection of the fin and tube layout dimensions. Here are some basic guidelines:

- **Fin density (spacing).** To produce the maximum outside area per unit of tube length, the highest permissible fin density is used, limited only by:
  
a. Too high pressure drop, which is a strong function of fin density;
b. The penalty for incomplete gas penetration, especially with high fins, as shown in Fig. 3.
c. The potential for increased fouling. Recommended fin densities are shown in Table 1.
Table 1. Recommended fin densities

<table>
<thead>
<tr>
<th>Fin-side fluid type</th>
<th>Fins/in</th>
<th>mm/fin</th>
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<tbody>
<tr>
<td>Clean air</td>
<td>8 - 11</td>
<td>3.0 - 2.3</td>
</tr>
<tr>
<td>Nat. gas combustion</td>
<td>6 - 8</td>
<td>4.2 - 3.0</td>
</tr>
<tr>
<td>Light oils combustion</td>
<td>4 - 6</td>
<td>6.3 - 4.2</td>
</tr>
<tr>
<td>Heavy oils combustion</td>
<td>3 - 5</td>
<td>8.5 - 5.0</td>
</tr>
<tr>
<td>Solid fuel combustion</td>
<td>1 - 2</td>
<td>25 - 12</td>
</tr>
</tbody>
</table>

**Fin Height.** Increasing the fin height increases the outside surface, but this creates complex interactions, requiring a compromise:

a. Fin height dictates the tube pitch and thus the basic parameter of mass flow rate, which affects heat transfer and pressure drop;
b. Manufacturing limitations, more strongly so for plain than for segmented fins;
c. Fin efficiency decreases, again more severely for plain than for segmented fins;
d. For fin height less than 12 mm or 0.5 inch, plain fins are preferred.

For example, if all variables are kept the same and only fin height is increased, the exchanger cost first decreases, then flattens out and finally starts to increase again, as gains in area are offset by effects of large tube pitch, mainly lower gas flow velocity, lower fin efficiency and lower effectiveness of flow penetration.

**Fin thickness.** Low fin thickness permits the highest fin density, but has also the lowest fin efficiency and low structural rigidity. The minimum fin thickness is usually 0.036 in. or 0.9 mm. Demand for rugged construction, handling of corrosive/abrasive fluids or high temperature fluids, dictates the use of thicker fins, which are available up to 0.165 in. or 4.2 mm, subject to some restrictions for smaller diameters. The most popular thickness is 1.2 mm.

**Tube pitch and layout type.** A staggered, equilateral triangular layout has the highest area density per volume and is regularly used. The tube pitch is usually selected so that about 1/4 of bare tube OD clearance would exist between fin tips, which results in standard dimensions for each tube size. While smaller tube pitch is hardly practical, larger tube pitch will decrease pressure drop, improve "pressure drop to heat transfer conversion", but hurt compactness.

In-line 90 degree layout produces very low pressure drop but also lower heat transfer, as the adjoining tubes are in the "shadow" of the upstream tube rows. Furthermore, stream bypassing in the lane between the tubes distorts the temperature profile and is hard to predict, until for more than about 10 tube rows lateral mixing starts being effective, as discussed by Weierman et al. (1974). For these reasons, in-line layouts will be used only when specific advantages, usually connected with cleaning procedures such as soot blowing, will become significant.

**Comparison of fin area.** The fin segmentation appears to decrease the heat transferring area (Fig.1), but if we exclude fin heights of 0.5 in. and lower, and compensate for the sector-cut by adding the area of the fin thickness periphery, the decrease is usually only a few percent, more than compensated for by the higher heat transfer coefficient. Usually segmented fin type is selected simply because the desired fin height cannot be manufactured in plain type.

**Gas-side Fouling.** The many types of fouling encountered in heat recovery system are still not quite understood despite considerable progress in this area, as documented in a recent comprehensive review by Marner (1990). The most common fouling forms are fly-ash, chemical reaction products, usually as gluey deposits and hard crust connected with high temperature. Of interest for the fin type selection are the following general observations.

Plain fins with smooth surfaces may have some advantages in particle type fouling as well as for effectiveness and ease of cleaning. On the other hand segmented fins create turbulence and permit more uniform gas velocity over the fin height. On hard crust fouling deposits, the edges of segmented fins may create faults and promote fouling removal, as was observed on low fin tubes.

A major problem appears to be too close fin spacing, which may cause "bridging" of fouling deposits and thus substantial deterioration of performance. In the absence of data-supported information, the recommendations be Weierman (1974) and Marner and Suitor (1987) will remain the best guidelines.
**Design strategy.** The selection of the combination of the fin tube design elements must reconcile often contradictory demands. Practical solutions lean on experiences from similar cases, if such exist, but each case is usually sufficiently different to require detailed attention. The user should rely on a computer based design analysis, which utilizes the "case-study optimization" technique: selected design elements are varied within limits compatible with manufacturing and operational practices, until a solution for the least expensive unit is obtained. Help from reliable manufacturers is often available.

