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:

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

For isothermal condensing of the hot fluid:

\[ Q = 0.27 \dot{m}_H \lambda_H \]  
Eq 9-2

For no phase change of the cold fluid:

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

For isothermal boiling of the cold fluid:

\[ Q = 0.27 \dot{m}_C \lambda_C \]  
Eq 9-4

**Basic Heat Transfer Relations**

\[ Q = UA (LMTD) \]  (single-pass design)  
Eq 9-5a

\[ Q = UA (CMTD) \]  (multi-pass design)  
Eq 9-5b

---

**Nomenclature**

- \( A \): area, m²
- \( BP \): baffle spacing, mm
- \( C \): tube count factor
- \( C_p \): specific heat, kJ/(kg °C)
- \( \text{CMTD} \): Corrected Log Mean Temperature Difference, °C
- \( \text{D} \): diameter, mm
- \( F \): LMTD correction factor
- \( f \): ratio of one value to another
- \( \text{FD} \): free diameter, mm
- \( G \): mass velocity, kg/(m² s)
- \( \text{GTTD} \): Greatest Terminal Temperature Difference, °C
- \( h \): film coefficient, W/(m² °C)
- \( H \): height, mm
- \( k \): thermal conductivity, W/(m °C)
- \( L \): length, mm
- \( \text{LMTD} \): Log Mean Temperature Difference, °C
- \( \text{LTTD} \): Least Terminal Temperature Difference, °C
- \( \dot{m} \): mass flow rate, kg/hr
- \( N \): number of exchangers
- \( N_p \): number of passes
- \( p \): temperature efficiency
- \( \Delta P \): pressure drop, kPa
- \( P \): pressure, kPa (abs)
- \( \text{PHE} \): plate and frame heat exchanger
- \( Q \): heat transferred, W
- \( R \): heat capacity rate ratio
- \( \text{Re} \): Reynolds number = \((1.001 DG)/\mu \)
- \( \text{RC} \): tube rows crossed
- \( r \): film resistance (m² °C)/W
- \( \text{SP} \): number of baffle spaces
- \( T \): temperature, °C
- \( t \): temperature, °C
- \( \text{TMTD} \): True Mean Temperature Difference, °C
- \( U \): overall heat transfer coefficient, W/(m² °C)
- \( W \): width, mm
- \( \text{WTD} \): weighted temperature difference, °C
- \( X \): weight fraction
- \( \lambda \): latent heat, kJ/kg
- \( \mu \): viscosity, mPa s
- \( \rho \): density, kg/m³

**Subscripts**

- \( b \): boiling
- \( C \): cold fluid
- \( c \): condensing
- \( f \): fouling
- \( H \): hot fluid
- \( i \): inside
- \( in \): inlet
- \( \text{L} \): liquid
- \( m \): mean value
- \( n \): nth value
- \( o \): outside
- \( \text{out} \): outlet
- \( 2\Phi \): 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.

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

**FIG. 9-2**

Countercurrent Flow and Co-current Flow

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.

\[
\text{CMTD} = \text{LMTD} \times F = \text{Corrected Mean Temperature Difference} \quad \text{Eq 9-6}
\]

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:

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

And the weighted overall heat transfer coefficient becomes:

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

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

\[
U_t = \frac{1}{[\frac{1}{h_o} + \frac{A_o}{A_i} \left(\frac{1}{h_i} + r_w + r_f \right) \frac{A_i}{A_o} r_w]} \quad \text{Eq 9-11}
\]

Metal Resistance for Plain Tubes

The metal resistance is calculated by the following equation:

\[
r_w = \frac{D_o}{2 \times 1000 \times \text{kw}} \ln \frac{D_o}{D_i} \quad \text{Eq 9-12}
\]

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 TEMAx Standards.

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. 9-9. 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 \( U = 1/2r \) (as in Eq 9-11).
FIG. 9-3
LMTD Chart

\[ \text{LMTD} = \frac{\text{GTTD} \cdot \text{LTTD}}{\ln \frac{\text{GTTD}}{\text{LTTD}}} \]

GTTD AND LTTD INTERSECT AT LMTD ON LMTD SCALE

10 20 30 40 50 60 70 80 90 100 200 300 400 500 600 800 1000

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

\[
P = \frac{t_2 - t_1}{t_1 - t_2}
\]

CMTD = \left(\text{LMTD}\right)F_1

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

\[
P = \frac{t_2 - t_1}{t_1 - t_2}
\]

\[
R = \frac{T_1 - T_2}{t_2 - t_1}
\]

CMTD = \left(\text{LMTD}\right)F_2
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 new service or to compare different designs for a given service, variables (subscripted "2") can be applied based on relating 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:

\[
\frac{h_2}{h_1} = \left( \frac{\rho_2}{\rho_1} \right)^{0.47} \left( \frac{k_2}{k_1} \right)^{0.67} \left( \frac{C_p_2}{C_p_1} \right)^{0.33} \left( \frac{C_l}{C_l} \right)^{0.8} \left( \frac{D_2}{D_1} \right)^{0.2} \tag{Eq 9-13}
\]

and,

\[
\frac{AP_2}{AP_1} = \left( \frac{\mu_2}{\mu_1} \right)^{0.2} \left( \frac{G_2}{G_1} \right)^{1.8} \left( \frac{\rho_1}{\rho_2} \right) \left( \frac{D_1}{D_2} \right)^{1.2} \left( \frac{Np_2}{Np_1} \right) \tag{Eq 9-14}
\]

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)

\[
\frac{Q_H}{Q_C} = \frac{[\dot{m}_H \text{Cp}_H(T_{H_i} - T_{H_f})]}{[\dot{m}_C \text{Cp}_C(T_{C_2} - T_{C_1})]}
\]

\[
= \frac{[215\,784\,(2.270)\,(92 - 38)]}{[295\,225\,(2.51)\,(51 - 16)]} = 1.0
\]

2. Calculate the LMTD.

\[
\begin{align*}
92 \rightarrow 38 \\
51 \leftarrow 16
\end{align*}
\]

\[
LMTD = \frac{41 - 22}{\ln(41/22)} = 30.5°C
\]

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

3. Check the required heat transfer coefficient.

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

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

\[
A = \frac{[784\,\pi\,(D_1)^2]}{4} = (784\,(194.78)) = 152\,706 \text{ mm}^2
\]

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

\[
Re = \frac{(1.001)(15.7)(537)}{0.21} = 40\,200
\]

Therefore, it is turbulent flow since Re > 2000.

