3.4. Examples of Design Problems for Low- and Medium-Finned Trufin in Shell and Tube Condensers

3.4.1. Condenser Design for a Pure Component: Example Problem

Statement of Problem

Design a condenser to condense 2,630,000 lb/hr of propane at 148°F and 200 psia (saturation temperature is 105.2°F) using water at 86°F. Maximum allowable pressure drop is 10 psi for the propane and 25 psi for the water. Minimum water velocity is 8 ft/sec. Fouling resistances are 0.001 hrft$_2$°F/Btu for the water and 0.0003 for the propane, each based upon the respective heat transfer areas. An E or J shell design is to be used, with maximum allowable baffle spacing and cut. Wolverine S/T Trufin tubes, 3/4 in. by 16 BWG, 19 fins/in., of 70/30 CuNi are to be used. Maximum allowable tube length is 80 feet. Fixed tube sheet construction is satisfactory.

Physical properties:	Propane at 105.2°F and 200 psia Liquid	Vapor	Water at 90°F
Density, lb/ft^3	29.3	1.85	62.1
Viscosity, lb/ft hr	0.194	0.021	1.79
Thermal conductivity, Btu/hr ft°F	0.074	--	0.358
Specific heat, Btu/lb°F	--	0.39	1.00
Latent heat, Btu/lb	138.1		–

Some Comments Upon the Design

This problem is similar to an actual design performed for the propane condensers for a large LNG plant using brackish water as a coolant. The high water velocity is intended to minimize fouling, and 70/30 CuNi is selected for its resistance to corrosion. The original design was for a rod baffle exchanger, but a conventionally baffled shell is chosen here solely to illustrate the shell side pressure drop calculation.

Tube characteristics (nominal)	
Outside diameter	0.750 in.
Inside diameter	0.508 in.
Root diameter	0.638 in.
Fin height	0.056 in.
Fin thickness	0.011 in.
Outside heat transfer area	0.503 ft^2/ft
Inside heat transfer area	0.130 ft^2/ft
Outside/inside area ratio	3.86
Inside flow area per tube	0.195 in.^{2}
Thermal conductivity	17 Btu/hr ft°F
Fin resistance	7.1×10^{-4} hr ft^2°F/Btu

Preliminary Estimate of Size

1. Heat duty

The heat duty is composed of two parts: the sensible heat of desuperheating the vapor from 148°F to 105.2°F and the latent heat of condensation at 105.2°F.

Desuperheat:

$$Q_{DSH} = WC_p(T_{SH} - T_{SAT}) \quad (3.92)$$

$$Q_{DSH} = (2{,}630{,}000 \text{ lb/hr})(0.39 \text{ Btu/lb°F})(148 - 105.2)\text{°F} = 4.39\text{x}10^7 \text{ Btu/hr}$$

Laten heat:

$$Q_c = (2{,}630{,}000 \text{ lb/hr})(138.1 \text{ Btu/lb}) = 3.63x10^8 \text{ Btu/hr} \quad (3.62)$$

Total duty = $4.07x10^8$ Btu/hr

It will be assumed that the total duty is transferred by condensation at a constant temperature of 105.2°F

2. Mean temperature difference

A design outlet water temperature must be chosen. Try 95°F for a first set of calculations.

Then

$$LMTD = \frac{95-86}{\ell n\left(\frac{105.2-86}{105.2-95}\right)} = 14.2°F \quad (3.56)$$

Since the condensing side is isothermal, F = 1.00.

We may also calculate the water requirement at this time:

$$w_{H_2O} = \frac{4.07x10^8 \quad Btu/hr}{(9.0°F)(1.00 \quad Btu/lb°F)} = 4.52x10^7 \quad lb/hr \quad (3.64)$$

3. Overall heat transfer coefficient

Estimate the various coefficients and resistances, based on their respective heat transfer areas:

Condensing: h_o = 500 Btu/hr ft^2°F, based on total outside heat transfer area.

Fouling, outside: $R_{fo} = 3.0 \times 10^{-4}$ hr ft^2°F/Btu, same basis

Fin resistance: $R_{fin} = 7.1 \times 10^{-4}$ hr ft^2°F/Btu, same basis
0.065 in.

Wall resistance: $$R_w = \frac{0.065 \quad in.}{(12 \quad in/ft)(17 \quad Btu/hr \quad ft°F)}$$

= 0.00032 hr ft^2°F/Btu, based on the mean wall diameter,

Water: h_i = 1600 Btu/hr ft^2°F, based on the inside tube area from Fig. 2.19

Fouling, inside: R_{fi} = 0.001 hr ft^2°F/Btu, same basis

Combining these, with the appropriate corrections to put the overall coefficient on the basis of the total outside heat transfer area gives:

$$U_o = \frac{1}{\frac{1}{500}+0.0003+7.1x10^{-4}+0.00032\left[\frac{0.503}{\pi\left(\frac{0.573}{12}\right)}\right]+\left(\frac{1}{1600}+0.001\right)\left(\frac{0.503}{0.130}\right)} = 96.4 \;\; Btu/hr \;\; ft^2 {}^\circ F \tag{3.55}$$

based on total outside heat transfer area.

4. Calculation of area

$$A_o = \frac{4.07x10^8 \;\; Btu/hr}{(96.4 \;\; Btu/hr \;\; ft^2 {}^\circ F)(14.2^\circ F)} = 2.97x10^5 \;\; ft^2 \tag{3.54}$$

5. Estimation of shell dimensions

We may use the methods described in Chapter 2. Specifically, we may use Fig. 2.26, but first we must compute the effective area A'_o with which to enter the chart:

$A'_o = AoF_1F_2F_3F_4$

where A_o is the actual area required in the heat exchanger, $2.97x10^5$ ft^2 for the present case

F_1 is a correction factor for the unit cell array. Since there is no apparent reason why a 3/4 in. OD tube on 15/16 in. triangular layout will not serve, F_1 = 1.00.

F_2 is a correction factor for the number of tube passes. If possible, we will use one pass, so F_2 = 1.00 for present.

F_3 is a correction factor for the shell construction. Fixed tube sheet construction is satisfactory, so F_3 = 1.00.

F_4 is the correction factor for the specific fin geometry and density. For 3/4 in. O.D. 19 fins/in. construction, F_4 = 1.00.

so

$A_o' = 2.97 \text{ X } 10^5 \; ft^2 \; (1.00)(1.00)(1.00)(1.00) = 2.97 \text{ x } 10^5 \; ft^2$

Referring to Figure 2.26, we see that a combination of diameter and length (on the figure) that will meet this requirement is a 120 in. I.D. shell, with tubes about 41 feet long. Referring to Table 2.6, we see that the tube count for a one pass exchanger is about 14,500. This gives a total area of

A_o = (14,500 tubes)(0.503 ft^2/ft)(41 ft) = 299,034 ft^2, compared to the 297,000 ft^2 estimated. This would allow a 40.33 ft long tube.

