

SECTION 9

Heat Exchangers

The process engineer is frequently required to analyze heat exchanger designs, specify heat exchanger performance, and determine the feasibility of using heat exchangers in new services. This section is prepared with these specific operations in mind and is not intended as a design manual.

FUNDAMENTALS OF HEAT TRANSFER

The basic definitions and equations used in heat exchanger calculations are reviewed below:

Heat Balances

For no phase change of the hot fluid:

$$Q = 0.27\dot{m}_H C_{pH} (T_{H1} - T_{H2}) \quad \text{Eq 9-1}$$

For isothermal condensing of the hot fluid:

$$Q = 0.27\dot{m}_H \lambda_H \quad \text{Eq 9-2}$$

For no phase change of the cold fluid:

$$Q = 0.27\dot{m}_C C_{pC} (T_{C2} - T_{C1}) \quad \text{Eq 9-3}$$

For isothermal boiling of the cold fluid:

$$Q = 0.27\dot{m}_C \lambda_C \quad \text{Eq 9-4}$$

Basic Heat Transfer Relations

$$Q = UA \text{ (LMTD) (single-pass design)} \quad \text{Eq 9-5a}$$

$$Q = UA \text{ (CMTD) (multi-pass design)} \quad \text{Eq 9-5b}$$

FIG. 9-1
Nomenclature

A = area, m ²	T = temperature, °C
BP = baffle spacing, mm	t = temperature, °C
C = tube count factor	TMTD = True Mean Temperature Difference, °C
C _p = specific heat, kJ/(kg · °C)	U = overall heat transfer coefficient, W/(m ² · °C)
CMTD = Corrected Log Mean Temperature Difference, °C	W = width, mm
D = diameter, mm	WTD = weighted temperature difference, °C
F = LMTD correction factor	X = weight fraction
f = ratio of one value to another	β = liquid expansion coefficient, 1/°C
FD = free diameter, mm	λ = latent heat, kJ/kg
G = mass velocity, kg/(m ² · s)	μ = viscosity, mPa · s
GTTD = Greatest Terminal Temperature Difference, °C	ρ = density, kg/m ³
h = film coefficient, W/(m ² · °C)	
H = height, mm	
k = thermal conductivity, W/(m · °C)	
L = length, mm	
LMTD = Log Mean Temperature Difference, °C	
LTTD = Least Terminal Temperature Difference, °C	
$\dot{m}$ = mass flowrate, kg/hr	
N = number of exchangers	
N _p = number of passes	
p = temperature efficiency	
ΔP = pressure drop, kPa	
P = pressure, kPa (abs)	
PHE = plate and frame heat exchanger	
Q = heat transferred, W	
R = heat capacity rate ratio	
Re = Reynolds number = (1.001 DG)/μ	
RC = tube rows crossed	
r = film resistance (m ² · °C)/W	
SP = number of baffle spaces	

Subscripts

b = boiling
C = cold fluid
c = condensing
f = fouling
H = hot fluid
i = inside
in = inlet
L = liquid
m = mean value
n = n th value
o = outside
out = outlet
2Φ = two-phase
w = wall
v = vapor
1 = first value
2 = second value

SHELL AND TUBE EXCHANGERS

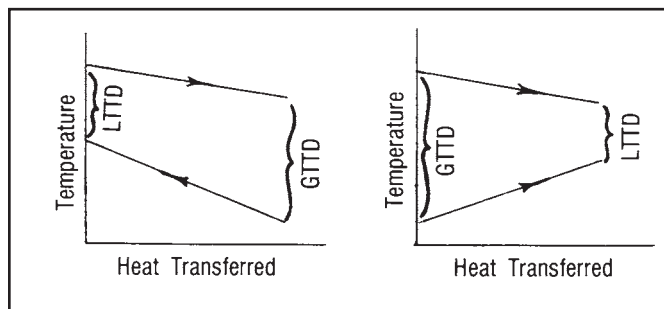
For tubular heat exchangers, the heat transfer area generally referred to is the effective outside bare surface area of the tubes, and the overall heat transfer coefficient must also be based on this area.

The numerous shell styles, baffle types, and tube pass arrangements allow shell and tube exchangers to handle a wide variety of thermal and hydraulic service requirements.

Effective Temperature Difference

In most instances the local temperature difference between the hot stream and the cold stream will not have a constant value throughout a heat exchanger, and so an effective average value must be used in the rate equation. The appropriate average depends on the configuration of the exchanger. For simple

FIG. 9-2
Concurrent Flow and Co-current Flow



countercurrent and co-current exchangers (Fig. 9-2), the Log Mean Temperature Difference (LMTD) applies.

Fig. 9-3 defines LMTD in terms of Greatest Terminal Temperature Difference (GTTD) and Least Terminal Temperature Difference (LTTD), where “terminal” refers to the first or last point of heat exchange in the heat exchanger.

For exchanger configurations with flow passes arranged to be partially countercurrent and partially co-current, it is common practice to calculate the LMTD as though the exchanger were in countercurrent flow, and then to apply a correction factor to obtain the effective temperature difference.

$$\text{CMTD} = (\text{LMTD}) (F) = \text{Corrected Mean Temperature Difference} \quad \text{Eq 9-6}$$

The magnitude of the correction factor, F , depends on the exchanger configuration and the stream temperatures. Values of F are shown in Figs. 9-4, 9-5, 9-6, and 9-7 for most common exchanger arrangements. In general, if the value obtained for F is less than 0.8, it is a signal that the selected exchanger configuration is not suitable, and that one more closely approaching countercurrent flow should be sought.

Heat Exchange with Non-Linear Behavior

The above Corrected Log Mean Temperature Difference (CMTD) implicitly assumes a linear relation between duty and stream temperature change. Some situations for which this assumption is not applicable include process streams which undergo a very large temperature change so that the physical

properties change significantly, multi-component condensing or boiling with non-linear duty vs. temperature curves, and exchangers in which the process stream undergoes both phase change and sensible cooling or heating.

These situations may be handled by dividing the exchanger into zones which may be treated individually with the linear assumption. The overall exchanger performance may be represented in terms of the weighted average performance of the zones in the overall rate equation. The following equations may be taken as the rate equations for the overall exchanger and for the n^{th} zone of the exchanger.

$$Q_{\text{Total}} = U_{\text{wtd}} A_{\text{Total}} (\text{WTD}) \quad \text{Eq 9-7}$$

$$Q_n = U_n A_n (\text{LMTD})_n \quad \text{Eq 9-8}$$

Then the weighted temperature difference may be defined as:

$$(\text{WTD}) = \frac{\sum [U_n A_n (\text{LMTD})_n]}{\sum [U_n A_n]} = \frac{Q_{\text{Total}}}{\sum [Q_n / (\text{LMTD})_n]} \quad \text{Eq 9-9}$$

And the weighted overall heat transfer coefficient becomes:

$$U_{\text{wtd}} = \frac{Q_{\text{Total}}}{A_{\text{Total}} (\text{WTD})} = \frac{\sum [Q_n / (\text{LMTD})_n]}{A_{\text{Total}}} \quad \text{Eq 9-10}$$

In multi-component, two-phase (vapor/liquid) flow regimes undergoing heat transfer, the vapor and liquid composition changes that occur are related to the extent of continuous contact of the two phases. If the vapor phase is maintained in contact with the liquid, the total change in enthalpy (or other properties) that accompanies the composition change is termed “integral.” If the vapor is continuously removed from contact with the liquid as it is formed, the property changes are termed “differential.” An accurate representation of temperature difference and heat transfer in these cases depends on correct consideration of the phase separation that occurs in the heat transfer equipment.