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

\[
(\tau_2)_2 = (f)(\tau_1)_1
\]
### FIG. 9-10

**Variables in Exchanger Performance**

<table>
<thead>
<tr>
<th>Variable*</th>
<th>Flow Regime</th>
<th>( \frac{\Delta P_2}{\Delta P_1} )†</th>
<th>( \frac{r_2}{r_1} )†</th>
<th>Shell (f)</th>
<th>Tube (f)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Viscosity</td>
<td>Turbulent</td>
<td>( \left( \frac{\mu_2}{\mu_1} \right)^{0.27} )</td>
<td>( \left( \frac{r_2}{r_1} \right)^{0.15} )</td>
<td>( \frac{\mu_2}{\mu_1} )</td>
<td>( \frac{r_2}{r_1} )</td>
</tr>
<tr>
<td>Viscosity – bulk to wall correction</td>
<td>Streamline</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Thermal conductivity</td>
<td>Turb. or Streamline</td>
<td>( \left( \frac{k_2}{k_1} \right)^{0.67} )</td>
<td>( \left( \frac{G_2}{G_1} \right)^{0.8} )</td>
<td>( \frac{k_2}{k_1} )</td>
<td>( \frac{G_2}{G_1} )</td>
</tr>
<tr>
<td>Sp. heat capacity</td>
<td>Turb. or Streamline</td>
<td>( \left( \frac{C_{p_1}}{C_{p_2}} \right)^{0.33} )</td>
<td>( \left( \frac{G_2}{G_1} \right)^{0.33} )</td>
<td>( \frac{C_{p_1}}{C_{p_2}} )</td>
<td>( \frac{G_2}{G_1} )</td>
</tr>
<tr>
<td>Mass velocity (or mass flowrate)</td>
<td>Turbulent &amp; Streamline</td>
<td>( \left( \frac{G_2}{G_1} \right)^{0.6} )</td>
<td>( \left( \frac{r_2}{r_1} \right)^{0.14} )</td>
<td>( \frac{G_2}{G_1} )</td>
<td>( \frac{r_2}{r_1} )</td>
</tr>
<tr>
<td>Density</td>
<td>Turb. or Streamline</td>
<td>( \left( \frac{\rho_2}{\rho_1} \right)^{0.4} )</td>
<td>( \left( \frac{L_2}{L_1} \right)^{0.33} )</td>
<td>( \frac{\rho_2}{\rho_1} )</td>
<td>( \frac{L_2}{L_1} )</td>
</tr>
<tr>
<td>Tube diameter</td>
<td>Turbulent</td>
<td>( \left( \frac{D_{o_2}}{D_{o_1}} \right)^{0.4} )</td>
<td>( \left( \frac{D_{o_2}}{D_{o_1}} \right)^{0.2} )</td>
<td>( \frac{D_{o_2}}{D_{o_1}} )</td>
<td>( \frac{D_{o_2}}{D_{o_1}} )</td>
</tr>
<tr>
<td>Tube diameter</td>
<td>Streamline</td>
<td>( \left( \frac{D_{i_2}}{D_{i_1}} \right)^{0.33} )</td>
<td>( \left( \frac{D_{i_2}}{D_{i_1}} \right)^{1.2} )</td>
<td>( \frac{D_{i_2}}{D_{i_1}} )</td>
<td>( \frac{D_{i_2}}{D_{i_1}} )</td>
</tr>
<tr>
<td>Tube length</td>
<td>Streamline</td>
<td>( \left( \frac{L_2}{L_1} \right)^{0.33} )</td>
<td>( \frac{L_2}{L_1} )</td>
<td>( \frac{L_2}{L_1} )</td>
<td>( \frac{L_2}{L_1} )</td>
</tr>
<tr>
<td>Tube passes</td>
<td>Turb. or Streamline</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>No. baffle spaces</td>
<td>Turb. or Streamline</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>No. tube rows crossed‡</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

* 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 \( f \)s.
‡ 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)**

<table>
<thead>
<tr>
<th>Fluid</th>
<th>Flow Regime</th>
<th>Local ( r )</th>
<th>( k )</th>
<th>( C_p )</th>
<th>( \rho )</th>
<th>( \Delta P/m )</th>
<th>( \mu^{(2)} )</th>
<th>( G_i )</th>
<th>( D_i )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water</td>
<td>Turbulent</td>
<td>0.000 16</td>
<td>0.620</td>
<td>4.19</td>
<td>995</td>
<td>1.54</td>
<td>0.764</td>
<td>1294</td>
<td>15.7</td>
</tr>
<tr>
<td>HC Oil</td>
<td>Turbulent</td>
<td>0.000 67</td>
<td>0.136</td>
<td>2.09</td>
<td>751</td>
<td>1.37</td>
<td>0.726</td>
<td>903</td>
<td>12.6</td>
</tr>
<tr>
<td>Methane</td>
<td>Turbulent</td>
<td>0.0010</td>
<td>0.035</td>
<td>2.26</td>
<td>4.32</td>
<td>3.10</td>
<td>0.0113</td>
<td>152</td>
<td>15.7</td>
</tr>
<tr>
<td>HC Oil</td>
<td>Streamline</td>
<td>0.0086</td>
<td>0.124</td>
<td>2.20</td>
<td>822</td>
<td>0.19</td>
<td>(3)</td>
<td>207</td>
<td>21.2</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Fluid</th>
<th>Flow Regime</th>
<th>Local ( r )</th>
<th>( k )</th>
<th>( C_p )</th>
<th>( \rho )</th>
<th>( \Delta P/m )</th>
<th>( \mu^{(5)} )</th>
<th>( G_o^{(7)} )</th>
<th>( D_o )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water</td>
<td>Turbulent</td>
<td>0.000 088</td>
<td>0.684</td>
<td>4.216</td>
<td>958</td>
<td>1.6</td>
<td>0.282</td>
<td>765.1</td>
<td>15.9</td>
</tr>
<tr>
<td>HC Oil</td>
<td>Turbulent</td>
<td>0.000 49</td>
<td>0.132</td>
<td>2.33</td>
<td>750</td>
<td>1.7</td>
<td>0.549</td>
<td>646.4</td>
<td>15.9</td>
</tr>
<tr>
<td>Methane</td>
<td>Turbulent</td>
<td>0.000 67</td>
<td>0.064</td>
<td>2.74</td>
<td>3.68</td>
<td>0.62</td>
<td>0.0182</td>
<td>30.2</td>
<td>15.9</td>
</tr>
</tbody>
</table>

(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 \( \Delta P/\text{baffle space}/10 \) tube rows crossed between centroids of cut openings
(7) Average crossflow mass velocity (see crossflow area calculation in Fig 9-13)
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.

\[
G = \frac{(215 784)}{(3600)(100 918/1 000 000)} = 593.9 \text{ kg/(m}^2 \cdot \text{s)}
\]

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

\[
(r_{n})_2 = (f)(r_{o})_1
\]

Use base values from Fig. 9-11 for \((r_{n})_1\) conditions.

\[
f = \left( \frac{\mu_2}{\mu_1} \right)^{0.47} \left( \frac{k_1}{k_2} \right)^{0.67} \left( \frac{C_{p1}}{C_{p2}} \right)^{0.33} \left( \frac{G_1}{G_2} \right)^{0.8} \left( \frac{D_2}{D_1} \right)^{0.2}
\]

\[
= \left( \frac{0.21}{0.726} \right)^{0.47} \left( \frac{0.136}{0.135} \right)^{0.67} \left( \frac{2.09}{2.51} \right)^{0.33} \left( \frac{903}{537} \right)^{0.8} \left( \frac{15.7}{12.6} \right)^{0.2}
\]

\[
= 0.840
\]

5. Basis: (Inside Area)

\[(r_{o})_2 = f(r_{o})_1 \text{ and } (r_{o})_1 = 0.000 67 \text{ from Fig. 9-11} = (0.840)(0.000 67) = 0.000 563 \text{ (m}^2 \cdot \text{°C)/W} \]

Use base values from Fig. 9-11 for \((\Delta P_{o})_1\) conditions.

