

EXXON EXXON ENGINEERING	HEAT EXCHANGE EQUIPMENT CALCULATION PROCEDURE, LOW-FINNED TUBES		DESIGN PRACTICES	
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Revision Memo

12/99	Added new sections on Finned Tube Screening Process, Grassroot vs. Debottlenecking Applications, and Specifying Finned Tubes. Revised required modifications to calculations in Sections IX-D, IX-E, and IX-F. Updated NOMENCLATURE .
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SCOPE

This practice deals with the application of external low finned tube technology. It addresses specific issues, including; the screening process for choosing finned tubes, special considerations for both debottlenecking and grassroots applications, specifying finned tubes, and accounting for the thermal and hydraulic effects of finned tubes on heat exchanger operation.

Low finned tubes are especially economical when the shellside resistance to heat transfer is greater than three times the inside resistance. This is typically true for coolers, condensers, and reboilers, and is often the case with oil-to-oil exchangers.

Low finned tubes can be used to increase heat transfer in an existing shell, simply by retubing the bundle with low fins. When this route will give the desired heat transfer, it is always cheaper than providing an additional shell.

REFERENCES

DESIGN PRACTICES

- Section IX-A Heat Exchanger Equipment - Exchanger Types and Applications
- Section IX-C Heat Exchanger Equipment - Design Considerations for Shell and Tube Exchangers
- Section IX-D Heat Exchanger Equipment - Calculation Procedure, No Change of Phase
- Section IX-E Heat Exchanger Equipment - Calculation Procedure, Vaporization
- Section IX-F Heat Exchanger Equipment - Calculation Procedure, Condensation

COMPANY TECHNICAL REPORTS

1. *Field Applications of Double Enhanced Tubes in Shell-and-Tube and Air-Cooled Heat Exchangers*, ER&E Report No. EE.108E.94, December 1994.
2. *Heat Transfer Performance Enhancement Using HiTRAN Inserts*, ER&E Report No. 94.ESC4.010, 1994.
3. *Enhancing Heat Exchanger Performance with Turbulence Promoters*, ER&E Report No. EE.14E.88, January, 1988.
4. *Enhanced Heat Transfer Technologies for Improving Heat Exchanger Performance*, ER&E Report No. EE.49E.88, November, 1988.
5. *Survey of Tube-Side Heat Transfer Enhancement Techniques*, ER&E Report No. EE.20E.83, March, 1983.

OTHER REFERENCES

1. Heat Transfer Research Inc. (HTRI), *Design Manuals*
2. Heat Transfer Research Inc. (HTRI), *Computer Support Volume*
3. Third Exxon Heat Exchanger Forum (October 29 - 31, 1957); Topic 3D-2
4. W. O. Weber Article, Humble Oil & Ref. - Baytown, TX (Chem. Engr. Vol 67, No. 6, Page 149 (1960))

BACKGROUND

Externally finned tubes, also commonly known as Integral-Fin Tubes (IFT) or Low Fin Tubes, have extended external heat transfer surface area that is approximately 2.5 times greater than that of the same diameter plain surface tubes. The extended surface area is achieved by a cold-rolling process that produces short fins ranging in density from 11 to 40 fins/in. (430 to 1575 fins/meter) and in height from 0.030 to 0.060 in. (0.75 to 1.5 mm). **Figure 1** is a sketch of a typical low-finned tube.

The low-finned technology, developed in the late 1940's, has been successfully applied in both clean and fouling services. In the 1960's and 1970's, finned tubes started becoming available in a wide range of corrosion resistant materials. Whereas early finned tube production was driven by the manufacturing techniques available at that time, there have been significant manufacturing and performance improvement developments associated with fin shape and fin density (i.e., fin spacing). Unlike other heat transfer enhancement technologies that have specialized application in vaporizing or single-phase streams, finned tubes have been used very effectively in single-phase, condensing, and vaporizing services.

SCREENING PROCESS FOR UTILIZING FINNED TUBE TECHNOLOGY

This section covers general items that need to be addressed when considering an application for external low finned tube technology. It is important to consider these items for finned tube applications in both new designs and existing exchanger modifications. The next section, **GRASSROOTS DESIGN VERSUS EXISTING EXCHANGER MODIFICATION**, addresses issues specific to the two types of applications.

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SCREENING PROCESS FOR UTILIZING FINNED TUBE TECHNOLOGY (Cont)

HEAT TRANSFER COEFFICIENTS

In order for the utilization of externally finned tubes to be beneficial, the shellside heat transfer coefficient must be lower than the tubeside heat transfer coefficient. The incentive for using finned tube technology increases as the difference between shellside and tubeside heat transfer coefficients increases and is especially attractive when the outside resistance is three or more times the inside resistance. When the shellside coefficient is not greater than the tubeside coefficient, it is unlikely that there will be significant economic incentive to using finned tubes over standard plain (i.e., bare) tubes.

FOULING

There is not a single general answer to the question of whether externally finned tubes can be used when the shellside stream fouls. Research and experiences, some within Exxon from more than 20 years ago (**Other References 3 and 4**) has shown that there are types of fouling for which finned tubes foul less and are easier to clean than bare tubes. In other cases, where a foulant formed between the fins and expanded during formation or correct corrosion resistant material was not available, tube failure has occurred. With the currently available range of materials and tube geometries, many fouling applications can be considered for a finned tube application.

CORROSION

The corrosion rate of the tube material must be considered when applying externally finned tubes. As shown in **Table 2**, finned tubes are available in a wide range of materials. The thickness of the fin depends on several factors, but is mainly controlled by the tube material and the fin spacing (i.e., number of fins per in. or meter). In general, the corrosion rates of higher alloy tube materials, such as titanium, are low and therefore standard fin thickness is adequate. However, when a corrosive service uses a low alloy tube, such as carbon steel, the designer should consider the effect of corrosion rate on fin thickness. A corrosion rate that is high relative to fin thickness may result in the fins corroding away in an unacceptably short period of time. When this situation is encountered, the designer should consider using tubes that have thicker fins or consider upgrading the metallurgy. See the "Specifying Finned Tubes" section below for more information on choosing the right type of finned tube.