Check the water side velocity:

$$V_{H_2O} = \frac{(4.52x10^7 \;\; lb/hr)(144 \;\; in^2/ft^2)}{14{,}500(62.1 \;\; lb/ft^3)(3600 \;\; \sec/hr)(0.195 \;\; in.^2)} = 10.3\, ft/\sec$$

This velocity is above the stated minimum; check the pressure drop:

$$\mathrm{Re}_i = \frac{\left(\frac{0.508}{12} \; ft\right)(62.1 \;\; lb/ft^3)(10.3 \;\; ft/\sec)(3600 \;\; \sec/hr)}{1.79 \;\; lb/ft \;\; hr} = 54{,}450 \qquad (2.19)$$

f_i = 0.0046 from Fig. 2.20

$$\Delta p_f = \frac{2(0.0046)(62.1 \;\; lb/ft^3)(10.4 \;\; ft/\sec)^2(40.33 \;\; ft)}{\left(\frac{0.508}{12} \; ft\right)\left(32.2 \frac{lb_m \, ft}{lb_f \sec^2}\right)\left(144 \frac{in^2}{ft^2}\right)} = 12.45 \;\; psi \qquad (2.26)$$

(neglecting the Sieder-Tate term)

The entrance pressure loss is:

$$\Delta p_{ent} = \frac{3(62.1 \;\; lb/ft^3)(10.3 \;\; ft/\sec)^2}{2\left(32.2 \frac{lb_m \, ft}{lb_f \sec^2}\right)\left(144 \frac{in^2}{ft^2}\right)} = 2.1 \;\; psi \qquad (2.25)$$

so the calculated tube-side pressure drop is within the design limit. Proceed with this design.

Shells of this diameter fall outside standard TEMA specifications on baffle spacing. It should be mechanically conservative to design to the maximum unsupported span of 52 in. (i.e., baffle spacing of 26 in. for a fully tubed bundle) recommended in Table R-4.52 of Ref. (1). If this gives excessive velocities or pressure drops, we may then switch the design to a J or an X shell, or consider a rod baffle design.

Heat Transfer Calculation

Having a general idea of the design, it is now necessary to check the specific values of the heat transfer coefficients.

1. The condensing coefficient

First, calculate the coefficient by Eq. (3.80). W is the mass of vapor condensed per tube in unit time:

$$W = \frac{2{,}630{,}000 \;\; lb/hr}{14{,}500 \;\; tubes} = 181.3 \;\; (lb/hr)/tube$$

Then

$$h_c = 0.951\left[\frac{\left(0.074\,\frac{Btu}{hr\ ft°F}\right)^3\left(29.3\,\frac{lb}{ft^3}\right)\left(27.4\,\frac{lb}{ft^3}\right)\left(4.71x10^8\ \frac{ft\ lb_m}{hr^2 lb_f}\right)\left(40.33\quad ft\right)}{\left(0.194\,\frac{lb}{ft\ hr}\right)\left(181.3\,\frac{lb}{hr}\right)}\right]^{1/3} = 511\quad Btu/hr\quad ft^2\,°F$$

We may check this against the Beatty-Katz prediction, Eq. 3.81 to 3.89). The various quantities are:

$$A_{root} = \pi\left(\frac{0.638}{12}\ ft\right)\left(40.33\quad ft\right) = 6.7\quad ft^2$$

$$A_{fin} = \frac{\pi}{2}\left[\left(\frac{0.750^2 - 0.638^2}{144}\right)ft^2\right]\left[19(12)\frac{fins}{ft}\right]\left(40.33\quad ft\right) = 15.6\quad ft^2$$

$$m = \left(\frac{0.056}{12}\ ft\right)\sqrt{\frac{2}{\left[\left(\frac{1}{511}+0.0003\right)\frac{hr\ ft^2°F}{Btu}\right]\left(17\,\frac{Btu}{hr\ ft°F}\right)\left(\frac{0.011}{12}\ ft\right)}} = 1.11$$

In principle, this quantity must be recalculated using the new value of h_c until the solution converges.

$$\Phi = \frac{1}{1+\frac{(1.11)^2}{3}\sqrt{\frac{0.750in.}{0.638in.}}} = 0.692$$

$$A_{eq} = 0.692(15.6\quad ft^2) + \pi\left(\frac{0.638}{12}\ ft\right)\left(40.33\quad ft\right)\left(\frac{0.0265\quad in.}{0.056\quad in.}\right) = 14.0\quad ft^2$$

$$a_{fin} = \frac{\pi}{4}\left(\frac{0.750^2 - 0.638^2}{144}\right)ft^2 = 8.5x10^{-4}\quad ft^2$$

$$\overline{L} = \frac{8.5x10^{-4}\quad ft^2}{\left(\frac{0.750}{12}\ ft\right)} = 0.0136\quad ft$$

$$\left(\frac{1}{d_{eq}}\right)^{1/4} = 1.3\left(0.692\right)\left(\frac{15.6\,ft^2}{14.0\,ft^2}\right)\left(\frac{1}{0.0136}\right)^{1/4} + \frac{6.7\,ft^2}{14.0\,ft^2}\left(\frac{12\ ft}{0.638}\right)^{1/4} = 3.93\,ft^{-1/4}$$

Estimate wall temperature to be 100°F

$$h_c = 0.689\left[\frac{\left(0.074\,\frac{Btu}{hr\ ft^2\,{}^\circ F}\right)^3\left(29.3\,\frac{lb}{ft^3}\right)\left(27.4\,\frac{lb}{ft^3}\right)\left(4.17x10^8\,\frac{ft\ lb_m}{hr^2 lb_f}\right)\left(138.1\,\frac{Btu}{lb}\right)}{\left(0.194\,\frac{lb}{ft\ hr}\right)(105.2-100)^\circ F}\right]^{1/4}(3.93\,ft^{-1/4})$$

$$= 1000\ \ Btu/hr\ \ ft^2\,{}^\circ F$$

This value should be converged as indicated above and will be somewhat smaller than indicated. Even so, it will be substantially higher than the 511 Btu/hr ft^2°F obtained from the unmodified Nusselt equation. However, the effect on the overall coefficient will be quite small (about 5 per cent increase) because the condensing process is only a small part (about 15%) of the total resistance. In view of the other uncertainties in the calculations (and realizing that the water side fouling resistance is both the most uncertain term and the largest single resistance), this difference is probably not enough to worry about.