Overall Heat Transfer Coefficient

$$U_o = \frac{1}{\left[\frac{1}{h_o} + \left(\frac{A_o}{A_i} \right) \left(\frac{1}{h_i} \right) + r_w + r_{fo} + \left(\frac{A_o}{A_i} \right) r_{fi} \right]} \quad \text{Eq 9-11}$$

Metal Resistance for Plain Tubes

The metal resistance is calculated by the following equation:

$$r_w = \frac{D_o}{2 \cdot 1000 k_w} \ln \left[\frac{D_o}{D_i} \right] \quad \text{Eq 9-12}$$

Values of the tube metal thermal conductivity are found in Fig. 9-8 for several materials of construction at different metal temperatures.

Fouling Resistances

Fouling resistances depend largely upon the types of fluid being handled, i.e., the amount and type of suspended or dissolved material which may deposit on the tube walls, susceptibility to thermal decomposition, etc., and the velocity and temperature of the streams. Fouling resistance for a particular service is usually selected on the basis of experience with similar streams. Some typical values are given in Fig. 9-9 and in the TEMA Standards.

FIG. 9-3
LMTD Chart

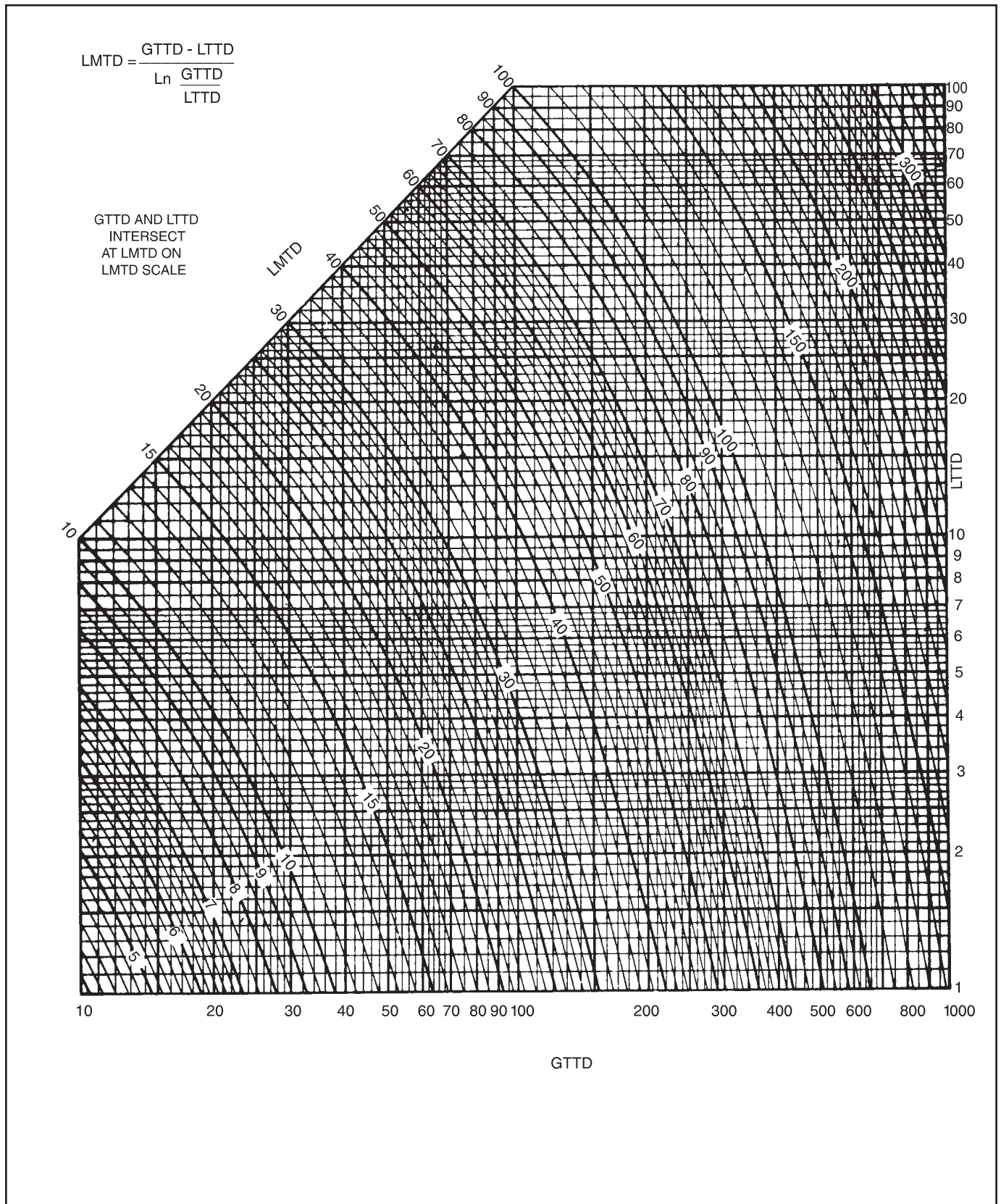


FIG. 9-4
LMTD Correction Factor (1 shell passes; 2 or more tube passes)

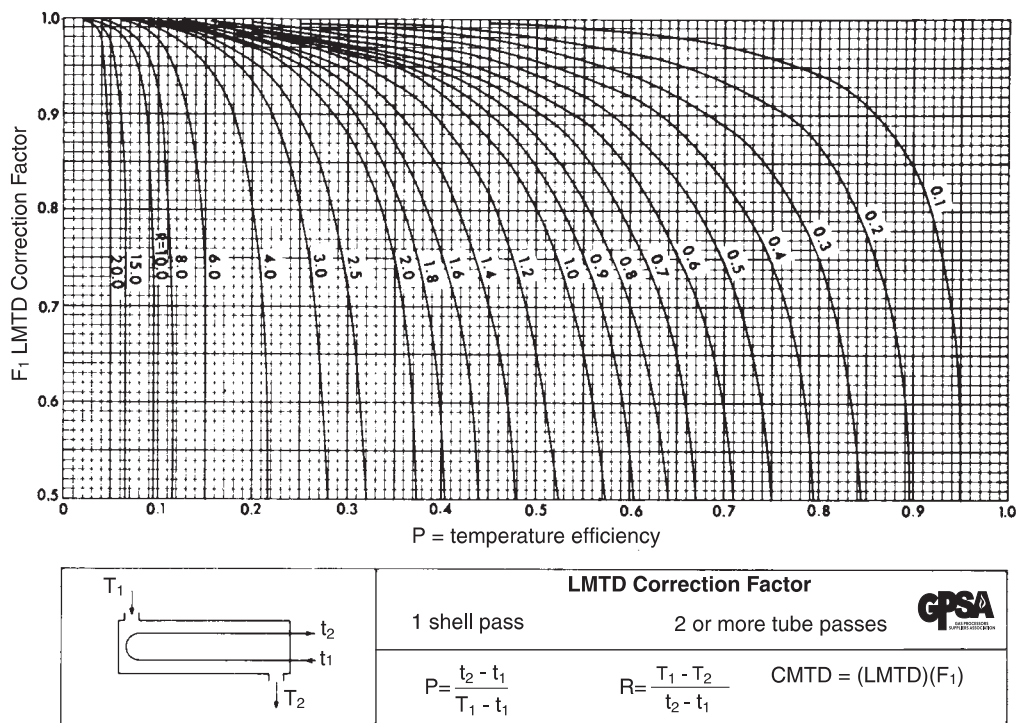


FIG. 9-5
LMTD Correction Factor (2 shell passes; 4 or more tube passes)