GRASSROOT DESIGN VERSUS EXISTING EXCHANGER MODIFICATION

Finned tubes can be installed during the design of a new heat exchanger or as a modification to an existing heat exchanger. Implementing finned tubes during a grassroots design can result in an exchanger that is smaller in size, while implementing finned tubes to an existing exchanger can increase the exchanger's heat transfer capacity.

GRASSROOT DESIGN

Using finned tubes can reduce an exchanger's size by either a reduction in tube count, shorter tube length, or a decrease in the number of required shells. In general, the size of the reduction increases as the difference between the shellside and tubeside coefficient increases, where the tubeside coefficient is larger than the shellside coefficient.

Economics

Although a finned tube exchanger will be smaller than the required plain tube exchanger(s), the finning process adds additional cost to each tube. The effect this has on the tube price per unit length depends on the tube material. On a percentage basis, the increase in price per foot is less for higher alloy tubes compared to low alloy tubes. For example, the per unit length cost for carbon steel tubes may increase 75% for a finned tube, while the per unit length cost for titanium may increase 20%.

In addition, since exchangers that use externally finned tubes weight less and are more compact, the overall installation costs are less than that required for a plain tube exchanger.

EXISTING EXCHANGER MODIFICATION

Fin tube technology can be used as a debottlenecking or revamping tool for an existing plain tube heat exchanger. The fact that the diameter over the finned (DOF) portion of the tube is equal to or less than the diameter of the unfinned (D_o) portion (see **Figure 1**), permits this type of tube to be physically installed in the same diameter holes as would be required for bare tubes. By adding finned tubes, the heat transfer area is increased without any additional modifications to the exchanger. For example, the tube-for-tube replacement of bare tubes in a water-cooled condenser provides 2.5 or more times the condensing surface area compared to plain tubes, while retaining the same shell, piping, insulation, and other total erected costs that would be associated with the purchase of a new, larger heat exchanger.

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GRASSROOT DESIGN VERSUS EXISTING EXCHANGER MODIFICATION (Cont)

In addition to the requirements stated in the above section, **SCREENING PROCESS FOR UTILIZING FINNED TUBE TECHNOLOGY**, listed below are several items to consider before applying externally finned tubes to an existing plain tube heat exchanger.

Tubeside Pressure Drop

Although the tube count and tube OD will remain identical to the original bare tube arrangement, changing to externally finned tubes results in an increase in tubeside pressure drop. This is caused by the reduction in tube inside diameter necessary to produce the required fin height and tube wall thickness under the fin. For example, if a tube 1 in. (25.4 mm) OD tube is specified to have a fin height (see section SPECIFYING FINNED TUBES) of 0.049 in. (1.25 mm) with an average wall thickness of 0.065 in. (1.65 mm), the resulting inside diameter in the finned section will be 0.772 in. (18.34 mm), which is 11% less than the tube inside diameter of a 1 in. (25.4 mm) OD bare tube with a tube wall thickness of 0.065 in. (1.65 mm). With all other parameters constant, in general, tubeside pressure drop is inversely proportional to tube inside diameter to the fifth power. Therefore, a reduction in tube inside diameter of 11% may increase the tubeside pressure drop by approximately 80% ($1/0.89^5 = 1.79$). In finned tube applications where metallurgy has been upgraded, the new corrosion rate may allow a reduction in the required tube wall thickness under the fin. This reduction could help offset the tube inside diameter penalty on pressure drop. However, before any wall thickness reduction is made, a flow induced vibration analysis for the new operating conditions must be performed. See **Section IX-C** for more information on flow induced vibration.

Shellside Pressure Drop

The addition of externally finned tubes to an existing exchanger generally results in an decrease in shellside pressure drop. The decrease in pressure drop is a result of the effectively larger gap between adjacent tubes caused by the fin spacing. The magnitude of the decrease in pressure drop will vary for each application. Depending on other factors, such as vibration and tube unsupported length, it may be possible to offset the change in pressure drop by altering the baffle arrangement. See **Section IX-C**, for additional information on shellside pressure drop in shell and tube heat exchangers.

Tube Vibration

A vibration analysis should be performed when replacing existing plain tubes with externally finned tubes. A vibration analysis is required because a finned tube has a different natural frequency and critical velocity than a plain tube with the same tube wall thickness and diameter. In addition, finned tubes may also affect the shellside velocity and vortex shedding frequency (see **Section IX-C, Flow Induced Vibration**). When adding finned tubes and upgrading metallurgy, the lower corrosion rate of the upgraded metallurgy allows for the use of thinner walled tubes. The thinner tube wall may increase the potential for tube vibration.

Economic Incentives

In debottlenecking and revamping applications, the incentive for replacing plain tubes with finned tubes is to increase a unit's capacity either by allowing for an increase in product throughput or to transfer more heat for the same operating condition. Since finned tubes require no additional modifications, they can provide an effective alternative to adding additional shells or replacing an existing exchanger.

SPECIFYING FINNED TUBES

The geometry (e.g., FPI, fin height, fin thickness) of a finned tube depends on many factors but is strongly influenced by both the fin tube manufacturer and the tube material. The example fin tube dimensions listed in **Table 1** can be used for scoping purposes. Whenever considering the application of finned tube technology, the designer should contact the fin tube manufacturer and obtain the correct tube dimensions. The designer must supply the vendor with the following information: tube material, tube OD, allowable minimum wall thickness under the fins. The vendor can then supply the geometrical parameters including, fin spacing, fin height, minimum wall thickness, and average fin thickness, for the various option(s) available.