Now compute the overall heat transfer coefficient based upon the revised values:

Condensing: h_o = 510 Btu/hr ft^2°F, based on total outside heat transfer area,

Fouling, outside: R_{fo} = 0.0003 hr ft^2 °F/Btu, same basis,

Fin resistance: R_{fin} = 7.1 x 10^{-4} hr ft^2 °F/Btu, same basis,

Wall resistance: R_w = 0.00032 hr ft^2 °F/Btu, based on mean wall diameter,

Water: h_i = 1990 Btu/hr ft^2 °F, based on inside area from Fig. 2.19

Fouling, inside: R_{fi} = 0.001 hr ft^2 °F/Btu, same basis

Then,

$$U_0 = \frac{1}{\frac{1}{510} + 0.0003 + 7.1x10^{-4} + 0.00032\left[\frac{0.503}{\pi\left(\frac{0.573}{12}\right)}\right] + \left(\frac{1}{1990} + 0.001\right)\left(\frac{0.503}{0.130}\right)} = 101.4\ \ Btu/hr\ \ ft^2\,{}^\circ F$$

And the corresponding area is

$$A_o = \frac{4.07x10^8\,Btu/hr}{(101.4\ \ Btu/hr\ \ ft^2\,{}^\circ F)(14.2^\circ F)} = 282{,}700\ \ \ ft^2$$

$$L = \frac{282{,}000\ \ \ ft^2}{(14{,}500\ \ tubes)(0.503\ \ \ ft^2/ft)} = 38.8\,ft$$

Say 39 ft. for further calculations.

Shell-Side Pressure Drop Calculation

So far we have established that the heat transfer characteristics and the tube-side pressure drop and velocity are satisfactory. We still need to calculate the shell-side pressure drop. The basic procedure is to calculate the shell-side pressure drop as if the vapor phase flowed un condensed for the entire length (using the Delaware method as given in Chapter 2 and then to correct it by a two-phase multiplier as shown in Fig. 3.25.

1. Basic geometrical data

   Tube outside diameter: d_o = 0.750 in.

   Tube root diameter: d_r = 0.638 in.

   Fin spacing: s = 0.0265 in.

   Fin thickness: Y = 0.011 in.

   Tube layout: Equilateral triangular with 1.5/16 in. = 0.9375 in. pitch

   Shell inside diameter: D_i = 120 in.

   Shell outer tube limit: D_{otl} = 119 in.

   Effective tube length: L = 39 ft.

   Baffle cut: ℓ_c = 58 in.

   Baffle spacing: ℓ_s = 26 in.

   Number of sealing strips per side: N_{ss} = 0

2. Shell-side geometrical parameters

   a. Total number of tubes in the exchanger: N_t = 14,5000

   b. Tube pitch parallel to flow: p_p = 0.814 in.
      Tube pitch normal to flow: p_n = 0.469 in.

   c. Number of tube rows crossed in one crossflow section:

$$N_c = \frac{(120\ \ in.)\left[1-2\left(\frac{58in.}{120in.}\right)\right]}{0.814\ \ in.} = 5 \qquad (2.38)$$

   d. Fraction of total tubes in crossflow:

$$\frac{\ell_c}{D_i} = \frac{58\ \ in.}{120\ \ in.} = 0.48$$

From Fig. 2.28

$F_c = 0.05$

e. Number of effective crossflow rows in each window:

$$N_{cw} = \frac{0.8(58\ \ in.)}{0.814\ \ in.} = 57 \tag{2.40}$$

f. Number of baffles:

$$N_b = \frac{39}{\left(\frac{26}{12}\ ft\right)} - 1 = 18 \tag{2.41}$$

An even number also means that the nozzles are on opposite sides of the shell as required in this case.

g. Crossflow area at centerline:

$$S_m = (26\ in.)\times \left\{120\ in. - 119\ in. + \frac{119\ in. - 0.750\ in.}{0.9375\ in.}\left[(0.9375\ in. - 0.750\ in.) + 2(0.056\ in.)\left(\frac{0.0265\ in.}{0.0265\ in. + 0.011\ in.}\right)\right]\right\}$$

$$S_m = 900\ in.^2$$

h. Fraction of crossflow area available for bypass flow:

$$F_{sbp} = \frac{(120\ in. - 119\ in.)(26\ in.)}{(900\ in.^2)} = 0.029 \tag{2.44}$$

i. Tube-to-baffle leakage area for one baffle:

$$S_{tb} = 0.0184(14{,}500)(1 + 0.05) = 280\ in.^2 \tag{2.46}$$

j. Shell-to-baffle leakage area for one baffle: Assume diametral shell-to-baffle clearance = 0.700 in.

$$S_{sb} = \frac{(120\ in.)(0.700\ in.)}{2}\left[\pi - \cos^{-1}\left(1 - \frac{2(58\ in.)}{120\ in.}\right)\right] = 67.4\ in.^2$$

k. Area for flow through window:

$$S_{wg} = \frac{(120\ in.)^2}{4}\left[\cos^{-1}\left[1-2\left(\frac{58\ in.}{120\ in.}\right)\right]-\left[1-2\left(\frac{58\ in.}{120\ in.}\right)\right]\sin\left\{\cos^{-1}\left[1-2\left(\frac{58\ in.}{120\ in.}\right)\right]\right\}\right] \quad (2.50)$$

$= 5{,}415\ \text{in.}^2$

$$S_{wt} = \frac{14{,}500}{8}(1-0.05)\pi(0.750\ in.)^2 = 3{,}043\ in.^2 \quad (2.51)$$

$S_w = 5{,}415\ \text{in.}^2 - 3{,}043\ \text{in.}^2 = 2{,}372\ \text{in.}^2$ (2.49)

3. Shell-side pressure drop calculation

a. Calculate shell-side Reynolds number:

$$\text{Re}_s = \frac{\left(\frac{0.0638}{12}\ ft\right)\left(2{,}630{,}000\ \frac{lb}{hr}\right)}{\left(0.021\frac{lb}{ft\ hr}\right)\left(\frac{900}{144}\ ft^2\right)} = 1.07\times10^6 \quad (2.54)$$

It is necessary to guess a friction factor for this case as being about 0.2. (See Fig. 2.17)

b. Pressure drop for an ideal crossflow section:

$$\Delta p_{b,i} = \frac{4(0.2)(2{,}630{,}000)^2(5)}{2\left(1.85\frac{lb}{ft^3}\right)\left(4.17\times10^8\ \frac{ftlb_m}{hr^2 lb_f}\right)\left(\frac{900}{144}\ ft^2\right)} = 459\ lb_f\ /\ ft^2 = 3.2\ psia \quad (2.57)$$