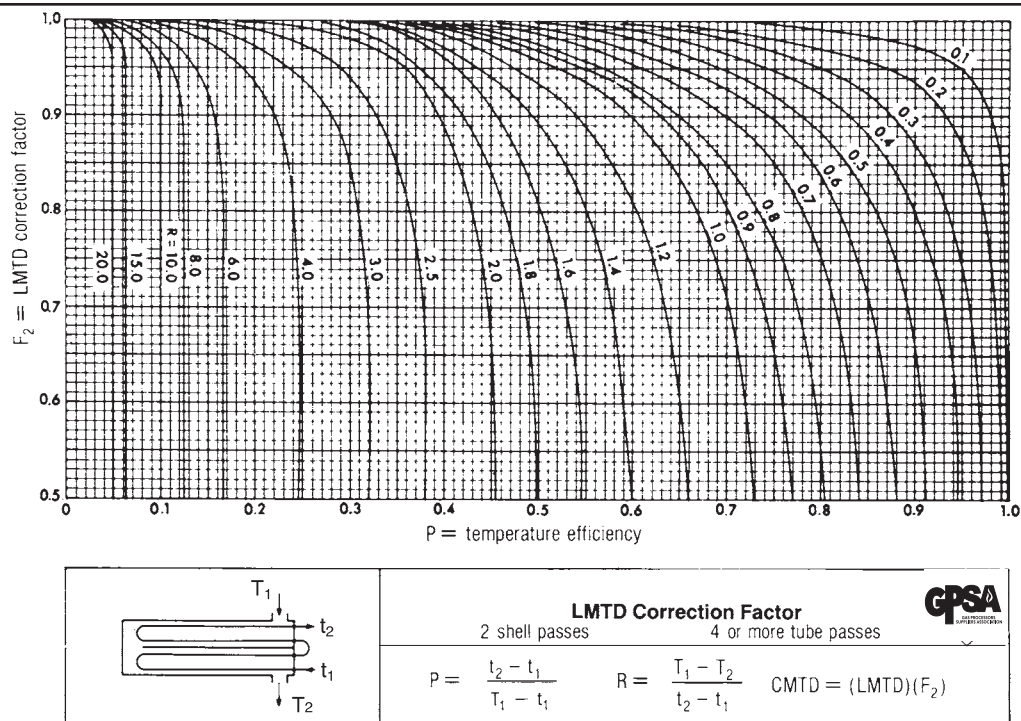


FIG. 9-6
LMTD Correction Factor (3 shell passes; 6 or more tube passes)

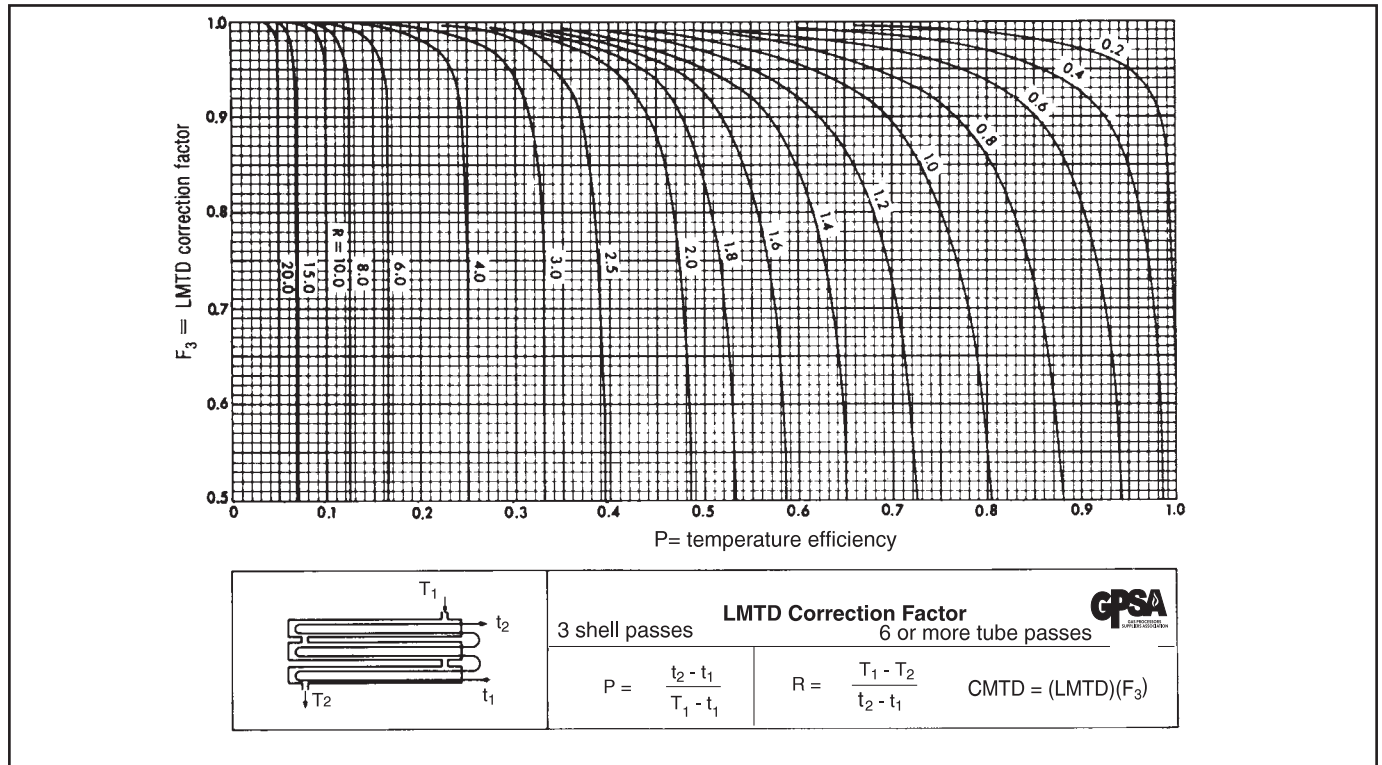
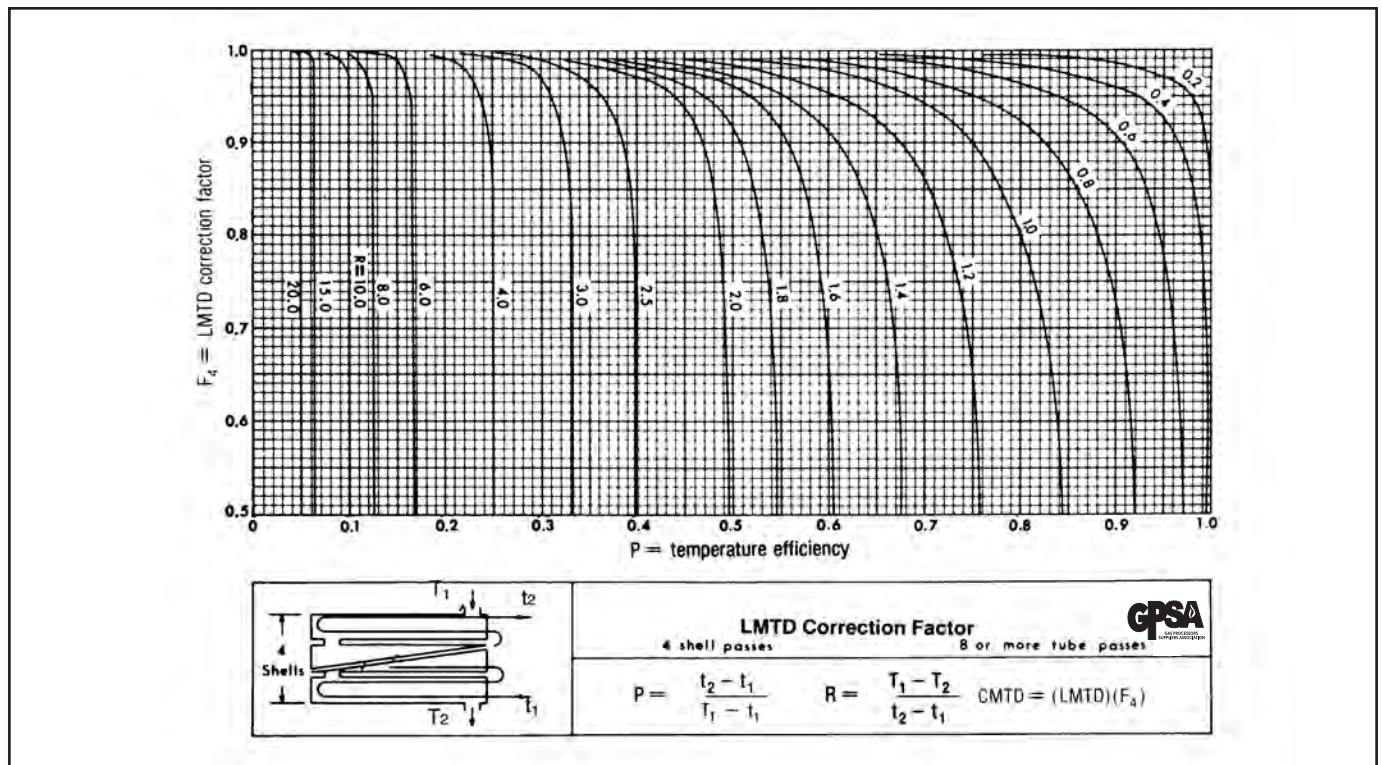