\[
f = \left( \frac{\mu_2}{\mu_1} \right)^{0.2} \left( \frac{G_2}{G_1} \right)^{1.8} \left( \frac{D_2}{D_1} \right)^{1.2} \left( \frac{N_{P_{2}}}{N_{P_{1}}} \right)
\]

\[
= \left( \frac{0.21}{0.726} \right)^{0.2} \left( \frac{537}{903} \right)^{1.8} \left( \frac{751}{614} \right)^{1.2} \left( \frac{1}{1} \right)
\]

\[
= 0.285
\]

\[
(\Delta P_{o})_2 = (f)(\Delta P_{o})_1 = (0.285)(1.37) = 0.390 \text{ kPa/m}
\]

For a 9.15 m tube length the total 9.15(0.390) = 3.569 kPa

6. Calculate the overall heat transfer coefficient.

\[
\Sigma r = r_1 \left( \frac{A_o}{A_i} \right) + r_0 + r_w + r_0 + r_0 \left( \frac{A_o}{A_i} \right)
\]

\[
= 0.000 563 \left( \frac{0.0182}{0.0151} \right) + 0.000 49 + 0.000 036 + 0.000 35
\]

\[
+ 0.000 2 \left( \frac{0.0182}{0.0151} \right) = 0.0018
\]

\[
U = \frac{1}{\Sigma r} = \frac{1}{0.001 78} = 555.6 \text{ W/(m}^2 \cdot \text{°C)}
\]

7. Calculate the tube metal resistance.

\[
D_o = 19.05 \text{ mm}
\]

\[
D_i = 15.75 \text{ mm}
\]

\[
k_w = 50 \text{ W/(m} \cdot \text{°C)} \text{ from Fig. 9-8.}
\]

\[
r_w = \frac{D_o}{2 \cdot 1000 k_w \ln \frac{D_o}{D_i}}
\]

\[
= 0.000 036(\text{m}^2 \cdot \text{°C)/W}
\]

8. Compare the required heat transfer coefficient calculated in step 3 to the values 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 determination 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.
### PERFORMANCE OF ONE UNIT

<table>
<thead>
<tr>
<th>Fluid Allocation</th>
<th>Shell Side</th>
<th>Tube Side</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fluid Name</td>
<td>Lean Oil</td>
<td>Rich Oil</td>
</tr>
<tr>
<td>Fluid Quantity, Total kg/h</td>
<td>215 784</td>
<td>295 225</td>
</tr>
<tr>
<td>Steam</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Water</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

| Temperature (In/Out) °C | 92       | 38       | 16       | 51       |
| Density kg/m³          | 660 @ 64.8°C | 614 @ 33.4°C |
| Viscosity/Liquid mPa * s | 0.34 @ 64.8°C | 0.21 @ 33.4°C |
| Molecular Mass, Vapor   |            |           |
| Molecular Mass, Noncondensable |      |          |
| Specific Heat Capacity kJ/kg * °C | 2.27 @ 64.8°C | 2.51 @ 33.4°C |
| Thermal Conductivity J/(s * m² * °C/m) | 0.130 @ 64.8°C | 0.135 @ 33.4°C |
| Latent Heat kJ/kg * °C |            |           |
| Inlet Pressure kPa (ga) | 760       | 3000     |          |
| Velocity m/s           |            |           |
| Pressure Drop, Allow./Calc. kPa | 85        | /        | 15 /    |
| Fouling Resistance (Min./Calc.) m² * °C/W | 0.000 35/ | 0.000 18/ |
| Heat Exchanged 7.327 MW; MTD (Corrected) W/m² * °C | 3000      |          |
| Transfer Rate, Service | Clean     |           |
| Construction of One Shell Sketch (Bundle/Nozzle Orientation) |          |
| Design/Test Pressure kPa (ga) | 1 300/   | 3500/    |
| Design Temperature °C | 340       | 340      |          |
| No. Passes per Shell   | 1         | 1        |          |
| Corrosion Allowance mm | 2         | 2        |          |
| Connections In         |            |           |
| Size & Rating          | Out       | Intermediate |
| Tube No. OD 18 mm; thk (Avg) 1.651 mm Length 9 m Pitch 2.4 mm ▲30 ▼60 △90 ◆45 |
| Tube Type ID OD Shell steel Material steel (Integ./Remov.) |            |           |
| Shell Cover —          | Shell Cover — |          |
| Channel or Bonnet steel | Integral | Channel Cover steel |
| Tubesheet-Stationary steel | Integral | Tubesheet-Floating — |
| Floating Head Cover —  | Impingement Protection |
| Baffles-Cross Type % Cut (Diam/Area) Spacing: c/c Inlet mm |            |          |
| Baffles-Long Seal Type  |            |           |
| Supports-Tube U-Bend   |            |           |
| Bypass Seal Arrangement Tube-Tubesheet Joint |          |
| Expansion Joint Type   |            |           |
| (m² - Inlet Nozzle) Bundle Entrance Bundle Exit |          |
| Gaskets-Shell Side Tube Side |       |
| Code Requirements ASME TEMA Class C |      |          |
| Weight/Shell Filled with Water Bundle kg |            |          |
| Remarks                |            |           |
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 (“dry-wall” 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)
  \[ U_v = 396.6 \text{ W/(m}^2 \cdot \text{°C)} \]
  \[ h_v = 630.4 \text{ W/(m}^2 \cdot \text{°C)} \]

- **Condensing** (42°C to 42°C)
  \[ U_c = 794.4 \text{ W/(m}^2 \cdot \text{°C)} \]

- **Subcooling** (42°C to 35°C)
  \[ U_L = 649.7 \text{ W/(m}^2 \cdot \text{°C)} \]

**Solution Steps**

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

   \[ T_{w0} = T_v - \frac{U_v(T_r - T_C)}{h_v} \]
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 $Q = UA \left( \frac{LMTD}{A} \right)$ for the required surface area in each zone. The sum of these areas is the surface required for the exchanger.

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

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.

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

$$Q_1 = \frac{(82 - 56)}{(82 - 42)} \times 2650 000$$

$$Q_1 = 1 730 000 W \quad \text{(Heat removed from vapor between 82°C and 56°C)}$$

$$T_C = 34 - \left[ \frac{1 730 000}{12 150 000} (34 - 28) \right] = 33.15°C$$

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

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

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

ZONE 1

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

$$\begin{align*}
82 & \rightarrow 56.0 \\
34 & \leftarrow 33.15 \\
48 & 22.8
\end{align*}$$

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

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

ZONE 2

$$Q = Q_v - Q_1 + Q_2 \quad (Q_v \text{ from Fig. 9-14})$$

$$= 2 650 000 - 1 730 000 + 8 880 000$$

$$= 9 800 000 W$$

$$T_C = 34 - \left[ \frac{1 730 000 + 9 800 000}{12 150 000} \right] (34 - 28) = 28.3°C$$

(Note: You do not use 56°C as the inlet temperature to this zone.)
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 to a reboiler has pulsations (intermittent liquid flow reversal) generated by an instantaneous vapor acceleration pressure drop that temporarily causes the total pressure drop to exceed available static head. The heat medium temperature affects the same limiting conditions and may be the controlling variable when recirculation is not possible. Design and operating analysis requires a study of hydraulics, heat medium temperature, and exchanger geometry for a particular fluid to define valid limitations on a reboiler. Analytical methods are available in the technical literature noted in the bibliography.