The viscosity gradient correction is ignored in condensing flows.

c. Pressure drop for an ideal window section:

$$\Delta p_{w,i} = \frac{(2{,}630{,}000 lb/hr)^2(2+0.6(57))}{2\left(4.17\times10^8\ \frac{ftlb_m}{hr^2 lb_f}\right)\left(\frac{900}{144}\ ft^2\right)\left(\frac{2{,}372}{144}\ ft^2\right)\left(1.85\frac{lb}{ft^2}\right)} = 1{,}576\ lb_f\ /\ ft^2 = 10.9\ psia \quad (2.58)$$

d. Correction factor for baffle leakage:

$$\frac{S_{sb}+S_{tb}}{S_m} = \frac{67.4\ in.^2 + 280\ in.^2}{900\ in.^2} = 0.386$$

$$\frac{S_{sb}}{S_{sb}+S_{tb}} = \frac{67.4\ in.^2}{67.4\ in.^2 + 280\ in.^2} = 0.19$$

From Fig. 2.38

$R_\ell = 0.48$

e. Correction factor for bundle bypass:

$F_{sbp} = 0.029$
$N_{ss} = 0$

From Fig. 2.39

$R_b = 0.90$

f. Total shell-side pressure drop for an all vapor flow:

$$\Delta p_s = [(18-1)(3.2\ psi)(0.90)+18(10.9\ psi)](0.48)+2(3.2\ psi)(0.90)\left(1+\tfrac{57}{5}\right)=215\ psi \qquad (2.60)$$

for an all-vapor flow.

g. Correction for a condensing flow:
From Fig. 3.25, we find that a totally-condensed flow has a pressure drop 0.29 times the pressure drop for the corresponding all vapor-flow, so the estimated pressure drop for the proposed condenser design would be 62.2 psi.

This value is substantially greater than the allowable pressure drop, and some means must be found to reduce this value.

Alternative Designs for This Case

Besides the excessive pressure drop calculated for the conventional E shell design in this case, there are several other concerns. Considering the high velocities, tube vibration is a definite possibility, especially near the inlet, and an annular distributor (vapor belt) should be specified for all E and J shells (including rod baffle designs). For an X shell (which may also be a rod baffle), a single over sized ("bathtub") nozzle should be used running nearly the full length of the top of the shell, or multiple conventional nozzles may be used along the top at some increase in the piping cost.

To deal with shell-side pressure drop itself, a conventionally baffled J shell design (with otherwise the same geometry as the computed example) would probably suffice. A J shell will reduce the shell-side pressure drop by a factor of 6 to 8 compared to an identical E shell. The decrease in shell-side velocity will not affect the calculated (design) value of the condensing coefficient, because no credit is taken for vapor shear enhancement.

Alternatively, an E shell using rod baffle construction would also reduce the pressure drop to within acceptable limits. This pressure drop could be estimated by calculating the longitudinal flow pressure drop through the tube array using an equivalent diameter based upon the outside diameter of the tubes; the same Δp correction factor for condensing flow across tubes, Fig. 3.25 could be used with reasonable accuracy.

Finally, an X shell using same shell diameter and length could be used either with full circle tube supports or rod baffle supports, as long as the vapor was well distributed along the length of the bundle as noted above.

3.4.2. Condenser Design for a Multi-component Mixture: Example Problem

Special Considerations of Multi-component Condensation

The mechanisms in condensing a multi-component mixture were described qualitatively in Chapter 1. The important differences compared to pure component condensation may be summarized as follows:

1. The heavier components preferentially condense first so that the compositions of each phase are changing from point to point.

2. As condensation proceeds, the condensing temperature decreases and the remaining vapor must be sensibly cooled also. The corresponding sensible heat transfer duty must be transferred from the vapor by a sensible heat transfer coefficient which is generally quite low compared to the condensing coefficient.

The multi-component condensation process has been analyzed in fundamental terms by Krishna and Panchal (26), but the use of these methods in design has not yet been completed and in any case requires a computer. A more heuristic procedure was proposed by Silver (27) and put in suitable form for condensers with multiple coolant passes by Bell and Ghaly (28). Variations of this method are in use in various computer-based condenser rating procedures, but even the simplified method is extremely tedious for hand calculations. So in the following section a reduced form of the Silver method is illustrated; this method will be generally suitable for well-behaved cases - ones in which the condensing curve (of temperature vs. heat release) is nearly linear or slightly concave upwards, in which the properties do not change greatly from start to finish, and in which the coolant temperature does not approach the vapor temperature too closely.

Under these conditions, the design integral from (27) can be replaced by the following approximation:

$$A_o = \left[\frac{1}{U'_o} + \frac{Z}{h_{sv}}\right]\frac{Q_T}{MTD} \tag{3.93}$$

The individual terms in Eq. 3.93 are defined as follows:

$$U'_o = \frac{1}{\dfrac{A_o}{h_i A_i} + R_{fi}\dfrac{A_o}{A_i} + \dfrac{\Delta x_w}{k_w}\dfrac{A_o}{A_w} + R_{fin} + R_{fo} + \dfrac{1}{h_o}} \tag{3.94}$$

The condensing heat transfer coefficient, h_o, is calculated from the methods of this section, such as Eq. (3.80).

$$Z = \frac{Q_{sv}}{Q_T} \tag{3.95}$$

where Q_{sv} is the sensible heat duty removed by cooling the vapor and can be calculated approximately by the following equation:

$$Q_{sv} = \overline{C}_{pv}\left[W_{v_{out}} + \tfrac{1}{2}\left(W_{v_{in}} - W_{v\,out}\right)\right]\left(T_{v_{in}} - T_{v_{out}}\right) \tag{3.96}$$

and Q_T is the total heat transferred in the condenser. Then, h_{sv} is the sensible heat transfer coefficient calculated at the half-condensed point and as if only the vapor phase were flowing. Finally, the MTD is calculated from the LMTD and the configuration correction factor (for multiple coolant passes) using the vapor and coolant terminal temperatures.

The application of these equations is illustrated in the following example problem.

Statement of the Problem

A saturated vapor mixture composed of 0.6 mol fraction nC_5 and 0.4 nC_4 at a pressure of 50 psia is to be totally condensed at the rate of 120,000 lb_m/hr on the shell-side of a horizontal E-shell and tube heat exchanger using Wolverine S/T low-finned Trufin with cooling water available at 85 F. Fixed tube sheet and low carbon steel construction will be acceptable. Pressure drop for the vapor side should not exceed 5 psi and for the water 10 psi.