FIG. 9-7
LMTD Correction Factor (4 shell passes; 8 or more tube passes)



Film Resistances

Equations for calculating the film coefficients, h_o , and h_i , for the simpler common geometries, as functions of flow rate and fluid properties, may be found in heat transfer references and in engineering handbooks. Some typical values of film resistances are given in Fig. 9-11. Some common overall heat transfer coefficients are shown in Fig. 9-9.

Film coefficients, film resistances, and overall heat transfer coefficient are related as follows: $h_i = 1/r_i$, $h_o = 1/r_o$, and $U = 1/\Sigma r$ (as in Equation 9-11).

Performance Evaluation With Sensible Heat Transfer

To predict the performance of a particular exchanger in a new service or to compare different designs for a given service, it is useful to understand the effects of changes in the variables on film resistance to heat transfer and pressure drop. If variables (subscripted "1") are used for a reference basis (as those values given in Fig. 9-11 are intended to be) a proration to a new condition (subscripted "2") can be applied based on ratioing the correlation of the variable at the new condition to the reference condition. For film coefficients and pressure drop determinations, Fig. 9-10 summarizes these ratios for the applicable variables. If tube side film resistance and pressure drop at new conditions involving turbulent flow were desired, the variable arrays would be:

$$\frac{h_i}{h_2} = \frac{r_2}{r_1} = \left(\frac{\mu_2}{\mu_1} \right)^{0.47} \left(\frac{k_1}{k_2} \right)^{0.67} \left(\frac{C_{p1}}{C_{p2}} \right)^{0.33} \left(\frac{G_1}{G_2} \right)^{0.8} \left(\frac{D_{i2}}{D_{i1}} \right)^{0.2} \quad \text{Eq 9-13}$$

and,

$$\frac{\Delta P_2}{\Delta P_1} = \left(\frac{\mu_2}{\mu_1} \right)^{0.2} \left(\frac{G_2}{G_1} \right)^{1.8} \left(\frac{\rho_1}{\rho_2} \right) \left(\frac{D_{i1}}{D_{i2}} \right)^{1.2} \left(\frac{N_{p2}}{N_{p1}} \right) \quad \text{Eq 9-14}$$

FIG. 9-8

Typical* Metal Thermal Conductivities, k_w

Material	W/(m • °C)			
	Metal Temperature			
	93°C	204°C	316°C	427°C
Aluminum, 3003 Tempered	180	183	—	—
Carbon Steel	50	48	45	42
Carbon Moly (1/2%) Steel	43	43	42	38
2 1/4% Cr, 1% Mo Steel	36	38	36	35
13 Cr	28	28	28	28
304 Stainless Steel	16	17	19	21
Admiralty	121	137	154	—
Copper	389	388	386	—
90-10 CuNi	52	59	73	85
70-30 CuNi	31	36	43	52
Nickel 200	67	61	57	57
NiFeCrMoCu (Alloy 825)	12	14	16	17
Titanium	21	20	19	19

*Excerpt from TEMA Standards

Shell side film resistance and shell side pressure drop have similar arrays. In Fig. 9-10 all the variables that change in a vertical column apply where the flow regime is appropriate.

The stream types and flow regimes shown in Fig. 9-11 are typical for most fluids encountered in gas plants. These base values of film resistance and pressure drop are used with the relationships given in Fig. 9-10 to evaluate an exchanger design or to project the performance of an exchanger in a new service. This can best be understood by following Example 9-1.

Example 9-1 — The heat exchanger specification sheet, Fig. 9-12, shows the heat transfer requirements and the mechanical design configuration for an oil-to-oil exchanger. Evaluate the indicated performance of this design.

Solution Steps

1. Check the heat balance on the data sheet. (See Fig. 9-12)

$$\begin{aligned} Q_H/Q_C &= \frac{[\dot{m}_H C_{pH} (T_{H1} - T_{H2})]}{[\dot{m}_C C_{pC} (T_{C2} - T_{C1})]} \\ &= \frac{[(215\,784)(2.270) (92 - 38)]}{[(295\,225)(2.51) (51 - 16)]} = 1.0 \end{aligned}$$

2. Calculate the LMTD.

$$\begin{array}{rcl} 92 & \rightarrow & 38 \\ 51 & \leftarrow & 16 \\ \hline 41 & & 22 \end{array}$$

$$\text{LMTD} = \frac{(41 - 22)}{\ln(41/22)} = 30.5^\circ\text{C}$$

FIG. 9-9

Typical Heat Transfer Coefficients, U , and Fouling Resistances, r_f

Service and (r_f)	U	Service and (r_f)	U
Water (0.0004)/ 700 kPa Gas (0.0002)	200-225	Rich (0.0002)/Lean Oil (0.0004)	450-570
2000 kPa Gas (0.0002)	225-285	C ₃ Liq/C ₃ Liq (0.0002)	625-740
5000 kPa Gas (0.0002)	340-400	MEA/MEA (0.0004)	680-740
7000 kPa Gas (0.0002)	450-570	700 kPa Gas/3400 kPa Gas	280-400
Kerosene (0.0002)	450-500	7000 kPa Gas/7000 kPa Gas	340-450
MEA (0.0004)	740-850	7000 kPa Gas/Cond. C ₃ (0.0002)	340-450
Air (0.0004)	110-140	Steam (0.0001) Reboilers	800-900
Water (0.0002)	1000-1140	Hot Oil (0.0004) Reboilers	510-680
Condensing with water (0.0004)/		Heat Transfer Fluid (0.0002) Reboilers	450-625
C ₃ or C ₄ (0.0002)	710-765		
Naphtha (0.0002)	400-450		
Still Overhead (0.0002)	400-450		
Amine (0.0004)	570-625		