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SPECIFYING FINNED TUBES (Cont)

FIN SPACING AND HEIGHT

The choices that a designer has for fin spacing depends on the tube material and the manufacturer from whom the tubes are being purchased. For example, carbon steel finned tubes are commonly available with fin spacing per in. (FPI) of 11, 19, 26 and 36. However, it is unlikely that one individual manufacturer will make all four variations. In general, manufacturers try to obtain an outside surface area that is at least 2.5 times the surface area of a plain tube with the same OD. Therefore, if a manufacturer makes a tube that has a higher fin spacing (e.g., 19 FPI or 748 fins/meter) the tube will usually have fins that have a larger fin height than a tube that has lower fin spacing. **Table 2** shows commonly available number of fins per in. for a given tube material. As shown in the table, titanium tubes are only available in FPIs above 30 (1180 fins/meter). The high fin count is required to compensate for the low fin height, which is a result of a manufacturing limitation caused by the hardness of titanium. Depending on the manufacturer, titanium finned tubes may have a FPI between 30 and 43 (1180 and 1693 fins/meter).

Finned tubes, like closely spaced parallel plates, can retain liquid between the fins by capillary (surface tension) forces. The retained liquid covers the heat transfer surface and makes it less effective. This condition usually occurs during condensation. The HTRI computer models account for this reduction in heat transfer. The penalty is proportional to both the fin density (fins/length) and the surface-tension-to-density-ratio of the process fluid. For example, water produces a high flooding penalty. Therefore streams condensing steam would generally only benefit from finned tubes with low fin density (e.g., 11 fins/in. or 433 fins/meter).

In single phase application where the shellside Reynolds Number is below 500, consider using tubes with fin spacing no greater than 11 fins per inch (433 fins per meter). Closely spaced fins will be less effective because of the low shellside mixing.

FIN SHAPE

When integral-fin tubes (IFT) were first manufactured, the fin shape was determined by the manufacturing technology available. Since then, research has shown that the shape of the fin can significantly enhance heat transfer performance in addition to increasing the surface area. In single phase applications with low conductivity metal, shorter, more closely-spaced fins provide area enhancement with higher fin efficiency than longer fins. For condensing service, the actual shape of the fin, in addition to length, affects the heat transfer coefficient and removal of condensate. Some manufacturers have developed special fin shapes for condensation. In vaporizing service, especially reboilers, finned tubes with special shaped fins (e.g., Wieland GEWA-T see **Figure 6B** of **Section IX-A**) provide significant enhancement in heat transfer over traditional finned tubes.

FIN THICKNESS

Fin thickness is important with low alloy materials that have high corrosion rates. Fin thickness varies depending on the manufacturer, tube material, and fin spacing. **Table 2** lists the range of fin thickness for commonly available finned tubes. The fin thickness should be considered relative to expected tube life and corrosion rate.

MINIMUM TUBE WALL THICKNESS

As shown in **Figure A**, the minimum wall thickness, for finned tubes occurs under the fin. The difference between the minimum wall thickness and the average wall thickness depends on the tolerances of the manufacturing process.

LANDS

Lands are unfinned areas on the tube. These areas are at each end where the tube is within the tubesheet and sometimes at intermediate points to provide support where the tube passes through the support baffles. Because the finned outside diameter on some manufacturers tubes is smaller than the unfinned ends, a land area can provide tighter fit where the tube passes through the baffle. This can be helpful in a case where tube vibration is a problem. However, manufacturing technology is now available to produce finned tubes with the finned outside diameter equal to the unfinned ends. For such tubes, lands at the baffles provide negligible additional support. Finned tube applications in Exxon generally use tubes without land areas at the baffles.

INTERNAL TUBE ENHANCEMENTS

External finned tubes can also be combined with one of the tube side enhancements described in **Section IX-A**. Such combined technologies (e.g., IFT with integral tube side ribs/fins) are commercially available and are in successful applications at Exxon affiliates. Combinations of finned tubes with insertable type turbulence promoters or continuous, on-line mechanical cleaning devices (see **Section IX-A** for descriptions) can also be very cost effective and are in Exxon affiliate applications. For additional information see the ER&E Technical Reports in the reference section.

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THERMAL AND HYDRAULIC ANALYSIS

THEORY

In general the shell side heat transfer coefficient for a bare tube h_{bare} , is related to the total duty Q , bare tube outside surface area A_{bare} , and the difference in temperatures between the bare outside tube wall and the shell side fluid bulk temperature ΔT , as expressed in the following equation:

$$Q = (h_{bare}) (A_{bare}) (\Delta T)$$

For low-finned tubes the following differences between bare tubes and low-finned tubes must be accommodated:

- The outside surface area must account for the extra surface of the fins.
- The tubewall temperature must account for the variation along the height of the fin (i.e., fin efficiency).

The following equation shows the required modifications for low-finned tube application:

$$Q = (h_{bare} A_{bare} + h_{fin} \eta_f A_{fin}) (\Delta T)$$

where:

A_{bare}	=	Total outside bare tube surface area, ft ² (m ²)
A_{fin}	=	Total outside fin surface area, ft ² (m ²)
η_f	=	Theoretical fin efficiency, Dimensionless
h_{bare}	=	Bare tube heat transfer coefficient, Btu/hr ft ² F (W/m ² K)
h_{fin}	=	Fin tube heat transfer coefficient, Btu/hr ft ² F (W/m ² K)
Q	=	Heat duty, Btu/hr (W)

For the purpose of manual calculations it will be assumed that $h_{fin} = h_{bare}$ so the equation reduces to:

$$Q = h_{bare} (A_{bare} + \eta_f A_{fin}) (\Delta T)$$

CALCULATION PROCEDURE FOR EXTERNAL LOW FIN TUBES

COMPUTER PROGRAMS

HTRI's computer programs are recommended for exchanger thermal and hydraulic modeling. The programs have the capability to accurately model externally finned tube exchangers. The HTRI programs also perform a simplified vibration analysis.

HAND CALCULATION FORMS

This section specifies the modifications required to the hand-calculations in [Sections IX-D, IX-F, and IX-E](#) to account for the effect that externally finned tubes have on heat transfer and pressure drop. This method is applicable only to low fins and cannot be used for other types of extended surface exchangers. Results from these hand calculations should be verified by an HTRI computer program.

The dimensions necessary to calculate pressure drop and heat transfer across banks of low-finned tubes are illustrated in [Figure 1](#).