**SELECTION CRITERIA FOR PLAIN AND SEGMENTED FINS**

The most important areas of welded finned tube applications and their specific design aspects are briefly summarized as follows:

1. Heat Recovery Steam Generators (HRSG) use exhaust gas from a turbine to generate steam for a variety of commercial uses, also called cogeneration. This is a rapidly growing application because the overall cycle efficiency is increased. The clean hot gas is ideally suited for segmented fin applications.

2. Fired Heater Convection Sections utilize hot gases in natural draft created by a stack and hence velocity is low and so is permissible pressure drop. Large diameter tubes at rather low fin densities are used. In general this application is also well suited for segmented fins, but fouling considerations and demands for ruggedness must be observed.

3. Waste Heat Boilers are widely used in power, petro-chemical and refinery operations as steam generators, economizers and superheaters. Demands on the finned tubes are severe, requiring high area extension ratios and often stainless steel fins. In all cases segmented fins have the advantage of producing higher heat transfer coefficients and heat flux than plain fins, and hence more compact design.

**Manufacturing aspects and cost**

Both, plain and segmented finned tubes are manufactured by the same high frequency welding machines. The actual segmenting is done between the fin strip spooling and the welding station without slowing down the welding process. For plain fins, the segmenting station is simply bypassed, but the fin shaping process is more difficult than for segmented fins. Thus, segmented fins are equally commercially available as plain fins, if the finning machinery is configured accordingly. This is the usual case in the USA, where segmented fins have been generally accepted, but not so in all cases in Europe and Asia.

The ease of manufacturing segmented finning is reflected in price comparison with plain fins. This advantage increases for high and dense fins, which are much easier to manufacture in segmented fins. For otherwise identical fin parameters, the fins for a typical boiler tube for turbine exhaust gases would weigh about 80% and cost about 84% in segmented fins compared to plain fins (not including cost and weight of the bare tube). For fins thicker than 2.6 mm or 0.105 in. and/or for fin densities below 8 mm/fin or 3 fins/in., there is little price difference between the two fin types.

**Usage and selection practices**

Segmented fin popularity has steadily increased in the last two decades, due to demonstrated reliability and cost advantage. In HRSG applications, segmented fin utilization in the USA is approaching 90%, with most of the remaining 10% being in the superheater coils. The design of convection sections and various heat recovery equipment is also converting to segmented tubes, where properly applicable.

On the other hand, plain fins will remain the exclusive choice for solid fuel fired systems. There is some history and design inertia of using thicker plain fins in such applications, and this is largely responsible for their reputation for durability and life. Plain fins also continue to enjoy a reputation for easier coil fabrication and handling with less damage.

The greater structural strength of plain fins has advantages in bending and tube support bearing, although present designs seldom call for such strength. At least some of the above items are remainders of past practices, and segmented fins were shown to perform satisfactorily in most equipment design and fabrication methods, while providing the advantage of weight and price.
CONCLUSION

The criteria for selection of the segmented or plain fins can be summarized as follows.

- Segmented fin formation uses faster welding speeds and hence the manufacturing cost is lower.
- Plain fin formation depends on expanding radially the fin material, thus limiting the fin height which can be manufactured. Segmented fin formation requires only "spreading" the segments, thus permitting the manufacture of higher fins.
- The fin segmentation increases turbulence and improves gas penetration to the fin root area, thus equalizing the flow velocity over the fin height. This results in higher heat transfer coefficient than for plain fins by as much as 20 percent.
- Uniform flow velocity over the fin height removes reasons for velocity distribution penalty for fin efficiency, a substantial advantage for higher fins and low conductivity materials.
- Because of the higher coefficient, the design usually requires fewer tube rows, compensating for the slightly increased pressure drop for segmented fins.
- The segmented fin thickness remains constant over the height of the fin, which results in more rugged construction and resistance to high temperatures.
- In general, plain fins withstand more physical abuse. Fins lower than 12 mm or 0.5 inch, and/or fin density less than 8.5 mm/fm or 3 fins/in. are used exclusively in plain fin design, as the advantages of segmentation would not be cost-effective.

There are applications and exchanger manufacturing methods for which one or the other fin type is preferred. While in some cases fouling and cleaning consideration will favor plain fins, segmented fins are superior with respect to weight, compactness and price. Based on the established field performance record for the last two decades, segmented fins should be considered for all future designs.

REFERENCES