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.

![FIG. 9-15](image1)

**A Typical Pool Boiling Curve**

![FIG. 9-16](image2)

**Typical Overall Boiling Heat Flux Ranges**

<table>
<thead>
<tr>
<th>Heat Medium</th>
<th>Boiling Fluid</th>
<th>Heat Flux Range, W/m²</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hot Oil</td>
<td>C₁-C₂ HC</td>
<td>22 000 - 25 000</td>
</tr>
<tr>
<td></td>
<td>C₃-C₄ HC</td>
<td>20 000 - 44 000</td>
</tr>
<tr>
<td></td>
<td>Rich Oil</td>
<td>8 000 - 11 000</td>
</tr>
<tr>
<td></td>
<td>Amines</td>
<td>11 000 - 17 000</td>
</tr>
<tr>
<td>HC Gas</td>
<td>C₁-C₂ HC</td>
<td>6 000 - 14 000</td>
</tr>
<tr>
<td>Steam</td>
<td>C₁-C₂ HC</td>
<td>25 000 - 41 000</td>
</tr>
<tr>
<td></td>
<td>C₃-C₅ HC</td>
<td>31 000 - 47 000</td>
</tr>
<tr>
<td></td>
<td>Rich Oil</td>
<td>13 000 - 19 000</td>
</tr>
<tr>
<td></td>
<td>Amines</td>
<td>15 000 - 21 000</td>
</tr>
</tbody>
</table>

![FIG. 9-17](image3)

**Two-Phase Flow Regimes in Vertical Tubes**

[Caption: Courtesy of HTRI]
Types of Reboilers

Kettle — Kettle reboilers are commonly applied when a wide range of process operations (high turndown capability), large heat exchange surface, or high vapor quality is required. Installations include column bottom reboilers, side reboilers, or vaporizers. Fig. 9-18 shows a typical kettle. Kettles are generally more costly than other reboiler types due to shell size, surge volume size, and uncertainty in the TMTD. Without TMTD or fouling problems, a column-internal (stab-in) reboiler would be suitable if the required surface is relatively small.

Recirculating thermosyphon — Recirculating thermosyphon reboilers are applicable when process operations are consistently near design rates. Typically, these are vertical tube side boiling, like Fig. 9-19, or for large surface requirements, horizontal shell side boiling. Installation requires a fixed static head, such as a partitioned column sump or a head drum, for recirculation. Recirculating thermosyphon reboilers are generally the least costly of reboiler types (other than column-internal type) due to maximized heat transfer, accurate TMTD, and relatively low fouling tendencies (due to higher velocities).

“Once-through” — Once-through reboilers are applicable when the feed is available without the capability for recirculation. These boilers may be called “thermosyphons” when taking a column tray liquid as feed such as shown in Fig. 9-20. Once-through reboilers and vaporizers have the lowest fluid residence time on the hot surface and have a fixed downstream pressure which fixes the inlet pressure to the reboiler (externally fixed head is not required). However, they have the narrowest range of stable hydraulics and heat medium temperatures in the wet wall regions of boiling due to the fixed flowrate. Substantial process judgment and analytical support are required for satisfactory performance. Once-through reboilers can be in either the horizontal or vertical position and have been designed for either shellside or tubeside boiling.

“Pump-through” — Pump-through or pump-around reboilers are applicable when handling viscous liquid or particulate-laden liquid, and when liquid heating by pressure suppressed vaporization are desirable. Any arrangement of shell side or tube side boiling, vertical or horizontal may be used, but Fig. 9-21 is a typical arrangement. Pump-through reboilers may or may not include recirculated liquid, but usually do. Suppressed vaporization operation requires a throttling valve in the outlet line of the reboiler to generate vapor at the downstream fixed pressure.
Type Selection — Reboiler type selection generally follows the guidelines of Fig. 9-22.

SELECTION OF EXCHANGER COMPONENTS

Industry Standards

Shell and tube heat exchanger technology for gas, chemical, and petroleum processing plants has developed a broad basis of common understanding through the “Standards of Tubular Exchangers Manufacturers Association” (TEMA). These “TEMA Standards” provide nomenclature, dimensional tolerances, manufacturer’s and purchaser’s responsibilities, general installation and operating guidelines, and specific design and fabrication practices.

The design and fabrication practices of TEMA are in three classifications, called Class “R,” “C,” or “B.” Class “R” is applied to services with severe operating and maintenance characteristics. Class “C” is for the least severe characteristics. Class “B” is for chemical process applications between Classes “R” and “C.” All classes are intended to be limited to ASME Code, Section VIII, Div. 1, cylinder wall thicknesses of less than about 2”, and stud diameters of less than about 3”; though thicker components can be applied by the design practices specified.

TEMA Standards provide a “Recommended Good Practice” for the designer’s consideration in areas outside of the limits of the specified standards. Guidance and references are noted for seismic design, large diameter exchangers, tube vibration, tube-to-tubesheet stress analysis, nozzle loading analysis, and numerous other design-limiting features.

Detailed understanding of shell and tube exchangers for use in the process industry requires an understanding of the TEMA Standards. Other industry standards as may be offered by ASME, API, or ANSI can be applied in a particular situation with or without TEMA Standards. The purchase order and specification sheet for a particular service will normally identify the applicable industry standards.

Nomenclature

Fig. 9-23 summarizes the major shell-and-tube exchanger components other than tubes and baffles. The letters are used for a standard nomenclature in the industry. A three-letter
FIG. 9-23
Shell and Tube Exchanger Nomenclature

<table>
<thead>
<tr>
<th>FRONT END STATIONARY HEAD TYPES</th>
<th>SHELL TYPES</th>
<th>REAR END HEAD TYPES</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>E</td>
<td>L</td>
</tr>
<tr>
<td>BONNET (INTEGRAL COVER)</td>
<td>ONE PASS SHELL</td>
<td>FIXED TUBESHEET LIKE &quot;A&quot; STATIONARY HEAD</td>
</tr>
<tr>
<td>B</td>
<td>F</td>
<td>M</td>
</tr>
<tr>
<td>BONNET (INTEGRAL COVER)</td>
<td>TWO PASS SHELL WITH LONGITUDINAL BAFFLE</td>
<td>FIXED TUBESHEET LIKE &quot;B&quot; STATIONARY HEAD</td>
</tr>
<tr>
<td>C</td>
<td>G</td>
<td>N</td>
</tr>
<tr>
<td>REMOVABLE TUBE BUNDLE ONLY</td>
<td>SPLIT FLOW</td>
<td>FIXED TUBESHEET LIKE &quot;N&quot; STATIONARY HEAD</td>
</tr>
<tr>
<td>D</td>
<td>H</td>
<td>P</td>
</tr>
<tr>
<td>REMOVABLE TUBE BUNDLE ONLY</td>
<td>DOUBLE SPLIT FLOW</td>
<td>OUTSIDE PACKED FLOATING HEAD</td>
</tr>
<tr>
<td>E</td>
<td>J</td>
<td>S</td>
</tr>
<tr>
<td>CHANNEL INTEGRAL WITH TUBESHEET AND REMOVABLE COVER</td>
<td>DIVIDED FLOW</td>
<td>FLOATING HEAD WITH BACKING DEVICE</td>
</tr>
<tr>
<td>F</td>
<td>K</td>
<td>T</td>
</tr>
<tr>
<td>CHANNEL INTEGRAL WITH TUBESHEET AND REMOVABLE COVER</td>
<td>KETTLE TYPE REBOILER</td>
<td>PULL THROUGH FLOATING HEAD</td>
</tr>
<tr>
<td>G</td>
<td>L</td>
<td>U</td>
</tr>
<tr>
<td>CROSS FLOW</td>
<td></td>
<td>U-TUBE BUNDLE</td>
</tr>
<tr>
<td>H</td>
<td>M</td>
<td>W</td>
</tr>
<tr>
<td></td>
<td></td>
<td>EXTERNALLY SEALED FLOATING TUBESHEET</td>
</tr>
</tbody>
</table>