U in W/(m² • °C) r_f in (m² • °C)/W

FIG. 9-10
Variables in Exchanger Performance

Variable*	Flow Regime	$r_2 = (f) \cdot (r_1)^\dagger$		$\Delta P_2 = (f) \cdot (\Delta P_1)^\dagger$	
		Shell (f)	Tube (f)	Shell (f)	Tube (f)
Viscosity	Turbulent	$(\mu_2/\mu_1)^{0.27}$	$(\mu_2/\mu_1)^{0.47}$	$(\mu_2/\mu_1)^{0.15}$	$(\mu_2/\mu_1)^{0.2}$
Viscosity – bulk to wall correction	Streamline		$\left[\frac{\mu_1}{\mu_2} \cdot \frac{\mu_{w2}}{\mu_{w1}} \right]^{0.14}$		(μ_2/μ_1)
Thermal conductivity	Turb. or Streamline	$(k_1/k_2)^{0.67}$	$(k_1/k_2)^{0.67}$		
Sp. heat capacity	Turb. or Streamline	$(C_{p1}/C_{p2})^{0.33}$	$(C_{p1}/C_{p2})^{0.33}$		
Mass velocity	Turbulent	$(G_1/G_2)^{0.6}$	$(G_1/G_2)^{0.8}$	$(G_2/G_1)^{1.85}$	$(G_2/G_1)^{1.8}$
(or mass flowrate)	Streamline		$(G_1/G_2)^{0.33}$		(G_2/G_1)
Density	Turb. or Streamline			(ρ_1/ρ_2)	ρ_1/ρ_2
Tube diameter	Turbulent	$(D_{o2}/D_{o1})^{0.4}$	$(D_{i2}/D_{i1})^{0.2}$	$(D_{o1}/D_{o2})^{0.15}$	$(D_{i1}/D_{i2})^{1.2}$
Tube diameter	Streamline		$(D_{i2}/D_{i1})^{0.33}$		$(D_{i1}/D_{i2})^2$
Tube length	Streamline		$(L_2/L_1)^{0.33}$		
Tube passes	Turb. or Streamline				(N_{p2}/N_{p1})
No. baffle spaces	Turb. or Streamline			SP_2/SP_1	
No. tube rows crossed‡				RC_2/RC_1	

* Use consistent units for any one variable in both cases.
† f is the ratio of the new value to the old value for a given variable. The overall f is the product of the individual fs.
‡ Number of rows of tubes exposed to cross flow (as opposed to parallel flow). This number is determined by baffle and bundle geometry

FIG. 9-11
Base Values for Use with Fig. 9-10⁽¹⁾

Fluid		Flow Regime	Local r	Tubeside (One Pass)						
				k	C _p	ρ	ΔP/m	μ ⁽²⁾	G _i	D _i
Water	SI	Turbulent	0.000 16	0.620	4.19	995	1.54	0.764	1294	15.7
HC Oil	SI	Turbulent	0.000 67	0.136	2.09	751	1.37	0.726	903	12.6
Methane	SI	Turbulent	0.0010	0.035	2.26	4.32	3.10	0.0113	152	15.7
HC Oil	SI	Streamline	0.0086	0.124	2.20	822	0.19	(3)	207	21.2
				Shellside						
				k	C _p	ρ	ΔP ⁽⁶⁾	μ ⁽⁵⁾	G _o ⁽⁷⁾	D _o
Water	SI	Turbulent	0.000 088	0.684	4.216	958	1.6	0.282	765.1	15.9
HC Oil	SI	Turbulent	0.000 49	0.132	2.33	750	1.7	0.549	646.4	15.9
Methane	SI	Turbulent	0.000 67	0.064	2.74	3.68	0.62	0.0182	30.2	15.9

- (1) Symbols and units are defined in Fig. 9-1
(2) Bulk average viscosity
(3) 6.62 and Wall viscosity is 27.75
(4) 3.16 kPa for a 5.2 m tube
(5) Average film viscosity
(6) Crossflow ΔP/baffle space/10 tube rows crossed between centroids of cut openings.
(7) Average crossflow mass velocity (see crossflow area calculation in Fig. 9-13)

Since the exchanger is countercurrent flow, the CMTD is the LMTD.

3. Check the required heat transfer coefficient.

$$U = \frac{7\,327\,000}{(420)(30.5)} = 572.0 \text{ W/(m}^2 \cdot ^\circ\text{C)}$$

4. Calculate the tube side pressure drop and resistance to heat transfer with the relationships shown in Fig. 9-10 and the values shown in Fig. 9-11.

The total cross sectional flow area

$$= \frac{[784 \pi (D_i)^2]}{4} = (784)(194.78) = 152\,706 \text{ mm}^2$$

$$G = \frac{(295\,225)}{[(3600)(152\,706/1\,000\,000)]} = 537.0 \text{ kg/(m}^2 \cdot \text{s)}$$

$$\text{Re} = \frac{(1.001)(15.7)(537)}{0.21} = 40\,200$$

Therefore, it is turbulent flow since $\text{Re} > 2000$.

From Fig. 9-10 (see Note †), the ratio of the second to the first resistance is:

$$(r_i)_2 = (f) (r_i)_1$$

Use base values from Fig. 9-11 for $(r_i)_1$ conditions.

$$\begin{aligned} f &= \left(\frac{\mu_2}{\mu_1}\right)^{0.47} \left(\frac{k_1}{k_2}\right)^{0.67} \left(\frac{C_{p1}}{C_{p2}}\right)^{0.33} \left(\frac{G_1}{G_2}\right)^{0.8} \left(\frac{D_{i2}}{D_{i1}}\right)^{0.2} \\ &= \left(\frac{0.21}{0.726}\right)^{0.47} \left(\frac{0.136}{0.135}\right)^{0.67} \left(\frac{2.09}{2.51}\right)^{0.33} \left(\frac{903}{537}\right)^{0.8} \left(\frac{15.7}{12.6}\right)^{0.2} \\ &= 0.840 \end{aligned}$$

Basis: (Inside Area)

$$\begin{aligned} (r_i)_2 &= f(r_i)_1 \text{ and } (r_i)_1 = 0.000\,67 \text{ from Fig. 9-11} \\ &= (0.840)(0.000\,67) \\ &= 0.000\,563 \text{ (m}^2 \cdot ^\circ\text{C)/W} \end{aligned}$$

From Fig. 9-10 (see Note †), the ratio of the second to the first pressure drop is:

$$(\Delta P_i)_2 = (f) (\Delta P_i)_1$$

Use base values from Fig. 9-11 for $(\Delta P_i)_1$ conditions.

$$\begin{aligned} f &= \left(\frac{\mu_2}{\mu_1}\right)^{0.2} \left(\frac{G_2}{G_1}\right)^{1.8} \left(\frac{\rho_1}{\rho_2}\right) \left(\frac{D_{i1}}{D_{i2}}\right)^{1.2} \left(\frac{N_{p2}}{N_{p1}}\right) \\ &= \left(\frac{0.21}{0.726}\right)^{0.2} \left(\frac{537}{903}\right)^{1.8} \left(\frac{751}{614}\right) \left(\frac{12.6}{15.7}\right)^{1.2} \left(\frac{1}{1}\right) \\ &= 0.285 \end{aligned}$$

$$(\Delta P_i)_2 = (f) (\Delta P_i)_1 = (0.285)(1.37) = 0.390 \text{ kPa/m}$$

For a 9.15 m tube length the total $9.15(0.390) = 3.569 \text{ kPa}$

5. Calculate the shell side pressure drop and resistance to heat transfer with the relationships shown in Fig. 9-10, the values shown in Fig. 9-11, and the data shown in Fig. 9-13.

$$G = \frac{(215\,784)}{(3600)(101\,283/1\,000\,000)} = 591.8 \text{ kg/(m}^2 \cdot \text{s)}$$

From Fig. 9-10 (see Note †), the ratio of the new to the old resistance is:

$$(r_o)_2 = (f) (r_o)_1$$

Use base values from Fig. 9-11 for $(r_o)_1$ conditions.

$$\begin{aligned} f &= \left(\frac{\mu_2}{\mu_1}\right)^{0.27} \left(\frac{k_1}{k_2}\right)^{0.67} \left(\frac{C_{p1}}{C_{p2}}\right)^{0.33} \left(\frac{G_1}{G_2}\right)^{0.6} \left(\frac{D_{o2}}{D_{o1}}\right)^{0.4} \\ &= \left(\frac{0.34}{0.549}\right)^{0.27} \left(\frac{0.132}{0.133}\right)^{0.67} \left(\frac{2.33}{2.27}\right)^{0.33} \left(\frac{646.4}{591.8}\right)^{0.6} \left(\frac{19.05}{15.7}\right)^{0.4} \\ &= 1.000 \end{aligned}$$

$$(r_o)_2 = (f) (r_o)_1 = (0.998)(0.000\,49) = 0.000\,49 \text{ (m}^2 \cdot ^\circ\text{C)/W}$$

From Fig. 9-10 (see Note †), the ratio of the new to the old pressure drop is:

$$(\Delta P_o)_2 = (f) (\Delta P_o)_1$$

Use base values from Fig. 9-11 for $(\Delta P_o)_1$ conditions. Obtain tube rows crossed between baffle window centroids from Fig. 9-13.

$$\text{RC}_2 = 23 \text{ (RC}_1 = 10 \text{ per note on Fig. 9-11)}$$

Obtain the number of crossflow spaces, which is one more than the number of baffles, from Fig. 9-12.