Fouling resistances are 0.001 hrft^2°F/Btu for the water and 0.0005 for the hydrocarbon mixture. Wolverine S/T Trufin, 3/4 in. x 16 BWG, 19 fins/in. of low carbon steel are to be used.

Physical properties:

The physical properties for the pure components are taken from Perry (29) and the properties of the mixture are calculated using the mixing rules in the same source. These values will be assumed constant throughout the problem:

	Liquid	Vapor
Density, lb_m/ft_3	33.4	0.57
Viscosity, lb_m/ft hr	0.375	0.017
Specific heat, Btu/lb_m°F	0.58	0.42
Thermal conductivity, Btu/hr ft°F	0.077	0.0098

The condensing curve can be constructed from vapor pressure and enthalpy data from the same source. This curve is shown in Fig. 3.26.

Water properties are given at 100°F and assumed constant throughout.

Density, lb./ft^3	62.0
Viscosity, lb_m/ft hr	1.63
Specific heat, Btu/lb_mF	1.00
Thermal conductivity, Btu/hr ft°F	0.361

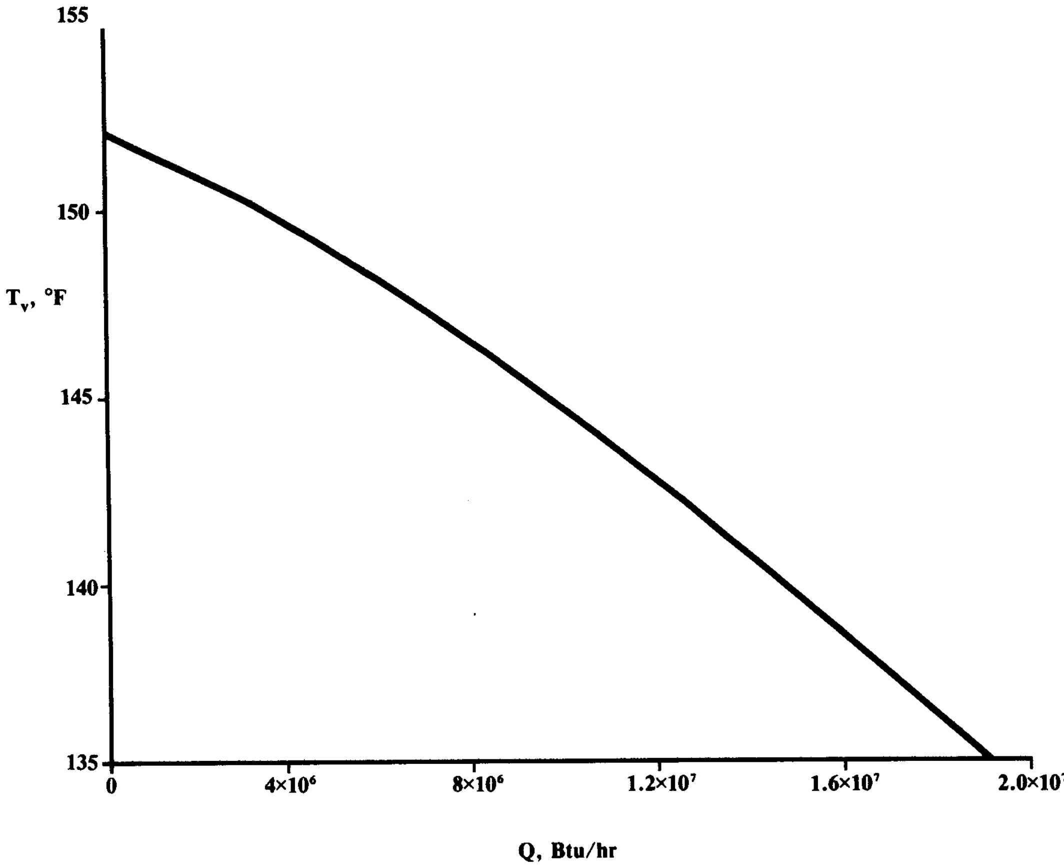

Fig. 3.26 Condensing curve for C_4 - C_5 example problem.

Tube Characteristics (nominal)	
Outside diameter	0.750 in.
Inside diameter	0.495 in.
Root diameter	0.625 in.
Fin height	0.052 in.
Fin thickness	0.011 in.
Outside heat transfer area	0.503 ft^2/ft
Inside heat transfer area	0.1303 ft^2/ft
Outside/inside area ratio	3.86
Inside flow area per tube	0.195 $in.^2$
Thermal conductivity	26 Btu/hr ft°F
Fin resistance	3.1×10^{-4} hr ft^2°F/Btu

Preliminary Estimate of Design

1. Heat duty

   $Q = 1.798 \times 10^7$ Btu/hr (from Fig. 3.26)

2. Mean temperature difference

   Try a design outlet water temperature of 115°F.

$$LMTD = \frac{(152-115)-(136-85)}{\ell n\left(\frac{152-115}{136-85}\right)} = 43.6°F$$

   Assume multiple tube passes

$$P = \frac{115-85}{152-85} = 0.448$$

$$R = \frac{152-136}{115-85} = 0.533$$

   F = 0.955

   MTD = 0.955 (43.6) = 41.6°F

3. Estimated overall heat transfer coefficient, based on outside surface (not to be confused with U'):

$$U_o = \frac{1}{\frac{1}{h_o} + R_{fo} + R_{fin} + \frac{\Delta x_w}{k_w}\frac{A_o}{A_w} + \left(\frac{1}{h_i} + R_{fi}\right)\left(\frac{A_o}{A_i}\right)}$$

$$= \frac{1}{\frac{1}{250} + 5\times10^{-4} + 3.1\times10^{-4} + \frac{0.065}{12(26)}\left[\frac{0.503}{\pi\,\frac{0.506}{12}}\right] + \left(\frac{1}{100} + 0.001\right)\left(\frac{0.503}{0.1303}\right)}$$

$$= 75.5\ Btu/hr\ ft^2\,°F$$

   Note that in order to reflect the extra resistance of the vapor phase, the condensing coefficient was estimated at 250 Btu/hr ft^2°F, rather than the 400 typical of pure component condensation under these conditions.

4. Calculation of area

   $A_o = \dfrac{1.789\times10^7}{41.6(75.5)} = 5696\ ft^2$, estimated total outside finned surface.

5. Estimation of shell dimensions. Again, we may use Fig. 2.26 of chapter 2.

$A'_o = A_oF_1F_2F_3F_4$

$A_o = 5696\ ft^2$

$F_1 = 1.00$ (3/4 in. tube on 15/16 in. triangular pitch)

$F_2 = 1.03$ (Assume - for the moment - 2 tube passes in a 31 in. shell)

$F_3 = 1.00$ (Fixed tube sheet)

$F_4 = 1.00$ (S/T Trufin 19 fins/in.)