SECTION IX-D MODIFICATIONS

Required modifications to calculation form in [Section IX-D Calculation Procedure, No Change of Phase](#) are identified below. See [Section IX-D](#) for any undefined nomenclature. In addition, [Tables A-1 and A-1M](#), of [Section IX-G](#), provide a detailed calculation form which includes the modifications necessary to account for finned tubes when applied in single phase applications.

Estimation Method Calculation Form, Shell and Tube Heat Exchangers (No Change of Phase)

1. Values for single tube outside area per unit length, A_T , (equal to AO') should be obtained from [Table 1 or 1M](#) of this section.
2. Values for tube wall thickness in finned section, ℓ , should be defined as:

$$\ell = \frac{d_r - d_i'}{2}$$

Eq. (1)



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CALCULATION PROCEDURE FOR EXTERNAL LOW FIN TUBES (Cont)

- Equations using tube inside diameter, d_i , (e.g. velocity, mass velocity, and Reynolds Number) should be revised to use inside diameter of the low-finned tube, d'_i , where d'_i is shown in **Figure 1**.
- Equations using inside fouling resistance based on outside surface area, r_{io} , should be revised to use, r'_{io} , to account for finned surface area, where:

$$r'_{io} = r_i \left(\frac{AO'}{AI'} \right) \quad (AO' / AI' \text{ obtained from Table 1 or 1M}) \quad \text{Eq. (2)}$$

- Equations using inside heat transfer coefficient based on outside surface area, h_{io} , should be revised to use, h'_{io} , to account for the finned surface area, where:

$$h'_{io} = h_{io} \left(\frac{AI'}{AO'} \right) \quad \text{Eq. (3)}$$

- Equations using wall resistance, r_w , should be revised to use, r'_w , where:

$$r'_w = \frac{\ell}{(12)k_w} \frac{[d_r + 2 N'_f (12 \ell') (d_r + 12 \ell')]}{(d_r - \ell)} \quad (\text{Customary}) \quad \text{Eq. (4)}$$

$$r'_w = \frac{\ell}{1000 k_w} \frac{[d_r + 2 N'_f (1000 \ell') (d_r + 1000 \ell')]}{(d_r - \ell)} \quad (\text{Metric}) \quad \text{Eq. (4)M}$$

- Number of finned tubes, NTT' , should be calculated as:

$$NTT' = \frac{A_s}{(L - 0.5) AO'} \quad (AO' \text{ is obtained from Table 1}) \quad (\text{Customary}) \quad \text{Eq. (5)}$$

$$NTT' = \frac{A_s}{(L - 0.15) AO'} \quad (AO' \text{ is obtained from Table 1M}) \quad (\text{Metric}) \quad \text{Eq. (5)M}$$

- In the "Iteration, Shellside" subsection, equations using outside diameter, d_o , should be revised to use, DEQ , to account for the low-finned tube equivalent diameter, DEQ , instead of the bare tube outside diameter, where:

$$DEQ = d_r + (DOF - d_r) N'_f t'_f \quad \text{Eq. (6)}$$

- The calculation of Total Flow Reynolds Number, Re_{xt} , should be revised to account for the low-finned tube equivalent diameter as follows:

$$Re'_{xt} = \frac{(DEQ) (G'_{xt})}{29 \mu_b} \quad (\text{Customary}) \quad \text{Eq. (7)}$$

$$Re'_{xt} = (10^{-3}) \frac{(DEQ) (G'_{xt})}{\mu_b} \quad (\text{Metric}) \quad \text{Eq. (7)M}$$

- Equations using the shellside heat transfer coefficient, h_s , should be revised to use, h'_o , to account for the finned surface area, where:

$$h'_o = \frac{1}{R'_o} = 0.415 c_p (G'_{xt}) (FFB_h) (j_i) \left(\frac{c_p \mu_b}{k} \right)^{-2/3} \left(\frac{\mu_b}{\mu_w} \right)^{0.14} \beta \alpha_{fn} \quad (\text{Customary}) \quad \text{Eq. (8)}$$

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CALCULATION PROCEDURE FOR EXTERNAL LOW FIN TUBES (Cont)

$$h'_o = \frac{1}{R'_o} = 7.5 c_p (G'_{xt}) (FFB_h) (j_i) \left(\frac{c_p \mu_b}{0.414k} \right)^{-2/3} \left(\frac{\mu_b}{\mu_w} \right)^{0.14} \beta \alpha_{fh} \quad (\text{Metric}) \quad \text{Eq. (8)M}$$

where: α_{fh} is the finned tube heat transfer coefficient correction factor from **Figure 2** in **Section IX-G**

β is the shellside maldistribution correction factor from **Figure 5**

11. The shellside bundle friction term, HF, needs to be revised to use HF' to account for the low finned tubes as follows:

$$HF' = 0.00875 \left[\frac{12(f) (L) (m)}{(a) (PR) (DEQ) (n)} \right] \left[\frac{1}{\phi} \right] [F_s] \quad (\text{Customary}) \quad \text{Eq. (9)}$$

$$HF' = 0.00875 \left[\frac{1000(f) (L) (m)}{(a) (PR) (DEQ) (n)} \right] \left[\frac{1}{\phi} \right] [F_s] \quad (\text{Metric}) \quad \text{Eq. (9)M}$$

where: F_s is from **Table 4 Section IX-D** (December, 1995)

Overall Calculations

1. The calculated, clean overall heat transfer coefficient, U_c , as indicated on page 17 of **Section IX-D** should be revised to account for the weighted fin efficiency, E_w (obtained from **Figure 3 or 3M**) as follows:

$$U'_c = \frac{1}{R'_c} \quad \text{Eq. (10)}$$

where: R'_c is defined below:

$$R'_c = R'_{io} + r'_w + \frac{R'_o}{E_w} \quad \text{Eq. (11)}$$

2. The calculated, fouled overall heat transfer coefficient as indicated on page 17 of **Section IX-D** should be revised to account for the weighted fin efficiency, E_w (obtained from **Figure 3 or 3M**) as follows:

$$U'_D = \frac{1}{R'_t} \quad \text{Eq. (12)}$$

where: R'_t is defined below:

$$R'_t = R'_{io} + r'_{io} + r'_w + \frac{R'_o}{E_w} + r'_o \quad \text{Eq. (13)}$$

SECTION IX-E MODIFICATIONS

Required modifications to the **Section IX-E** procedure are identified below. See **Section IX-E** for any undefined nomenclature.