$\text{SP}_2 = 19$ [$\text{SP}_1 = 1$ per note on Fig. 9-11 since $(\Delta P_o)_1$ is for one baffle space.]

$$\begin{aligned} f &= \left(\frac{\mu_2}{\mu_1}\right)^{0.15} \left(\frac{G_2}{G_1}\right)^{1.85} \left(\frac{\rho_1}{\rho_2}\right) \left(\frac{D_{o1}}{D_{o2}}\right)^{0.15} \left(\frac{\text{SP}_2}{\text{SP}_1}\right) \left(\frac{\text{RC}_2}{\text{RC}_1}\right) \\ &= \left(\frac{0.34}{0.549}\right)^{0.15} \left(\frac{591.8}{646.4}\right)^{1.85} \left(\frac{750}{660}\right) \left(\frac{15.7}{19.05}\right)^{0.15} \left(\frac{19}{1}\right) \left(\frac{23}{10}\right) \\ &= 38.0 \end{aligned}$$

$$(\Delta P_o)_2 = (f) (\Delta P_o)_1 = (38.0)(1.7) = 64.6 \text{ kPa}$$

6. Calculate the tube metal resistance.

$$D_o = 19.05 \text{ mm}$$

$$D_i = 15.75 \text{ mm}$$

$$k_w = 50 \text{ W/(m} \cdot ^\circ\text{C)} \text{ from Fig. 9-8.}$$

$$\begin{aligned} r_w &= \frac{D_o}{2 \cdot 1000 k_w} \ln \left[\frac{D_o}{D_i} \right] \\ &= 0.000\,036 \text{ (m}^2 \cdot ^\circ\text{C)/W} \end{aligned}$$

7. Calculate the overall heat transfer coefficient.

$$\begin{aligned} \Sigma r &= r_i \left(\frac{A_o}{A_i} \right) + r_o + r_w + r_{fo} + r_{fi} \left(\frac{A_o}{A_i} \right) \\ &= 0.000\,563 \left(\frac{0.0182}{0.0151} \right) \\ &\quad + 0.000\,49 + 0.000\,036 + 0.000\,35 \\ &\quad + 0.000\,2 \left(\frac{0.0182}{0.0151} \right) = 0.0018 \end{aligned}$$

$$U = \frac{1}{\Sigma r} = \frac{1}{0.001\,78} = 555.6 \text{ W/(m}^2 \cdot ^\circ\text{C)}$$

FIG. 9-12
Shell and Tube Heat Exchanger Specification Sheet

1					Job No.
2	Customer				Reference No.
3	Address				Proposal No.
4	Plant Location				Date Rev.
5	Service of Unit Oil to Oil Exchanger				Item No. E-4
6	Size 30-360	Type NEN	(Hor)	Connected In Parallel	Series
7	Surf./Unit (Gross/Eff.)	m ² Shells/Unit		Surf/Shell (Gross/Eff.)	m ²
8	PERFORMANCE OF ONE UNIT				
9	Fluid Allocation		Shell Side	Tube Side	
10	Fluid Name		Lean Oil	Rich Oil	
11	Fluid Quantity, Total		kg/h	295 225	
12	Vapor (In/Out)				
13	Liquid		215 784	215 784	295 225
14	Steam				
15	Water				
16	Noncondensable				
17	Temperature (In/Out)		°C	92	38
18	Density		kg/m ³	660 @	64.8°C
19	Viscosity, Liquid		mPa · s	0.34 @	64.8°C
20	Molecular Weight, Vapor				
21	Molecular Weight, Noncondensable				
22	Specific Heat (AVG.)		kJ/(kg · °C)	2270 @	64.8°C
23	Thermal Conductivity		J/(s · m ² · °C/m)	0.133 @	64.8°C
24	Latent Heat		kJ/(kg @ C)		
25	Inlet Pressure		kPa (abs)	860	3 100
26	Velocity		m/s		
27	Pressure Drop, Allow./Calc.		kPa	85	/
28	Fouling Resistance (Min.)/Calc.		m ² · °C/W	0.000 35 /	0.000 18 /
29	Heat Exchanged 7.327 M/W: MTD (Corrected)				°C
30	Transfer Rate, Service Clean				W/(m ² · °C)
31	CONSTRUCTION OF ONE SHELL				Sketch (Bundle/Nozzle Orientation)
32			Shell Side	Tube Side	
33	Design/Test Pressure kPa (ga)		1 300/	3500/	
34	Design Temperature °C		340	340	
35	No. Passes per Shell		1	1	
36	Corrosion Allowance mm		2	2	
37	Connections Size & Rating	In			
38		Out			
39		Intermediate			
40	Tube No. 784	OD 18 mm; thk 1.651 mm.;	Length 9 m;	Pitch 2.4 m	↔ 30 ↗ 60 ↘ 90 ↙ 45
41	Tube Type		Material steel		
42	Shell Steel	ID OD	Shell Cover	— (Integ.) (Remov.)	
43	Channel or Bonnet	steel Integ.	Channel Cover	steel	
44	Tubesheet — Stationary	steel Integ.	Tubesheet-Floating	—	
45	Floating Head Cover	—	Impingement Protection		
46	Baffles-Cross	Type	% Cut (Diam/Area)	Spacing: c/c	Inlet mm
47	Baffles-Long		Seal Type		
48	Supports-Tube	U-Bend	Type		
49	Bypass Seal Arrangement	Spacers	Tube-Tubesheet Joint		
50	Expansion Joint	NONE	Type		
51	pv ² -Inlet Nozzle	Bundle Entrance		Bundle Exit	
52	Gaskets-Shell Side	Tube Side			
53	-Floating Head				
54	Code Requirements	ASME	TEMA Class	C	
55	Weight/Shell	Filled with Water	Bundle	kg	
56	Remarks				

8. Compare the required heat transfer coefficient calculated in step 3 to the value calculated in step 7. (Available $U = 561.8$; required $U = 570.4$) The available value is 1.6% less than the required value, and the calculated pressure drops are less than the pressure drops allowed in Fig. 9-12. Therefore, by these calculations, the unit will perform adequately.

CONDENSERS

The purpose of a condenser is to change a fluid stream from the vapor state to the liquid state by removing the heat of vaporization. The fluid stream may be a pure component or a mixture of components. Condensation may occur on the shell side or the tube side of an exchanger oriented vertically or horizontally.