$A_o' = 5696\ (1.00)\ (1.03)(1.00)(1.00)$

$A_o' = 5867\ ft^2$

The possibilities (from Fig. 2.26) are:

Shell ID, in.	Effective Tube Length, ft.
35	10
33	11
31	13
29	15
27	17.5
25	20
23 1/4	23.5

Choose the 31 in. ID case for further analysis.

Check water-side velocity: 2 passes:

$$w_{H_2O} = \frac{1.798 \times 10^7\ Btu/hr}{(1.00\ Btu/lb_m{}^\circ F)(30^\circ F)} = 599{,}000\ lb/hr$$

For two tube passes, $N_t = 878$, or 439/pass

So:

$$V_{H_2O} = \frac{(599{,}000\ lb/hr)(144\ in.^2/ft^2)}{439(0.195\ in.^2)(62.0\ lb/ft^3)(3600\ sec/hr)} = 4.51\ \ ft/sec$$

which is quite acceptable for the moment.

Thermal Design

The problem here is to calculate h_o and h_{sv} for the shell side and h_i for the tube-side so that Eq. (3.93) and (3.94) can be evaluated.

1. Calculation of h_o:

   Using Eq. 3.80 and assuming L = 13 ft.

$$W = \frac{120{,}000 lb/hr}{878\ tubes} = 136.7\ lb/hr\ \ per\ tube$$

$$h_o = 0.951\left[\frac{\left(0.077\frac{Btu}{hr\ ft^2 {}^\circ F}\right)^3\left(33.4\frac{lb}{ft^3}\right)\left(32.8\frac{lb}{ft^3}\right)\left(4.17\times 10^8\frac{ft}{hr^2}\right)(13\ ft)}{\left(0.375\frac{lb}{ft\ hr}\right)\left(136.7\frac{lb}{hr}\right)}\right]^{1/3}$$

$= 357\ Btu/ft^2 {}^\circ F$, based on outside finned tube surface area.

2. Calculation of h_{sv}:

The Delaware method will be used based upon the vapor flow rate at the half-condensed point, i.e., 60,000 lb/hr. For a preliminary calculation, a baffle spacing equal to 22.28 in. and a baffle cut of 35 percent of the diameter (10.85 in.) will be assumed. Then the shell-side parameters are: (From Chapter 2)

   a. N_t = 878

   b. p_p = 0.814 in.

      p_n = 0.469 in.

   c. $$N_c = \frac{31\ in.\left[1-2\left(\frac{10.85\ in.}{31\ in.}\right)\right]}{0.814\ in.} = 11$$

   d. $$\frac{\ell_c}{D_i} = \frac{10.85\ in.}{31\ in.} = 0.35$$

      F_c = 40

   e. $$N_{cw} = \frac{0.8(10.85\ in.)}{0.814\ in.} = 11$$

   f. $$N_b = \frac{13\ ft}{\frac{22.80\ in.}{12\ in./ft}} - 1 = 6\ baffles$$

g. $S_m = (22.28 in.)\left\{31\ in. - 30.5\ in. + \left(\frac{30.5\ in. - 0.750\ in.}{0.9375\ in.}\right)\left[(0.9375\ in. - 0.750\ in.) + 2(0.052\ in.)\left(\frac{0.0416\ in.}{0.0416\ in. + 0.011\ in.}\right)\right]\right\}$

= 202 in.^{2}

h. $F_{sbp} = \frac{(31.0\ in. - 30.5\ in.)22.28}{201.9\ in.^2} = 0.055$

i. S_{tb} = 0.0184 (878) (1 + 0.40) = 22.6 in.^{2}

j. Assume diametral clearance of 0.300 in.

$$S_{sb} = \frac{(31\ in.)(0.300\ in.)}{2}\left[\pi - \cos^{-1}\left(1 - \frac{2(10.85\ in.)}{31\ in.}\right)\right] = 8.7\ in.^2$$

k. $S_{wg} = \frac{(31 in.)^2}{4}\left[\cos^{-1}\left(1 - \frac{2(10.85 in.)}{31 in.}\right) - \left(1 - \frac{2(10.85 in.)}{31 in.}\right) \times \sin\left\{\cos^{-1}\left(1 - \frac{2(10.85 in.)}{31 in.}\right)\right\}\right] = 235.4\ \ in.^2$

$$S_{wt} = \frac{878}{8}(1 - 0.40)\pi\ (0.750 in.)^2 = 116.4\ in.^2$$

S_w = 235.4 in.^{2} - 116.4 in.^{2} = 119 in.^{2}

The heat transfer calculation is as follows:

a. $\mathrm{Re}_s = \frac{0.625\ in.(60{,}000\ lb/hr)(12\ in./ft)}{(0.017\ lb/ft\ hr)(201.9\ in.^2)} = 131{,}000$

b. j_s = 0.0039

c. $h_{o,i} = 0.0039\left(0.42\,\frac{Btu}{lb°F}\right)\left[\frac{(60{,}000\ lb/hr)(144\ in.^2/ft^2)}{201.9\ in.^2}\right]\left[\frac{0.0098\ Btu/hr\ ft°F}{(0.42\ Btu/lb\ °F)(0.017 lb/ft\ hr)}\right]$

= 86.6 Btu/hr ft$^{2\circ}$F

d. J_c = 0.85

e. $\frac{S_{sb} + S_{tb}}{S_m} = \frac{8.7\ in.^2 + 22.6\ in.^2}{201.9\ in.^2} = 0.155$

$$\frac{S_{sb}}{S_{sb} + S_{tb}} = \frac{8.7\ in.^2}{8.7\ in.^2 + 22.6\ in.^2} = 0.278$$

$J_\ell = 0.775$

f. $\dfrac{N_{ss}}{N_c} = 0$; $F_{sbp} = 0.055$

$J_b = 0.93$

g. $h_{sv} =$ 86.6(0.85)(0.775)(0.93) Btu/hr ft^2°F

= 53.1 Btu/hr ft^2°F, based on outside tube surface area.

3. Calculation of h_i: use Fig. 2.19.

h_i = 1.04(1140 Btu/hr ft^2°F) = 1190 Btu/hr ft^2°F, based on inside tube surface area.