Courtesy of TEMA
type designation in the order of front head type, shell type, and rear head type is used. For example, an AJIS would have a front head that is removable with a removable cover, a shell that is arranged for divided flow, and a rear floating head with a backing device (usually a split-ring). Factors to consider in selecting a shell and tube exchanger type are summarized in Fig. 9-24.

**Tube Wall Determination**

The required tube wall thickness is determined from the ASME Code, Section VIII, Division 1 for cylinders under internal or external pressure. If U-tubes are used, the thinning of the tube wall in the bends must be considered. A minimum wall tube whose thickness is equal to or greater than the calculated thickness may be used, or an average wall tube whose minimum thickness is equal to or greater than the calculated thickness may be used. It is satisfactory to use an average wall tube that is one BWG heavier than the required minimum wall thickness; however, it is not always possible to substitute a minimum wall tube that is one BWG thinner than a specified average wall thickness tube. If the calculated wall thickness is less than the value recommended by TEMA, the TEMA values are used. Fig. 9-25 summarizes standard tube data.

**Shell Size and Tube Count Estimation**

The tube count in a given shell diameter varies with the tube diameter, tube spacing and layout (pitch), type of tube bundle, number of tube passes, and the shell side entrance and exit area allowed. After selecting an appropriate tube outside diameter and tube length, the number of tubes required to result in a given heat transfer surface can be calculated using the external square meter/meter data from Fig. 9-25.

Fig. 9-26 is a plot of tube count vs. diameter for four different triangular tube pitches most commonly used in shell and tube exchangers. Entering these curves with the required tube count will give a diameter which can be corrected for the various factors noted to determine the actual shell diameter required.

To correct for square pitch, multiply the shell inside diameter from Fig. 9-26 by 1.075. No correction factor is needed for any other pitch. To allow for entrance or exit areas, multiply shell inside diameter from Fig. 9-26 by 1.02 for each inlet or outlet area to be used. Fig. 9-27 is a table of factors to correct inside shell diameter for pass arrangement.

Fig. 9-28 is a table of adders to correct for type of construction.

**Example 9-3** — Determine the shell diameter for 320 tubes, 25 mm OD spaced on a 32 mm square pitch layout, four-pass tubes, in a split ring type floating head shell and tube exchanger, with inlet flow area allowed.

**Solution Steps**

1. From the top curve of Fig. 9-26 read 630 mm corresponding to 320 tubes for the given tube spacing and pitch.
2. Correct for square pitch by multiplying by 1.075.
3. Using Fig. 9-27 correct for four pass by multiplying by 1.05.
4. Correct for inlet flow area by multiplying by 1.02.

Accumulative multiplier is $1.075 \times 1.05 \times 1.02 = 1.15$. Partially corrected diameter $= 630 \text{ mm} \times 1.15 = 725 \text{ mm}$.

5. From Fig. 9-28, correct for split ring floating head by adding $25 + 725 = 750 \text{ mm}$.

So use a 750 mm ID shell for this tube count and configuration.

**Enhanced Surface Tubing**

Heat exchanger applications in which one of the fluids has a high heat transfer coefficient relative to the other fluid can benefit (either from lower first cost of a new exchanger or increased capacity in an existing unit) by use of specially enhanced tube surfaces on the side with the low coefficient. One

---

**FIG. 9-24**

Shell and Tube Exchanger Selection Guide (Cost Increases from Left to Right)

<table>
<thead>
<tr>
<th>Type of Design</th>
<th>&quot;U&quot; Tube</th>
<th>Fixed Tubesheet</th>
<th>Floating Head Outside Packed</th>
<th>Floating Head Split Backing</th>
<th>Floating Head Pull-Through Bundle</th>
</tr>
</thead>
<tbody>
<tr>
<td>Provision for differential expansion</td>
<td>individual tubes free to expand</td>
<td>expansion joint in shell</td>
<td>floating head</td>
<td>floating head</td>
<td>floating head</td>
</tr>
<tr>
<td>Removeable bundle</td>
<td>yes</td>
<td>no</td>
<td>yes</td>
<td>yes</td>
<td>yes</td>
</tr>
<tr>
<td>Replacement bundle possible</td>
<td>yes</td>
<td>not practical</td>
<td>yes</td>
<td>yes</td>
<td>yes</td>
</tr>
<tr>
<td>Individual tubes replaceable</td>
<td>only those in outside row</td>
<td>yes</td>
<td>yes</td>
<td>yes</td>
<td>yes</td>
</tr>
<tr>
<td>Tube interiors cleanable</td>
<td>difficult to do mechanically, can do chemically</td>
<td>yes, mechanically or chemically</td>
<td>yes, mechanically or chemically</td>
<td>yes, mechanically or chemically</td>
<td>yes, mechanically or chemically</td>
</tr>
<tr>
<td>Tube exteriors with triangular pitch cleanable</td>
<td>chemically only</td>
<td>chemically only</td>
<td>chemically only</td>
<td>chemically only</td>
<td>chemically only</td>
</tr>
<tr>
<td>Tube exteriors with square pitch cleanable</td>
<td>yes, mechanically or chemically</td>
<td>chemically only</td>
<td>yes, mechanically or chemically</td>
<td>yes, mechanically or chemically</td>
<td>yes, mechanically or chemically</td>
</tr>
<tr>
<td>Number of tube passes</td>
<td>any practical even number possible</td>
<td>normally no limitations</td>
<td>normally no limitations</td>
<td>normally no limitations</td>
<td>normally no limitations</td>
</tr>
<tr>
<td>Internal gaskets eliminated</td>
<td>yes</td>
<td>yes</td>
<td>yes</td>
<td>no</td>
<td>no</td>
</tr>
</tbody>
</table>
### Characteristics of Tubing