Heat Transfer Calculation Form for Kettle and Internal Reboilers

Exchanger Geometry

- Equations using inside diameter, d_i , (e.g. velocity, mass velocity, and Reynolds number) should be revised to use inside diameter of the low-finned tube, d'_i , where d'_i is shown in **Figure 1**.
- Ratio of outside to inside surface area, AOI, should be obtained from **Table 1 or 1M** using AOI'. All equations using AOI should be revised to use AOI', to account for the finned surface area.

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CALCULATION PROCEDURE FOR EXTERNAL LOW FIN TUBES (Cont)

Nucleate Boiling Calculations

The design practice for vaporization determines an average shellside boiling coefficient, h_{ab} , based on correction factors as shown on page 22 item 12. The finned tube correction, F_f , already accounts for the difference between bare tube outside surface and low-finned tube outside surface, F_s , as well as fin efficiency, F_e . When using low-finned tubes, incorporate the correction factors.

- Equations using wall resistance, r_w , should be revised to use, r'_w , were:

$$r'_w = \frac{\ell}{(12)k_w} \frac{[d_r + 2 N_f' (12 \ell') (d_r + 12 \ell')]}{(d_r - \ell)} \quad (\text{Customary}) \quad \text{Eq. (14)}$$

$$r'_w = \frac{\ell}{1000 k_w} \frac{[d_r + 2 N_f' (1000 \ell') (d_r + 1000 \ell')]}{d_r - \ell} \quad (\text{Metric}) \quad \text{Eq. (14)M}$$

Heat Transfer Calculation Form for Vertical Tubeside Thermosiphon Reboilers

The annular external finned tubes discussed in this design practice, should not be used in vertical thermosiphon reboilers when the shellside fluid is condensing. The condensing fluid flowing in the vertical direction does not effectively utilize the external finned surface area. Other heat transfer enhancement technologies are available, such as the Double Enhanced Tube-Internal Nucleate Boiling and Shellside Longitudinal Finned Tube shown in **Figure 6A Section IX-C** (December 1995).

DESIGN PRACTICE IX-F MODIFICATIONS

Required modifications to the **Section IX-F** procedure are identified below. See **Section IX-F** for any undefined nomenclature.

I. Zone Duties and Temperatures

- No modifications required.

II. Shellside Condensation

Note, external annular finned tubes should not be used in vertical shellside condensing applications for reasons indicated in the previous section.

- In all equations, the total exchanger area, A , refers to the total (bare plus finned) outside surface area.
- Equations using inside diameter, d_i , (e.g. velocity, mass velocity, and Reynolds number) should be revised to use inside diameter of the low-finned tube, d'_i , where d'_i is shown in **Figure 1**.
- Revise calculation of number of tubes, N_t , in step 6 on page 22 and step 7 on page 35 of **Section IX-F** as follows:

$$N_t = \frac{A_s}{(L - 0.5) AO'} \quad (\text{Customary}) \quad \text{Eq. (15)}$$

$$N_t = \frac{A_s}{(L - 0.0152) AO'} \quad (\text{Metric}) \quad \text{Eq. (15)M}$$

- All equations using tube inside coefficient referred to the outside area h_{io} should be revised to use tube side inside coefficient referred to finned area, h'_{io} , where:

$$h'_{io} = h_{io} \left(\frac{AI'}{AO'} \right) \quad \text{Eq. (16)}$$

- Equations using inside fouling resistance based on outside surface area, r_{io} should be revised to use r'_{io} , where:

$$r'_{io} = r_{io} \left(\frac{AO'}{AI'} \right) \quad \text{Eq. (17)}$$

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CALCULATION PROCEDURE FOR EXTERNAL LOW FIN TUBES (Cont)

6. Equations using tube wall resistance r_w should be revised to use r'_w where r'_w is calculated as follows:

$$r'_w = \frac{\ell}{(12)k_w} \frac{[d_r + 2 N'_f (12 \ell') (d_r + 12 \ell')]}{(d_r - \ell)} \quad (\text{Customary}) \quad \text{Eq. (18)}$$

$$r'_w = \frac{\ell}{1000 k_w} \frac{[d_r + 2 N'_f (1000 \ell') (d_r + 1000 \ell')]}{(d_r - \ell)} \quad (\text{Metric}) \quad \text{Eq. (18)M}$$

7. Values for A_{dh} , A_{ds} , refer to the outside surface area including the finned area.
8. All shell side heat transfer coefficients calculated per procedures used in [Section IX-D](#) should be multiplied by E_w , obtained from [Figure 3 or 3M](#) of this section to correct for fin efficiency.
9. All shell side condensing coefficients (h_λ) calculated per [Section IX-F](#) should be multiplied by the weighted fin efficiency, E_w , obtained from [Figure 3 or 3M](#) of this section.
10. For Note 1 on page 49, [Section IX-F](#) (June, 1994), tube sizes refer to the plain tube end outside diameter, d_o .
11. Correct maximum area per shell in Note 2 on page 49, [Section IX-F](#) (June, 1994) by multiplying the values by:

$$\left(\frac{AO'}{AI'} \right) \left(\frac{d'_i}{d_o} \right) \quad \text{Eq. (19)}$$

III. Tubeside Condensation

1. In all equations, the total (bare plus finned) exchanger area, A , refers to the total outside area.
2. Equations using inside diameter, d_i , (e.g. velocity, mass velocity, and Reynolds number) should be revised to use the inside diameter of the low-finned tube, d_i .
3. Calculation for number of tubes, N_t , should be revised as follows:

$$N_t = \frac{A_s}{(L - 0.5) AO'} \quad (\text{Customary}) \quad \text{Eq. (20)}$$

$$N_t = \frac{A_s}{(L - 0.0152) AO'} \quad (\text{Metric}) \quad \text{Eq. (20)M}$$