4. Calculation of U'_o: Refer to Equation (3.94).

$$U'_o = \frac{1}{\left(\frac{1}{1190}+0.001\right)\left(\frac{0.503}{0.1303}\right)+\frac{0.065}{26(12)}\left(\frac{0.503}{0.1303}\right)+3.1\times10^{-4}+5\times10^{-4}+\frac{1}{357}} = 86.8 \text{ Btu/hr ft}^2\,°F$$

5. Calculation of Z:

$$Q_{sv} = 0.42\frac{Btu}{lb°F}\left[0+\frac{1}{2}\left(120{,}000\frac{lb}{hr}-0\right)\right](152°F-136°F) = 403{,}200\ Btu/hr$$

$$Z = \frac{Q_{sv}}{Q_T} = \frac{403{,}200\ Btu/hr}{1.798\times10^7\ Btu/hr} = 0.0224$$

6. Calculation of A_o: Refer to Eq. (3.93).

$$A_o = \left(\frac{1}{86.8}+\frac{0.0224}{53.1}\right)\left(\frac{1.798\times10^7}{41.6}\right) = 5162\ ft^2$$

compared to the estimated 5696 ft^2. The actual area provided by the design rated is:

A_o = (878 tubes)(13 ft)(0.503 ft^2/ft) = 5741 ft^2

which is 10 percent greater than the calculated required area. This is a reasonable safety factor in this kind of problem, and it is suggested that the designer not be tempted to pare the design down.

Shell-side Pressure Drop Calculations

The method applied here is to calculate the pressure drop for vapor-only flowing through the shell-side using the Delaware method, followed by application of the Diehl Unruh two-phase correction factor for total condensation.

1. Delaware calculation of Δp_s, for vapor only.

a. $$\mathrm{Re}_s = \frac{0.625\ in.(120{,}000\ lb/hr)(12\ in./ft)}{(0.017\ lb/ft\ hr)(201.9\ in.^2)} = 262{,}000$$

f = 0.25

b. $$\Delta p_{b,i} = \frac{4(0.25)(120{,}000 lb/hr)^2(11)}{2\left(0.57\frac{lb}{ft^3}\right)\left(4.17\times10^8\frac{ftlb_m}{hr^2lb_f}\right)\left(\frac{201.9\ in^2}{144\ in^2/ft^2}\right)} = 238\ lb_f/ft^2 = 1.65\ psi$$

c. $$\Delta p_{w,i} = \frac{(120{,}000 lb/hr)^2(2+0.6(11))}{2\left(4.17\times10^8\frac{ftlb_m}{hr^2lb_f}\right)\left(\frac{201.9\ in^2}{144in.^2/ft^2}\right)\left(\frac{119\ in^2}{144\ in.^2/ft^2}\right)\left(0.57\frac{lb}{ft^3}\right)} = 225\ lb_f/ft^2 = 1.56\ psi$$

d. R_ℓ = 0.54

e. R_b = 0.80

f. Δp_s = [(6 – 1)(238 lb_f/ft^2)(0.80) + 6(225 lb_f/ft^2)] 0 .54 + 2(238 lb_f/ft^2)(0.80) + (1 + 1/11)

= 2005 lb_f/ft^2 = 13.9 psi

This is the pressure drop for the case in which only vapor flows the entire length of the shell.

g. Correction for a condensing flow. From Fig. 3.25, we find that a totally condensing flow gives about 0.3 times as much pressure drop as that for vapor flowing the entire distance, so we estimate the actual shell side (nozzle-to-nozzle) pressure drop as 0.3 (13.9 psi) = 4.2 psi which is slightly lower than the problem statement. However, it should be noted that these calculations are quite imprecise, and one could gain a small margin of safety (about 8 percent on the pressure drop) by specifying an exchanger with total tube length of 14 feet (with the same baffle spacing), giving an effective tube length of about 13.7 feet. Otherwise, the design seems to meet the requirements quite well.

NOMENCLATURE

A	Heat transfer area. A*, reference area; A_{fin}, fin heat transfer area; A_{eq}, effective area for a finned tube; A_i, A_o, inside and outside tube areas, respectively; A_m, mean wall heat transfer area; A_{root}, root heat transfer area for finned tube; A_T, total heat transfer area in exchanger.	ft^2
a	Heat transfer area per foot of tube.	ft^2/ft
a_{fin}	Area of one side of one fin.	ft^2
C_p,c_p	Specific heat of vapor and coolant, respectively.	Btu/lb°F
$\overline{C}_{p,v}$	Mean specific heat of the vapor phase in a multi-component mixture.	Btu/lb°F
d	Diameter; d_{eq}, equivalent diameter of a finned tube in condensation; d_i,d_o, inside and outside diameters of a tube; d_r, root diameter of a finned tube.	in. or ft.
E	Mueller correction factor for total condensation, Eq. (3.50).	dimensionless
F	Correction factor for the logarithmic mean temperature difference (LMTD) to make it applicable to heat exchangers in which the flow is not entirely countercurrent or cocurrent.	dimensionless
F, F'	Fraction of tubes flooded in a condenser with integral subcooler.	dimensionless
F_1, F_2	Parameters in the Diehl-Koppany flooding velocity correlations, Eq. (3.30 and 3.3 1)	dimensionless
F_1, F_2, F_3, F_4	Correction factors for approximate sizing procedures.	dimensionless
F_c	Chen two-phase convective heat transfer multiplier.	dimensionless
F_{vc}	Parameter in Carpenter-Colburn equation for in-tube condensation, defined by Eq. (3.83).	$lb_m/ft\ hr^2$
f_i	Friction factor for pressure drop inside tubes.	dimensionless
g	Gravitational acceleration.	ft/sec^2
g_c	Gravitational conversion constant.	32.2 $lb_m\ ft/lb_f\ sec^2$
G	Mass velocity (mass flow rate of fluid per unit cross-sectional area for flow). G_v and G_ℓ are superficial mass velocities for vapor and liquid respectively; "superficial" means the values are calculated as if the given fluid were flowing alone, using the entire cross-sectional area. $G_{v,i}$, $G_{v,m}$, and $G_{v,o}$,. are respectively the inlet, mean, and outlet superficial vapor mass velocities for	$lb_m/ft^2\ hr$