<table>
<thead>
<tr>
<th>Tube O.D.</th>
<th>Nominal Tube Size</th>
<th>B.W.G. Gauge</th>
<th>Thickness</th>
<th>Internal Area</th>
<th>Sq. Meter External Surface</th>
<th>Sq. Meter Internal Surface</th>
<th>Weight Per Meter Steel</th>
<th>Tube I.D.</th>
<th>Moment of Inertia</th>
<th>Section Modulus</th>
<th>Radius of Gyration</th>
<th>Constant C**</th>
<th>O.D. T.D.</th>
<th>Transverse Metal Area</th>
</tr>
</thead>
<tbody>
<tr>
<td>3/8</td>
<td>9</td>
<td>0.711</td>
<td>0.079</td>
<td>0.0155</td>
<td>0.099</td>
<td>4.828</td>
<td>50.67</td>
<td>16.02</td>
<td>2.009</td>
<td>46</td>
<td>1,268</td>
<td>12.60</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1/4</td>
<td>9</td>
<td>0.711</td>
<td>0.079</td>
<td>0.0155</td>
<td>0.099</td>
<td>4.828</td>
<td>50.67</td>
<td>16.02</td>
<td>2.009</td>
<td>46</td>
<td>1,268</td>
<td>12.60</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1/4</td>
<td>12</td>
<td>0.711</td>
<td>0.079</td>
<td>0.0155</td>
<td>0.099</td>
<td>4.828</td>
<td>50.67</td>
<td>16.02</td>
<td>2.009</td>
<td>46</td>
<td>1,268</td>
<td>12.60</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1/4</td>
<td>18</td>
<td>0.889</td>
<td>0.029</td>
<td>0.0221</td>
<td>0.234</td>
<td>7.036</td>
<td>59.56</td>
<td>2.960</td>
<td>94</td>
<td>1,354</td>
<td>32.36</td>
<td>19.69</td>
<td></td>
<td></td>
</tr>
<tr>
<td>3/8</td>
<td>22</td>
<td>0.889</td>
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<td>0.0221</td>
<td>0.234</td>
<td>7.036</td>
<td>59.56</td>
<td>2.960</td>
<td>94</td>
<td>1,354</td>
<td>32.36</td>
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<td>0.0221</td>
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<td>59.56</td>
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<td>7.036</td>
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<td>32.36</td>
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<td>7.036</td>
<td>59.56</td>
<td>2.960</td>
<td>94</td>
<td>1,354</td>
<td>32.36</td>
<td>19.69</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

*Weights are based on low carbon steel with a density of 7850 kg/m³. For other metals multiply by the appropriate factors:*  
Aluminum 0.35  
Titanium 0.58  
A.I.S.I. 400 Series Stainless Steels 0.99  
A.I.S.I. 300 Series Stainless Steels 1.02  
Aluminium Bronze 1.04  
Aluminium Brass 1.06  
Nickel - Chrome - Iron 1.07  
Admiralty 1.09  
Nickel and Copper - Nickel 1.13  
Copper and Copper - Nickel 1.14  

**Liquid Velocity (m/s) =**  

\[ \frac{\text{mass flow per tube (kg h} \times \text{tube})}{1672 \text{ C density (kg m}^{-3})} \]

Derived from TEMA data.
commonly used tube is a "low finned" tube which has extruded fins on the outside of the tube and the diameter outside the fins is no greater than the outside diameter of the plain ends so the exchanger can be assembled or retubed in the same way as a bare tube exchanger. The effect is to increase the heat transfer surface of the tube approximately 250% to result in a more compact exchanger for a given service compared to one using bare tubes. These tubes perform favorably in clean applications such as light hydrocarbon condensers where vapor velocity permits a condensate film to be distributed over more surface per tube. These tubes are available in metals commonly used in most heat exchangers.

High heat flux tubes with special coatings to create a porous surface are sometimes used where liquid velocities permit nucleate boiling to increase the heat flux per tube provided the porous surface remains exposed to the liquid.

For even more specialized considerations of fluid properties and operating requirements, a tube wall may be extruded at or near thickness to a variety of shapes. A convoluted spirally extruded tube wall offers a range for the hydraulic diameter that may be optimized for the fluids considered.

Other than low finned tubes, most enhanced surface tubes are limited to materials uniquely suited to the particular enhanced surfaces and special fabrication limitations. The limitations on application and availability as dictated in specific supplier’s literature must be considered.

**OPERATING CHARACTERISTICS**

**Inlet Gas Exchanger**

The familiar feed-to-residue gas exchanger is characterized by a close temperature approach between the two streams over a long temperature range which requires countercurrent flow arrangement. For overall economy this service will have very long tubes and low pressure drops in an optimized design. Such design will include adequate protection from hydrate formation in the feed gas and a baffle arrangement suitable for low shellside pressure drop and no significant tube vibration.

In wet gas streams hydrate formation is normally prevented by spraying methanol or ethylene glycol on the face of the front tubesheet. Critical to the effectiveness of that injection is the spray coverage of the tube field and a tube side velocity sufficient to achieve annular (wet wall) flow in each tube as shown in Fig. 9-17.

To maintain countercurrent flow arrangement baffle variations may be considered to minimize shell side pressure drop. A variety of multi segmented baffles offer lower pressure drop per cross pass than the segmental type. Proprietary low pressure drop devices such as wire (or rod) web baffles may be appropriate if the loss in heat transfer is not significant. When tube vibration is a prime concern, a segmentally cut baffle arrangement with no tubes in the cut out window provides nonproprietary maximum tube support for a given pressure drop.

**Tube Vibration**

Tubes or tube bundles can be excited to sufficient movement to create noise, tube damage, and/or baffle damage. Some tube field geometries are particularly susceptible to acoustical resonance. Any tube has a natural frequency of vibration dependent on its supported span, size, and density. When velocity of a fluid induces cyclic forces approximating that natural frequency, vibration occurs. The first mode of vibration (lowest natural frequency) occurs at the half wave length equal to the supported span and is the usual case for analysis. However, higher modes of vibration are possible when multiple half wave lengths coincide with the supported span length. Since tube bundles have damping characteristics, damage may or

---

**FIG. 9-26**
Tube Count vs. Diameter for Triangular Tube Pitch

<table>
<thead>
<tr>
<th>Shell Diameter, mm</th>
<th>Number of Tube Passes</th>
<th>Two</th>
<th>Four</th>
<th>Six</th>
<th>Eight</th>
</tr>
</thead>
<tbody>
<tr>
<td>less than 300 mm</td>
<td>1.10</td>
<td>1.20</td>
<td>1.35</td>
<td>—</td>
<td></td>
</tr>
<tr>
<td>300 to 600 mm</td>
<td>1.03</td>
<td>1.08</td>
<td>1.12</td>
<td>1.25</td>
<td></td>
</tr>
<tr>
<td>610 to 1000 mm</td>
<td>1.02</td>
<td>1.05</td>
<td>1.07</td>
<td>1.08</td>
<td></td>
</tr>
<tr>
<td>greater than 1000 mm</td>
<td>1.01</td>
<td>1.03</td>
<td>1.04</td>
<td>1.06</td>
<td></td>
</tr>
</tbody>
</table>

** FIG. 9-27**
Correction Factors for Number of Tube Passes

<table>
<thead>
<tr>
<th>Shell Diameter, mm</th>
<th>Type of Construction</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Fixed Tubesheet</td>
</tr>
<tr>
<td>less than 700 mm</td>
<td>None</td>
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<tr>
<td>710 to 1000 mm</td>
<td>None</td>
</tr>
<tr>
<td>greater than 1000 mm</td>
<td>None</td>
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</tbody>
</table>
may not occur at a particular mode of vibration. A substantial
bibliography of analytical methods as well as calculation pro-
cedures for this subject are presented in the Recommended
Good Practice section of TEMA standards.