4. Equations using tube inside heat transfer coefficient referred to the outside area should be revised to use tube inside coefficient referred to finned area.
5. Equations using inside fouling resistance based on outside surface area, r_{io} , should be revised to use r'_{io} , where:

$$r'_{io} = r_{io} \left(\frac{AO'}{AI'} \right) \quad \text{Eq. (21)}$$

6. Equations using tube wall resistance, r_w , should be revised to use r'_w where r'_w is calculated as follows:

$$r'_w = \frac{\ell}{(12)k_w} \frac{[d_r + 2 N'_f (12 \ell') (d_r + 12 \ell')]}{(d_r - \ell)} \quad (\text{Customary}) \quad \text{Eq. (22)}$$

$$r'_w = \frac{\ell}{1000 k_w} \frac{[d_r + 2 N'_f (1000 \ell') (d_r + 1000 \ell')]}{(d_r - \ell)} \quad (\text{Metric}) \quad \text{Eq. (22)M}$$

7. Values for A_{dh} , A_{ds} , A_{sc} , refer to the outside surface area including the finned area.
8. For Note 1 on page 49, [Section IX-F](#) (June, 1994), tube sizes refer to the plain tube end outside diameter, O.D.
9. Correct maximum area per shell in Note 2 on page 49, [Section IX-F](#) (June, 1994) by multiplying the values by:

$$\left(\frac{AO'}{AI'} \right) \left(\frac{d'_i}{\text{O.D.}} \right) \quad \text{Eq. (23)}$$

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NOMENCLATURE

a	=	Tube row spacing factor, dimensionless
A	=	Bundle outside surface area (bare + finned), ft ² (m ²)
A_s	=	Surface area per shell, ft ² (m ²)
AI'	=	Finned tube inside surface area per unit length, ft ² / ft (m ² / m)
AO'	=	Finned tube outside surface area per unit length, ft ² / ft (m ² / m)
AOI'	=	Finned tube ratio of outside to inside surface area, dimensionless
A_T	=	Single tube outside surface area per unit length, ft ² / ft (m ² / m)
A_x	=	Free flow area between baffles, ft ² (m ²)
BC	=	Baffle cut as percent of (DS)
$(BC)_e$	=	Effective baffle cut factor, dimensionless
c_p	=	Specific heat, Btu/lb °F (kJ/kg °C)
d_i	=	Unfinned end inside diameter, in. (mm)
d'_i	=	Inside diameter of finned section for low-finned tube, in. (mm)
d_o	=	Plain tube end outside diameter, in. (mm)
d_r	=	Root diameter of finned tube, in. (mm)
DEQ	=	Projected equivalent outside diameter of low finned tubes, in. (mm)
DN	=	Nominal nozzle I.D., in. (mm)
DOF	=	Tube outside diameter (over fins), in. (mm)
$DOTL$	=	Diameter of bundle outer tube limit, in. (mm)
DS	=	Shell inside diameter, in. (mm)
$DSNI$	=	Shell side inlet nozzle I.D., in. (mm)
$DSNO$	=	Shell side outlet nozzle I.D., in. (mm)
$DTNI$	=	Tube side inlet nozzle I.D., in. (mm)
$DTNO$	=	Tube side outlet nozzle I.D., in. (mm)
E_w	=	Weighted fin efficiency, dimensionless
f	=	Non-isothermal friction factor, dimensionless
f_{is}	=	Isothermal friction factor, dimensionless
F_n	=	Correction factor for log mean temperature difference, dimensionless
F_s	=	Shell side pressure drop correction factor, dimensionless
FFB_h	=	Crossflow fraction for heat transfer, dimensionless
FFB_p	=	Crossflow fraction for pressure drop, dimensionless
Gr	=	Grashof Number, dimensionless
G'_{xt}	=	Total cross-flow mass velocity with finned tubes, lb/hr ft ² (kg/sm ²)
h_{io}	=	Inside film coefficient based on tube outside surface area, Btu/hr ft ² °F (W/m ² °C)
h'_{io}	=	Inside film coefficient, corrected to outside finned tube area, Btu/hr ft ² °F (W/m ² °C)
h_o	=	Outside film coefficient based on finned tube, Btu/hr ft ² °F (W/m ² °C)
HF'	=	Finned tube shellside friction term, dimensionless
HM	=	Shell side momentum term, dimensionless
j_i	=	Stanton number type heat transfer factor, dimensionless
k	=	Thermal conductivity, Btu/hr ft ² °F (W/m ² °C)
k_w	=	Thermal conductivity of tube wall at average tube temperature, Btu/hr ft ² °F (W/m ² °C)
K_e	=	Tube side pressure drop coefficient, dimensionless



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NOMENCLATURE (Cont)