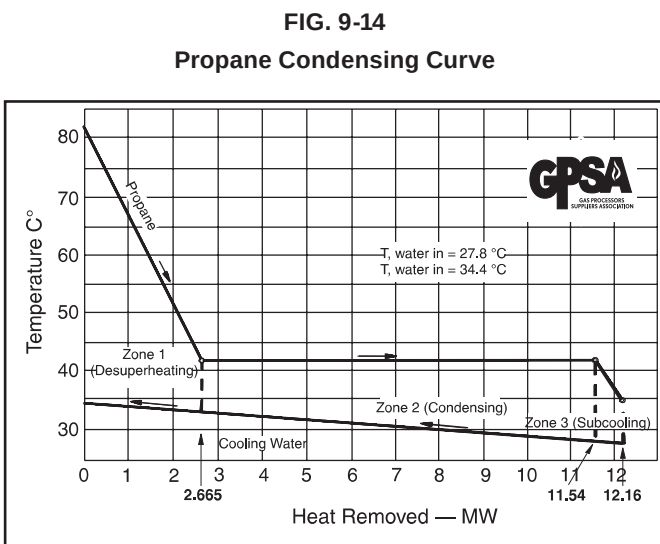
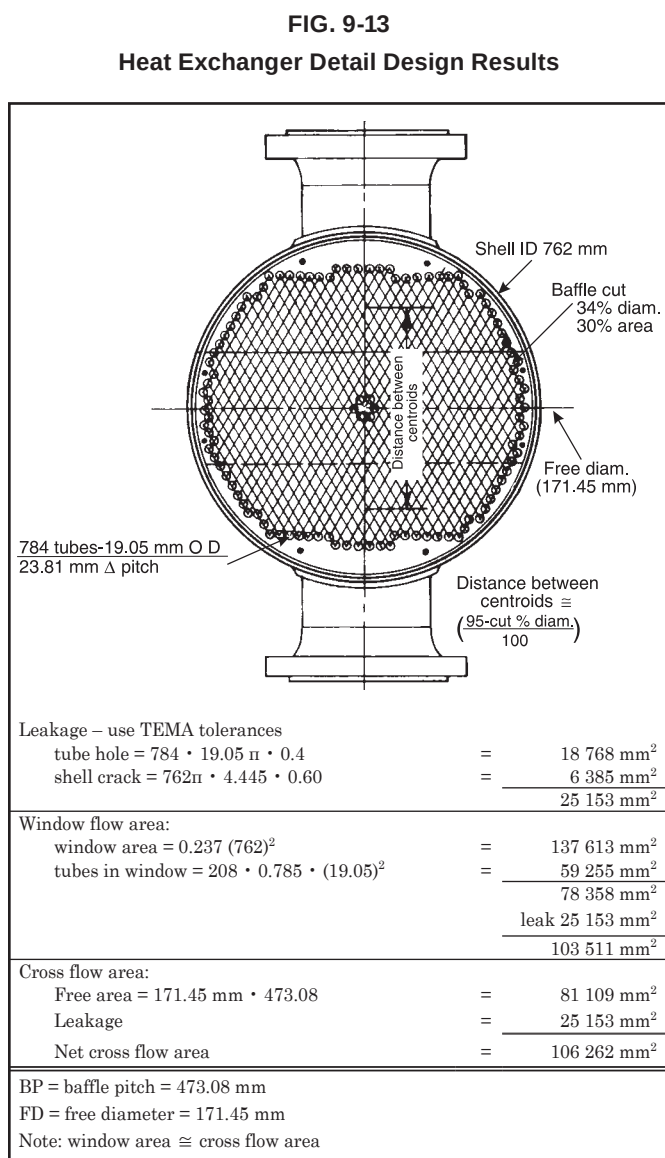
Condensing the overhead vapors of a distillation column is an example of condensing a mixed vapor stream. A vertical exchanger flanged directly to the top of the column might be used. The condensed liquid drains back into the column countercurrent to the vapor entering the condenser. The major concerns in designing this type exchanger are keeping the vapor velocity

sufficiently low to prevent flooding the exchanger and evaluating an appropriate temperature profile at the condensing surface to determine an effective temperature difference. The technical literature addresses criteria for flooding determination¹ and special flow characteristics of falling liquid films. A useful estimate for determining an effective temperature difference can be made by assuming an isothermal condensate film at the saturation temperature of the last condensate formed. If the condensing temperature range exceeds 5°C, consulting a specialist is recommended for a more rigorous calculation procedure.

The condensing of a pure component occurs at a constant temperature equal to the saturation temperature of the incoming vapor stream. Frequently a vapor enters a condenser superheated and must have the sensible heat removed from the vapor before condensation can occur. If the condensing surface temperature is greater than the incoming vapor saturation temperature, the superheat in the vapor is transferred to the cold surface by a sensible heat transfer mechanism ("drywall" condition). If the condensing surface temperature is less than the saturation temperature of the incoming vapor, a condensate film will be formed on the cold surface. The sensible heat is removed from the vapor at the condensate-vapor interface by vaporizing (flashing) condensate so that the heat of vaporization is equal to the sensible heat removed from the vapor. Under this "wet wall" condition, the effective temperature of the vapor is the saturation temperature, and the effective heat transfer mechanism is condensation. The determination of the point in the desuperheating zone of a condenser where "drywall" conditions cease and "wet wall" conditions begin is a trial and error procedure. A method frequently employed to give a safe approximation of the required surface is to use the condensing coefficient and the CMTD based on the vapor saturation temperature to calculate the surface required for both the desuperheating zone and the condensing zone.

The following Example 9-2 will illustrate the use of the heat release curve to calculate the surface required and the LMTD for each zone in a condenser for a pure component application.

Example 9-2 — A propane refrigerant condenser is required to condense the vapor stream using the heat release curve as shown in Fig. 9-14. This stream enters the condenser superheated and leaves the condenser as a subcooled liquid. Assume that a single-tube pass, single-shell pass, counterflow exchanger is



used so that LMTD correction factors do not apply. Note that the propane is on the shell side. The overall heat transfer coefficients for each zone are as follows:

Desuperheating: (82°C to 42°C)

$$U_v = 396.6 \text{ W/(m}^2 \cdot ^\circ\text{C)}$$

$$h_v = 630.4 \text{ W/(m}^2 \cdot ^\circ\text{C)}$$

Condensing: (42°C to 42°C)

$$U_c = 794.4 \text{ W/(m}^2 \cdot ^\circ\text{C)}$$

Subcooling: (42°C to 35°C)

$$U_L = 649.7 \text{ W/(m}^2 \cdot ^\circ\text{C)}$$

Solution Steps

1. Calculate the surface temperature (outside wall) on the vapor side at the refrigerant stream inlet using the following equation:²

$$T_{wo} = T_v - [U_v (T_v - T_c)/h_v]$$

2. If the surface temperature calculated in step 1 is greater than the vapor saturation temperature, calculate the amount of desuperheating that will be done by a sensible heat transfer mechanism. If the surface temperature is less than the vapor saturation temperature, assume that the desuperheating duty will be done by a condensing heat transfer mechanism.

3. Obtain the duty for the appropriate temperature ranges from Fig. 9-14.

4. Solve the equation $Q = UA$ (LMTD) for the required surface area in each zone. The sum of these areas is the surface required for the exchanger.

$$T_{wo} = 82 - \left[\frac{396.6 (82 - 34)}{630.4} \right] = 51.8^\circ\text{C}$$

The surface temperature at the vapor inlet is greater than the saturation temperature, therefore, “drywall” desuperheating will take place initially. By trial and error, calculate the duty required when the assumed bulk vapor temperature results in a surface temperature less than the saturation temperature, thereby marking the transition from “drywall” to “wet wall” desuperheating. Assume the vapor bulk temperature is 56°C.

$$Q_v = 1\,980\,000 \text{ W from Fig. 9-14.}$$

$$\begin{aligned} Q_1 &= \frac{(82 - 56)}{(82 - 42)} (2\,650\,000) \\ &= 1\,730\,000 \text{ W (Heat removed from vapor between 82°C and 56°C)} \end{aligned}$$

$$T_c = 34 - \left[\frac{1\,730\,000}{12\,150\,000} (34 - 28) \right] = 33.15^\circ\text{C}$$

(T_c is the water temperature at $Q_v - Q_1$, see Fig. 9-14)

$$T_{wo} = 56 - \left[\frac{396.6}{630.4} (56 - 33.15) \right] = 41.6^\circ\text{C}$$

Include the remainder of the desuperheating duty in the condensing zone. (Zone 2.)