Evaluating Altered Performance

Exchanger performance will deviate when:
1. Process conditions are altered by feedstock, throughput,
   control/instrumentation, or mechanical failure of adjoin-
ing equipment.
2. Corrosion or fouling material achieves a limiting condition.
3. Internal leakage or mechanical failure has occurred in
   the exchanger.

Operating records and overall process analysis can address
most problems except fouling, corrosion, internal leakage, and
mechanical failure within the exchanger.

If a relief valve is overpressured on the low pressure side of
an exchanger, it suggests interstream leakage or a near total
flow restriction on the low pressure side. Substantial loss of
pressure on the high pressure side confirms interstream leak-
age. The soundness of tubes, tubesheets, internally gasketed
joints, and/or internal expansion joints must be tested and the
failed components repaired, replaced, or plugged. A relief
valve overpressuring on the high pressure side suggests a flow
restriction downstream of the relief valve connection.

Flow restriction not accountable to operating changes in the
process analysis is probably attributable to fouling debris
somewhere in one or both stream systems. If such flow restric-
tion occurs gradually (several days to several months), a sys-
tematic inspection with cleaning as needed is probably
required. If such flow restriction occurs quickly (seconds to
hours), mechanical failure or a process step-change probably
occurred somewhere in the stream system. Only an available
flow bypass around the exchanger can isolate and identify the
flow restriction in the exchanger. A flow restriction anywhere
in a stream system will alter an exchanger’s heat transfer ef-
fect on both stream systems involved. The process analysis
should indicate which stream has consequential limits and
which stream is a problem source.

Perhaps the most difficult performance problem to isolate in
operation is the discrimination between pass partition leakage
and fouling, though fouling, being expected, is often presumed.
Obviously in a new or clean exchanger, a bad gasket or fit-up
might immediately come to mind; but when the unit is partially
fouled, pressure drop data may or may not indicate which specific
problem is occurring. Comparing pressure drop data to normal
operation may be the best available indication while the unit is
in service. In cases where continued operation would not have
serious consequences in reaction products, product quality, cor-
rosion, or economics due to unachieved heat transfer, this judg-
ment may best be delayed until the unit is out-of-service. Though
fouling may be observable, close examination of all pass plate
dges, gaskets, flatness, and groove edges in tubesheets and
channel covers should expose the problem.

Hairpin Heat Exchangers

Hairpin heat exchangers are designed in a hairpin shape
and are fabricated in accordance with ASME code. The design
consists of shell and tube closures proprietary for each vendor.
Hairpins are divided into two major types: Double Pipe and
Multi-tube.

The Double Pipe type, shown in Fig. 9-29, consists of a single
tube or pipe, either finned or bare, inside a shell. The Multi-
tube type, shown in Fig. 9-30, consists of several tubes, either
finned or bare, inside a shell. The maximum pressure rating
of hairpin exchangers depends on a number of key design con-
siderations including nozzles, closures, and material of con-
struction. Standard designs are available for pressures up to
5000 psig on tube side and 500 psig on shell side, and special
designs can be fabricated for higher pressures.

Hairpin sections are specially designed units which are nor-
mally not built to any industry standard other than ASME
Code. However, TEMA tolerances are normally incorporated,
wherever applicable.

Advantages

1. The use of longitudinal finned tubes will result in a com-
pact heat exchanger for shellside fluids having a low heat
transfer coefficient.
2. Countercurrent flow will result in lower surface area re-
quirements for services having a temperature cross.
3. Potential need for expansion joint is eliminated due to U-tube construction.

4. Shortened delivery times can result from the use of stock components that can be assembled into standard sections.

5. Modular design allows for the addition of sections at a later time or the rearrangement of sections for new services.

6. Simple construction leads to ease of cleaning, inspection, and tube element replacement.

Disadvantages

1. Multiple hairpin sections are not always economically competitive with a single shell and tube heat exchanger.

2. Proprietary closure design requires special gaskets.

Application Guidelines

The suitability of using hairpin exchanger in a given application is frequently evaluated by computing the UA product from the basic heat transfer equation (see Eq 9-5):

\[ UA = \frac{Q}{\text{LMTD}} \]

For preliminary evaluation, \( UA = 79,000 \) may be considered to be the upper economical limit for applying hairpin type units.

- \( UA = 79,000 \text{ W/°C} \)

Above this value, the unit may be uneconomical for a hairpin type design. If a hairpin is applied, it may require multiple 400 mm multitube sections.

- \( UA = 53,000 \) to 79,000 W/°C

In this range, one or more 300 mm to 400 mm multitube sections will normally be required.

- \( UA = 26,000 \) to 53,000 W/°C

In this range, one or more 100 mm to 300 mm multitube sections will normally be required.

- \( UA = 26,000 \text{ W/°C} \)

Below this value, both double pipe and multitube sections should be evaluated.

Fig. 9-31 lists typical sizes for hairpin type exchangers.

**Tank Heaters**

Tank heaters are required to maintain or increase the temperature of a tank’s contents. Tank heaters are relatively inefficient in heat transfer since the fluid inside the tank normally does not flow with appreciable velocity over the heat transfer surface and the low velocities produce a high tankside film resistance. Many different methods exist for heating tanks. Some of the more common ones are described below.

**Wall Mounted Coils or Panels**

Wall mounted coils are common on small insulated tanks when the contents must be maintained at a constant temperature, usually at or near the inlet temperature. Design variations include simply tracing the tank with small tubing suitable for heating with steam or other hot fluid, to tanks whose walls are actually constructed from prefabricated heat transfer panels.

**Internal Prefabricated Tank Heaters**

Internal heaters are available in a wide variety of configurations. Steam is commonly used as the heating medium, but many other hot fluids have been used in actual practice. Externally finned pipe is commonly used to improve heat transfer efficiency. For large tanks, multiple units similar to that shown in Fig. 9-32 are supported near the bottom of the tank. For small tanks, the same tubing in a vertical configuration can be provided.

**Internal Pipe Coils**

Internal pipe coils are normally fabricated from 2 inch schedule 40 pipe supported near the bottom of the tank. Ser-
FIG. 9-34
Basic Components of a Three Stream Counterflow Brazed Aluminum Heat Exchanger

Courtesy ALTEC International, Inc.
pentine shaped coils are most common although circular coils are also used.

Prefabricated Stab-in Tube Bundle

Prefabricated stab-in bundles are similar to a kettle tube bundle. Removable U-tube bundles are mounted through an appropriately sized opening near the tank bottom.

Tank Suction Heaters

A tank suction heater is shown in Fig. 9-33. This type of unit raises the temperature and thus reduces the viscosity of a pumped fluid without heating the tank fluid that remains in the tank.

Plate-Fin Exchangers (Brazed Aluminum)

Brazed aluminum heat exchangers have been employed in cryogenic gas processing plants since the 1950s. This section briefly describes the basic configuration, advantages, hardware capabilities, rough section criteria, and user considerations for brazed aluminum heat exchangers used in gas processing applications.

BASIC CONFIGURATION

A brazed aluminum heat exchanger is composed of alternating layers of corrugated fins and flat separator sheets called parting sheets. A stack of fins and parting sheets comprise the heat exchanger, sometimes referred to as the “core.” The heat exchanger is normally specified by its outside dimensions in the following order: Width (W) x Stack Height (H) x Length (L). The number of layers, type of fins, stacking arrangement, and stream circuiting will vary depending on the application requirements. The basic aluminum components of the heat exchanger are shown in Fig. 9-34 for a typical three stream counterflow exchanger.