L	=	Tube length, ft (m)
L'	=	Tube length between baffles, ft (m)
L_e	=	Effective tube length, ft (m)
LBCC	=	Central baffle pitch, in. (mm)
LI	=	Tube flow length, in. (mm)
LMTD	=	Log mean temperature difference for true counter current flow °F (°C)
ℓ	=	Tube wall thickness under the fin, in. (mm)
ℓ'	=	Fin height, ft (m)
m	=	Shell side flow factor, dimensionless
MTD	=	Corrected log mean temperature difference, °F (°C)
n	=	Baffle spacing to bundle diameter ratio, dimensionless
N	=	Number of tubes per pass
N'_f	=	Number of fins per unit length, in. ⁻¹ (mm ⁻¹)
N_p	=	Number of shells in parallel
N_T	=	Total number of shells
N_t	=	Number of tubes per bundle
NTP	=	Number of tube passes per shell
NTT'	=	Number of finned tubes per bundle
p	=	Baffle flow factor, dimensionless
Pr	=	Prandtl number, dimensionless
PR	=	Tube pitch ratio, dimensionless
PT	=	Tube pitch, in. (mm)
ΔP_e	=	Tube entrance, expansion, and turnaround pressure drop, psi (kPa)
ΔP_{exch}	=	Total nozzle to nozzle shell side pressure drop, psi (kPa)
ΔP_n	=	Tube side nozzle pressure drop, psi (kPa)
ΔP_s	=	Shell side pressure drop (excluding nozzles), psi (kPa)
ΔP_{sn}	=	Shell side nozzle pressure drop, psi (kPa)
ΔP_t	=	Tube side frictional pressure drop, psi (kPa)
$(\Delta P_t)_{nn}$	=	Total tube side nozzle pressure drop, psi (kPa)
Q	=	Rate of heat transfer, Btu/hr (W)
r_i	=	Inside fouling factor referred to tube inside surface area, hr ft ² °F/Btu (m ² °C/W)
r_{io}	=	Inside fouling factor referred to outside surface area, hr ft ² °F/Btu (m ² °C/W)
r'_{io}	=	Inside fouling factor referred to tube outside surface area of low-finned tube, hr ft ² °F/Btu (m ² °C/W)
r'_o	=	Shell side fouling factor, based on finned tube hr ft ² °F/Btu (m ² °C/W)
r'_w	=	Resistance of low-finned tube wall at average wall temperature, hr ft ² °F/Btu (m ² °C/W)
RB	=	Geometry factor, dimensionless
RC	=	Reynolds number correction factor, dimensionless
R'_c	=	Total resistance (clean) to heat transfer for finned tube, hr ft ² °F/Btu (m ² °C/W)
Re	=	Reynolds number, dimensionless
Re'_{xh}	=	Low-finned tube cross flow Reynolds number for heat transfer, dimensionless
Re'_{xp}	=	Low-finned tube cross flow Reynolds number for pressure drop, dimensionless
Re'_{xt}	=	Low-finned tube total flow Reynolds number, dimensionless



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NOMENCLATURE (Cont)

R'_{io}	=	Inside resistance to heat transfer corrected to outside area of low-finned tube , hr ft ² °F/Btu (m ² °C/W)
R'_o	=	Outside film resistance to heat transfer based on low-finned tube, hr ft ² °F/Btu (m ² °C/W)
R'_t	=	Total resistance (duty) to heat transfer based on low-finned tubes, hr ft ² °F/Btu (m ² °C/W)
SC	=	Baffle spacing correction factor, dimensionless
S_{TT}	=	Tube sheet material allowable stress at design temperature, lb/sq. in. (kPa)
t_1	=	Inlet temperature of fluid being heated, °F (°C)
t_2	=	Outlet temperature of fluid being heated, °F (°C)
t'_f	=	Average fin thickness, in. (mm)
T_1	=	Inlet temperature of fluid being cooled °F (°C)
T_2	=	Outlet temperature of fluid being cooled, °F (°C)
T_{DS}	=	Shell side design temperature, °F (°C)
T_{DT}	=	Tube side design temperature, °F (°C)
T_M	=	Tube sheet design temperature, °F (°C)
TS_b	=	Bulk temperature of shell side fluid, °F (°C)
TS_{in}	=	Inlet temperature of shell side fluid, °F (°C)
TS_{out}	=	Outlet temperature of shell side fluid, °F (°C)
TT_b	=	Tube side fluid bulk temperature, °F (°C)
TT_{in}	=	Inlet temperature of tube side fluid, °F (°C)
TT_{out}	=	Outlet temperature of tube side fluid, °F (°C)
TTT	=	Total tube sheet thickness, ft (m)
T_w	=	Average wall temperature, °F (°C)
ΔT_a	=	Terminal temperature difference, °F (°C)
ΔT_s	=	Shell side temperature difference, °F (°C)
U'_c	=	Overall clean coefficient of heat transfer for low-finned tubes, Btu/hr ft ² °F (W/m ² °C)
U'_d	=	Calculated overall fouled coefficient of heat transfer for low finned tube, Btu/hr ft ² °F (W/m ² °C)
U'_o	=	Overall duty coefficient of heat transfer for low-finned tubes, Btu/hr ft ² °F (W/M ² °C)
V_n	=	Tube side average nozzle fluid velocity, ft/sec (m/s)
V_t	=	Fluid velocity in tubes, ft/sec (m/s)
W	=	Tube side mass rate Lb/hr (kg/s)
W_s	=	Shell side mass rate, Lb/hr (kg/s)
X_t	=	Length perpendicular to flow direction across a tube field for a single gap, ft (m) [see HTRI design manual Volume 1 Figure (2.2.1)]
Y_{th}	=	Tube side heat transfer correlation factor
α_{fh}	=	Low finned tube heat transfer correction, dimensionless
β	=	Heat transfer baffle cut correction, dimensionless
β'	=	Coefficient of thermal expansion used in Grashof Number, °F ⁻¹ (°C ⁻¹)
ε	=	Short tube laminar flow correction factor, dimensionless
η	=	Transitional flow heat transfer proration factor, dimensionless
η_f	=	Theoretical fin efficiency, dimensionless
λ	=	Tube length correction, dimensionless
γ^*	=	Natural convection correction, dimensionless
μ_b	=	Viscosity at bulk temperature, cP (mPa s)



NOMENCLATURE (Cont)

μ_w	=	Viscosity at wall temperature, cP (mPa•s)
ρ	=	Density, lb/ft ³ (kg/m ³)
ϕ	=	Viscosity correction for wall temperature
ψ	=	Low Prandtl Number correction factor, dimensionless
ψ_p	=	Natural convection correction factor, dimensionless
ξ	=	Baffle correction factor, dimensionless

TABLE 1
PHYSICAL DATA FOR TYPICAL LOW-FINNED TUBES (CUSTOMARY UNITS)

Note: These valves are for scoping purposes only.

- fins per inch = 11
- fin height = 0.117 in.
- fin thickness = 0.011 in.