ZONE 1

$$Q = Q_1 = 1\,730\,000 \text{ W from above.}$$

$$\begin{array}{ccc} 82 & \rightarrow & 56.0 \\ 34 & \leftarrow & 33.15 \\ \hline 48 & & 22.8 \end{array}$$

$$\text{LMTD} = \frac{48 - 22.8}{\ln(48/22.8)} = 33.8^\circ\text{C}$$

$$A = \frac{1\,730\,000}{(396.6)(33.8)} = 129 \text{ m}^2$$

ZONE 2

$$\begin{aligned} Q &= Q_v - Q_1 + Q_c \text{ (} Q_c \text{ from Fig. 9-14)} \\ &= 2\,650\,000 - 1\,730\,000 + 8\,880\,000 \\ &= 9\,800\,000 \text{ W} \end{aligned}$$

$$T_c = 34 - \frac{(1\,730\,000 + 9\,800\,000)}{12\,150\,000} (34 - 28) = 28.3^\circ\text{C}$$

(NOTE: You do **not** use 56°C as the inlet temperature to this zone.)

$$\begin{array}{ccc} 42.0 & \rightarrow & 42.0 \\ 33.15 & \leftarrow & 28.3 \\ \hline 8.9 & & 13.7 \end{array}$$

$$\text{LMTD} = \frac{13.7 - 8.9}{\ln(13.7/8.9)} = 11.1^\circ\text{C}$$

$$A = \frac{9\,800\,000}{(794.4)(11.1)} = 1111 \text{ m}^2$$

ZONE 3

$$Q = Q_L = 621\,000 \text{ W from Fig. 9-14}$$

$$\begin{array}{ccc} 42.0 & \rightarrow & 35 \\ 28.3 & \leftarrow & 28 \\ \hline 13.7 & & 7 \end{array}$$

$$\text{LMTD} = \frac{13.7 - 7}{\ln(13.7/7)} = 10.0^\circ\text{C}$$

$$A = \frac{621\,000}{(649.7)(10.0)} = 95 \text{ m}^2$$

Total Area

$$\Sigma A = 129 + 1111 + 95 = 1335 \text{ m}^2$$

The condensing of a vapor mixture requires additional considerations to those outlined above for pure components. If the mixture condenses over a narrow temperature range, the pure component analysis is applicable. If, on the other hand, the mixture condenses over a wide temperature range, the problem is complicated by the relationship between heat and mass transfer rates. This is also true in the case of condensing vapors in the presence of noncondensables.

The calculation of the condensing coefficient involves physical and thermodynamic properties of the condensing fluid, the two-phase flow regime involved, and the heat exchanger type, geometry, and orientation. The detailed design of such a condenser should be left to a specialist.

REBOILERS AND VAPORIZERS

The “Pool Boiling Curve”

Boiling, as applied to reboilers and vaporizers, can be a complicated relationship of heat and mass transfer. The simplest form is pool boiling where hot fluid inside a tube causes vapor generation on the outside surface of the tube from a pool of liquid. The heat exchange capability of a fluid in pool boiling is determined by empirically correlated data and represented as a “Pool Boiling Curve” that is specific for a composition and a pressure. Fig. 9-15 is a typical representation of single tube data. The shape and regions of the curve in Fig. 9-15 are characteristic of pure component fluids and most mixtures dominated by a single component. Mixtures that boil over a wide temperature range or are being vaporized near their critical point may not fit the characteristic shape or regions. With appropriate curve data and heating medium temperature level, a well designed ‘pool’ reboiler will have heat fluxes in the region analogous to the B-C-D region of Fig. 9-15. Exchangers designed in the E-F-G regions (high surface temperature) may foul faster from thermally activated chemical reactions than would a wet wall design. In the flux inversion region (D-E) erratic, if not reversed, control behavior will occur. Fig. 9-16 summarizes some typical heat flux ranges for some common fluids.

FIG. 9-15
A Typical Pool Boiling Curve

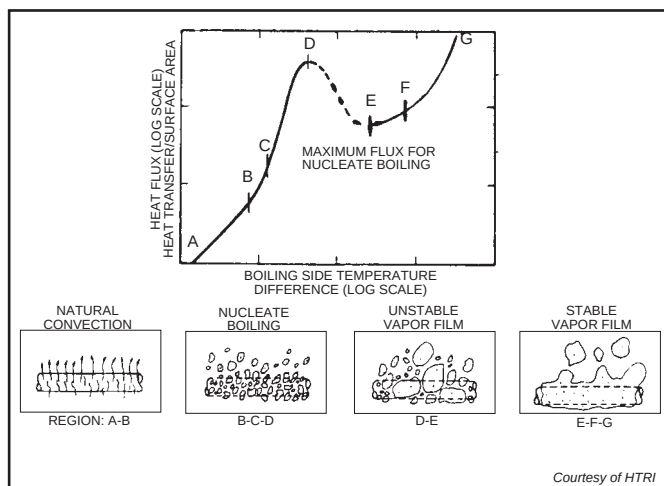


FIG. 9-16

Typical Overall Boiling Heat Flux Ranges

Heat Medium	Boiling Fluid	Heat Flux Range, W/m ²
Hot Oil	C ₁ -C ₂ HC	22 000 - 25 000
	C ₃ -C ₅ HC	20 000 - 44 000
	Rich Oil	8 000 - 11 000
	Amines	11 000 - 17 000
HC Gas	C ₁ -C ₂ HC	6 000 - 14 000
Steam	C ₁ -C ₂ HC	25 000 - 41 000
	C ₃ -C ₅ HC	31 000 - 47 000
	Rich Oil	13 000 - 19 000
	Amines	15 000 - 21 000

Effective Temperature Difference

Boiling, like condensing, may not occur at a constant heat transfer coefficient. The basic definition of Log Mean Temperature Difference may not apply. The effective temperature difference, often called the True Mean Temperature Difference (TMTD), must be determined based on installation and fluid conditions at the reboiler. Elevation of a fluid’s bubble point by static head being added to a column’s sump pressure means a subcooled liquid must be heated to a bubble point higher than the bottom tray liquid temperature. With countercurrent or co-current flow arrangements, an incrementally evaluated Weighted Temperature Difference (WTD) is appropriate. However, in crossflow and pool boiling, a different analysis must apply. In pool boiling, a temperature rise is not readily predictable along a particular geometric flowpath. For design purposes, the TMTD is often taken as if the pool were isothermal at the vapor outlet temperature.

Hydraulic Effects

When the geometric flowpath of a boiling fluid is well defined (all boiling except pool boiling), the effects of liquid and vapor velocities are part of a design or operating analysis. Liquid and vapor co-exist in ‘regimes’ as illustrated in Fig. 9-17. Typically, these regimes progress to termination between Slug Flow and Mist-Annular Flow in a reboiler. In these regimes the heat transfer coefficient has two important contributing parts, convective boiling and nucleate boiling. When liquid is recirculated to a reboiler, the heat transfer coefficient is maximized and such limiting conditions as Mist Flow, Vapor Film Boiling, and two-phase momentum transfer instability may be avoided. The latter form of instability occurs when liquid feed

FIG. 9-17

Two-Phase Flow Regimes in Vertical Tubes