	the Carpenter-Colburn correlation for intube condensation.	
H	Fin height.	in. or ft.
h	Film heat transfer coefficient; h_i, h_o, inside and outside coefficients, respectively; h_c, average condensing coefficient; $h_{c,N}$, average coefficient for a vertical bank of N tubes; $\overline{h}_{sv}$, average sensible heat transfer coefficient for the vapor-gas mixture in multicomponent condensation.	Btu/hr ft^2°F
k	Thermal conductivity; k_ℓ, liquid thermal conductivity; k_v, vapor thermal conductivity; k_w, thermal conductivity of tube wall.	Btu/hr ft^2°F
K	Parameter in Mueller total condensation analysis, Eq. (3.51)	dimensionless
L	Tube length.	ft.
$\overline{L}$	Equivalent length of fin.	ft.
LMTD	Logarithmic mean temperature difference.	°F
MTD	True mean temperature difference, F (LMTD).	°F
m	Parameter in fin efficiency equation.	dimensionless
N	Number of tubes in a vertical row.	dimensionless
N_f	Number of fins per unit length.	fins/inch
Nu_i	Nusselt number for heat transfer inside a tube.	dimensionless
p	Pressure	lb_f/in^2absolute
P_{crit}	Critical pressure of a fluid.	lb_f/in^2absolute
P_r	Prandtl number for a fluid.	dimensionless
P_R	Reduced pressure, defined by Eq. (3.10).	dimensionless
Δp	Pressure drop; Δp_{ent}, pressure drop for entrance to a tube; Δ, pressure drop due to friction for flow inside a tube; Δp, pressure drop on the shell side, nozzle to nozzle.	lb_f/ft^2 or lb_f/in^2
Δp_{TPF}	Total pressure effect in a two-phase flow.	lb_f/in^2
$\Delta p_{m,TPF}$	Pressure effect due to momentum changes in two phase flow.	lb_f/in^2
$\left(\frac{dp}{d\ell}\right)_{f,\ell}$	Pressure gradient due to friction for liquid flowing in a conduit.	$(lb_f/in^2)/ft$

$\left(\frac{dp}{d\ell}\right)_{f,TPF}$	Pressure gradient due to friction for a two-phase mixture flowing in a conduit.	$(lb_f/in^2)/ft$
$\left(\frac{dp}{d\ell}\right)_{f,v}$	Pressure gradient due to friction for a vapor flowing in a conduit.	$(lb_f/in^2)/ft$
$\left(\frac{dp}{d\ell}\right)_{g,TPF}$	Pressure gradient due to hydrostatic effect in two-phase flow.	$(lb_f/in^2)/ft$
$\left(\frac{dp}{d\ell}\right)_{m,TPF}$	Pressure gradient due to momentum changes in two-phase flow.	$(lb_f/in^2)/ft$
$\left(\frac{dp}{d\ell}\right)_{T,TPF}$	Total pressure gradient in two-phase flow.	$(lb_f/in^2)/ft$
Q	Heat load; Q_c, condensing heat load; Q_{sc} subcooling heat load; Q_{sv} heat heat load for sensible cooling of a vapor; Q_T total heat load in a condenser.	Btu/hr
R_{fi}, R_{fo}	Fouling resistances, inside and outside surfaces, respectively.	hr ft^2°F/Btu
R_{fin}	Fin resistance to heat transfer.	hr ft^2°F/Btu
R_ℓ, R_v	Liquid and vapor volume fractions, respectively: the actual volume occupied by a given phase, divided by the total volume of the system.	dimensionless
Re_c	Condensate Reynolds number defined by Eq. (3.37).	dimensionless
Re_i	Reynolds number for flow inside a tube.	dimensionless
s	Spacing between fins.	in. or ft.
T	Temperature on the condensing side; T_s, T_s', surface temperatures, actual and calculated, during desuperheating; T_{sat}, saturation temperature of vapor; T_{sc}, temperature of exiting subcooled condensate; T_w, surface temperature in condensing zone.	°F
t	Temperature of coolant; t_1 t_2, inlet and outlet temperatures, respectively; t*, t_o*, mixed mean turnaround and exit temperatures in condensers with subcooling zones; t_{ao}, t_{bo}, t_{do}, t_{eo}, t_{fo}, t_{go}, t_{jo}, t_{ko}, local turnaround temperatures in condensers with subcooling zones.	°F
U	Overall heat transfer coefficient; U*, overall coefficient based on a reference area, A*; U_i, U_o, overall coefficients based upon the inside and outside tube areas, respectively; U_c, U_s, and U_{sc}, overall coefficients for the condensing zone, the sensible heat transfer zone, and the subcooling zone, respectively; U', and U_o', partial overall coefficients, including condensate film wall, fouling, and coolant resistances.	Btu/hr ft^2°F
V_i	In-tube fluid velocity.	ft/sec or ft/hr

V_v'	Superficial incipient flooding velocity of the vapor in a knockback condenser by the Diehl-Koppany correlation.	ft/sec
$\vartheta_\ell, \vartheta_v$	Volume flow rates of liquid and vapor phases, respectively, in a two-phase flow.	ft^3/hr
W	Vapor mass flow rate. $W_{v_{in}}$, $W_{v_{out}}$, inlet and outlet vapor mass flow rates, respectively.	lb/hr
W_t	Condensate weight flow rate per tube defined in Eq. (3.46).	lb/hr
w	Coolant mass flow rate.	lb/hr
x	Distance from tube inlet.	ft.
x	For two-phase vapor-liquid flows, the quality of the flow: the weight fraction of the flow that is vapor.	dimensionless
X_o	Outlet vapor quali!y.	dimensionless
Δx_w	Wall thickness.	in. or ft.
Y	Fin thickness.	in. or ft.
Z	Fraction of total heat duty that is vapor/gas sensible cooling.	dimensionless

GREEK

λ	Latent heat of condensation.	Btu/lb
μ	Viscosity. μ_ℓ, liquid phase viscosity; μ_s, viscosity of fluid evaluated at surface temperature.	lb_m/ft hr
ρ	Density. ρ_ℓ, liquid density; ρ_v, vapor density.	lb_m/ft^3
σ	Surface tension of a liquid.	dyne/cm
Γ	Condensate loading per foot of tube drainage perimeter, defined by Eq. (3.34).	lb_m/ft hr
δ	Thickness of a condensate film.	in. or ft.
θ	Angular orientation of a tube, Fig. 3.6.	degrees
Λ	Modified Baker parameter for two-phase flows in a horizontal tube. $\Lambda = \sqrt{\rho_\ell \rho_v}$	$lb./ft^3$
Φ	Fin efficiency.	dimensionless

Φ_{gtt}^2	Multiplying factor for calculating total shell-side pressure drop in a condenser from vapor phase pressure drop.	dimensionless
$\Phi_{\ell tt}^2$	Martinelli-Nelson two-phase flow friction pressure drop multiplier.	dimensionless
X_{tt}	Martinelli-Nelson two-phase flow parameter defined by Eq. (3.15).	dimensionless
Ψ	Modified Baker parameter for two-phase flows in a horizontal tube. $\Psi = \mu_\ell^{1/3} / \sigma\rho_\ell^{2/3}$	$(ft^{5/3}cm) / (lb_m^{1/3}\ hr^{1/3}dyne)$

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