PLAIN TUBE END		FINNED SECTION					
O.D. (in.)	WALL THICKNESS (in.)	DOF (in.)	d_r (in.)	d'_i (in.)	ℓ' (in.)	AO' (ft ² /ft)	AO'/AI'
0.75	0.058	0.736	0.502	0.452	0.025	0.526	4.51
0.75	0.065	0.736	0.502	0.440	0.031	0.526	4.65
1.00	0.065	0.986	0.752	0.684	0.034	0.743	4.20
1.00	0.072	0.986	0.752	0.674	0.039	0.743	4.28

Note: These valves are for scoping purposes only.

- fins per inch = 19
- fin height = 0.057 in.
- fin thickness = 0.018 in.

PLAIN TUBE END		FINNED SECTION					
O.D. (in.)	WALL THICKNESS (in.)	DOF (in.)	d_r (in.)	d'_i (in.)	ℓ' (in.)	AO' (ft ² /ft)	AO'/AI'
0.75	0.042	0.741	0.627	0.577	0.025	0.508	3.40
0.75	0.047	0.741	0.627	0.565	0.031	0.508	3.48
0.75	0.058	0.741	0.627	0.553	0.037	0.508	3.57
0.75	0.065	0.741	0.627	0.539	0.044	0.508	3.66
0.75	0.083	0.741	0.627	0.511	0.058	0.508	3.90
0.75	0.095	0.741	0.627	0.479	0.074	0.508	4.21
0.75	0.109	0.741	0.627	0.459	0.084	0.508	4.44
1.00	0.058	0.991	0.877	0.803	0.037	0.695	3.35
1.00	0.065	0.991	0.877	0.789	0.044	0.695	3.41
1.00	0.083	0.991	0.877	0.761	0.058	0.695	3.55
1.00	0.095	0.991	0.877	0.729	0.074	0.695	3.73

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TABLE 1 (Cont)
PHYSICAL DATA FOR TYPICAL LOW-FINNED TUBES (CUSTOMARY UNITS)

Note: These valves are for scoping purposes only.

- fins per inch = 28
- fin height = 0.037 in.
- fin thickness = 0.012 in.

PLAIN TUBE END		FINNED SECTION					
O.D. (in.)	WALL THICKNESS (in.)	DOF (in.)	d_r (in.)	d_i (in.)	ℓ' (in.)	AO' (ft ² /ft)	AO'/AI'
0.75	0.049	0.750	0.676	0.620	0.028	0.521	3.216
0.75	0.065	0.750	0.676	0.578	0.049	0.521	3.450
0.75	0.083	0.750	0.676	0.546	0.065	0.521	3.643
0.75	0.109	0.750	0.676	0.510	0.083	0.521	3.888
1.00	0.049	1.000	0.926	0.870	0.028	0.704	3.088
1.00	0.065	1.000	0.926	0.828	0.049	0.704	3.244
1.00	0.083	1.000	0.926	0.796	0.065	0.704	3.385
1.00	0.109	1.000	0.926	0.760	0.083	0.704	3.538

Note: These valves are for scoping purposes only.

- fins per inch = 36
- fin height = 0.026 in.
- fin thickness = 0.012 in.

PLAIN TUBE END		FINNED SECTION					
O.D. (in.)	WALL THICKNESS (in.)	DOF (in.)	d_r (in.)	d_i (in.)	ℓ' (in.)	AO' (ft ² /ft)	AO'/AI'
0.75	0.049	0.750	0.698	0.642	0.028	0.5	2.98
0.75	0.058	0.750	0.698	0.628	0.035	0.5	3.05
0.75	0.065	0.750	0.698	0.614	0.042	0.5	3.11
0.75	0.072	0.750	0.698	0.600	0.049	0.5	3.19
0.75	0.083	0.750	0.698	0.568	0.065	0.5	3.36
0.75	0.109	0.750	0.698	0.532	0.083	0.5	3.60
1	0.058	1.0	0.948	0.878	0.035	0.671	2.92
1	0.065	1.0	0.948	0.864	0.042	0.671	2.97
1	0.072	1.0	0.948	0.850	0.049	0.671	3.01
1	0.083	1.0	0.948	0.818	0.065	0.671	3.14
1	0.109	1.0	0.948	0.782	0.083	0.671	3.27

TABLE 1M
PHYSICAL DATA FOR TYPICAL LOW-FINNED TUBES (METRIC UNITS)

Note: These valves are for scoping purposes only.

- fins per meter = 433
- fin height = 2.972 mm
- fin thickness = 0.279 mm

PLAIN TUBE END		FINNED SECTION					
O.D. (mm)	WALL THICKNESS (mm)	DOF (mm)	d_r (mm)	d'_i (mm)	ℓ' (mm)	AO' (m ² /m)	AO'/AI'
19.05	1.473	18.695	12.751	11.481	0.635	0.160	4.51
19.05	1.651	18.695	12.751	11.177	0.787	0.160	4.65
25.4	1.651	25.045	19.101	17.373	0.864	0.226	4.20
25.4	1.829	25.045	19.101	17.121	0.990	0.226	4.28

Note: These valves are for scoping purposes only.

- fins per meter = 748
- fin height = 1.448 m
- fin thickness = 0.457 mm

PLAIN TUBE END		FINNED SECTION					
O.D. (mm)	WALL THICKNESS (mm)	DOF (mm)	d_r (mm)	d'_i (mm)	ℓ' (mm)	AO' (m ² /m)	AO'/AI'
19.05	1.067	18.822	15.926	14.656	0.635	0.155	3.40
19.05	1.194	18.822	15.926	14.352	0.787	0.155	3.48
19.05	1.473	18.822	15.926	14.046	0.940	0.155	3.57
19.05	1.651	18.822	15.926	13.690	1.118	0.155	3.66
19.05	2.108	18.822	15.926	12.980	1.473	0.155	3.90
19.05	2.413	18.822	15.926	12.166	1.880	0.155	4.21
19.05	2.769	18.822	15.926	11.658	2.134	0.155	4.44
25.4	1.473	25.172	22.276	20.396	0.940	0.212	3.35
25.4	1.651	25.172	22.276	20.040	1.118	0.21	3.41
25.4	2.108	25.172	22.276	19.330	1.473	0.212	3.55
25.4	2.413	15.172	22.276	18.516	1.880	0.21	3.73