Nozzles — Nozzles are the pipe sections used to connect the heat exchanger headers to the piping.

Headers — Headers are the half cylinders which provide for the distribution of fluid from the nozzle to or from the ports of each appropriate layer within the heat exchanger.

Ports — Ports are the opening in either the side bar or the end bar, located under the headers, through which the fluids enter or leave the individual layers.

Distributor Fins — Distributor fins distribute the fluid between the port and the heat transfer fins.

Heat Transfer Fins — The heat transfer fin provides extended heat transfer surface. All fins, both heat transfer and distributor, provide connecting structure between the parting sheets, which is necessary for the structural and pressure holding integrity of the heat exchanger. Typical heat transfer fin thicknesses range from 0.15 to 0.58 mm.

Parting Sheets — The parting (separator) sheets contain the fluids within the individual layers in the exchanger and also serve as primary heat transfer surface. Typical parting sheet thicknesses range from 0.80 to 2.0 mm.

Outside Sheets — Outside (cap) sheets serve as the outside parting sheets. They are typically 6 mm thick and serve as an outer protective surface of the exchanger.

Bars — The side and end bars enclose the individual layers and form the protective perimeter of the exchanger. Solid extruded bars from 12 to 25 mm wide are typically used.

Support Angles — Support angles are typically 90 degree extruded aluminum angles welded to the bar face of the exchanger for the purpose of supporting or securing the exchanger in its installed position. Other support configurations are available.

Battery — A multiple exchanger assembly, sometimes referred to as a “battery,” consists of two or more exchangers piped or manifolded together into a single assembly, with the individual exchangers arranged either in a parallel, series, or combination parallel-series arrangement.

Cold Box — Individual exchangers and batteries are often installed in a “cold box.” A cold box consists of a welded, airtight carbon steel casing, rectangular or cylindrical in shape, which supports and houses the heat exchangers, piping, other related cryogenic equipment, and insulation material.

ADVANTAGES AND LIMITATIONS

Aluminum maintains excellent strength and ductility to temperatures as low as –270°C. Aluminum actually increases in strength at cold temperatures. Because of its relatively low melting temperature, however, aluminum is less resistant to fires and high temperatures than are some other materials. Aluminum is generally not employed for process temperatures above 65°C, especially when higher pressures are involved. Aluminum exchangers are less resistant to rough handling and mistreatment than steel equipment and are limited to use with fluids non-corrosive to aluminum.

Brazed aluminum heat exchangers are compact and lightweight. A typical high pressure brazed aluminum heat exchanger [4100 - 9500 kPa (ga) design pressure] will provide 28 to 37 m² of heat transfer surface per cubic foot of exchanger volume. This is six to eight times the surface density of comparable shell and tube exchangers. Additionally, a typical high pressure brazed aluminum heat exchanger will have a density of 1200 to 1400 kg per cubic meter of exchanger versus approximately 4000 kg per cubic meter for comparable shell and tube exchangers. The net effect of these differences is that a brazed aluminum heat exchanger will provide approximately 25 times more surface per pound of equipment than comparable shell and tube exchangers. This decrease in exchanger weight and volume reduces foundation, support, plot plan, and insulation requirements.

An important point to note when evaluating the size and efficiency of a brazed aluminum heat exchanger is that it is customary to include total surface in all streams, hot and cold. This is equivalent to counting both the inside and outside tube surfaces in a tubular heat exchanger. This method for specifying surface is used because brazed aluminum heat exchangers will often be designed with an unbalanced surface. More surface is provided on one stream of the heat exchanger than in the other stream(s) in order to balance the variation in heat transfer coefficients. Up to ten streams can be combined into a single brazed aluminum heat exchanger; combining counterflow, crossflow, and cross-counterflow circuiting. Temperature approaches of 1.5°C on single-phase fluids and 2.75°C on two-phase fluids can be achieved. Typically, corrected mean temperature differences of 3°C to 6°C are employed in brazed aluminum heat exchanger applications.
Brazed aluminum heat exchangers should be used with clean fluids since they are more susceptible to plugging than other types of heat exchanger equipment; however, proper filters will prevent heat exchanger fouling. Brazed aluminum should not be used with fluids which are corrosive to aluminum. Mercury and caustic soda are extremely corrosive to aluminum and should not be introduced into the exchanger. Hydrogen sulfide and carbon dioxide are not a corrosion problem in streams with water dewpoint temperatures below the cold end temperature of the exchanger.

APPLICATIONS

Brazed aluminum heat exchangers are used in the following cryogenic natural gas processing applications:
- Ethane Plus Recovery
- Nitrogen Removal
- Helium Recovery
- Liquefied Natural Gas (LNG)

Within these applications, brazed aluminum heat exchangers are used for the following heat exchanger services:
- Gas to Gas Exchangers
- Demethanizer Reboilers
- Demethanizer Reflux Condensers
- Feed Gas Exchangers
- Product Heaters
- Propane Chillers

HARDWARE CAPABILITIES

Materials and Codes of Construction

Brazed aluminum heat exchangers are designed and constructed to comply with the “ASME Boiler and Pressure Vessel Code,” Section VIII, Division I, or other applicable standards. The aluminum alloys used comply with ASME Section II, Part B, “Nonferrous Materials,” or the requirements of the specified code authority.

Aluminum alloy 3003 is generally used for the parting sheets, corrugated fins, and bars which form the rectangular heat exchanger block. These parts are metallurgically bonded by a brazing process at temperatures of about 1100°F. The brazing alloy is an aluminum silicon metal and is provided on or with the parting sheets. Headers and nozzles are made from aluminum alloys 3003, 5054, 5083, 5086, 5454, or 6061-T6. Alloy 5083 is the most commonly used.

Maximum Working Temperature, Pressure, and Sizes

The maximum design temperature rating for brazed aluminum heat exchangers is typically 65°C; however, special designs are available for design temperatures up to 200°C. The minimum design temperature is –269°C.

ASME code approved brazed aluminum heat exchangers are available for pressure ratings from zero absolute to 9650 kPa (ga). Different design pressures can be used for each stream in the exchanger. The maximum core size available will vary with the maximum design pressure as shown in Fig. 9-35. Some size variation from Fig. 9-35 will occur depending on a particular manufacturer’s capabilities, specific design, and flow configuration. Batteries of exchangers are much larger and are limited in size by transportation capabilities.

Fins

Fins are available to cover a wide range of applications for a variety of heat transfer and pressure drop requirements at low, medium, and high pressure. The economic justification for using a particular fin type is unique for each application and is highly dependent on the cost of power relative to other considerations. Three major types of fins are shown in Fig. 9-36. These include plain (straight), serrated (lanced), and perforated. These and other more specialized fins can provide heat exchanger designs optimized for the best combination of heat transfer, pressure drop, compactness, and cost for a specific application.

Distributor and Passage Arrangements

There are a large number of distributor and passage arrangements available in brazed aluminum heat exchangers. Fin arrangements frequently used in gas processing applications are shown in Fig. 9-37 for a gas/gas exchanger. The “A” stream layers are shown with center distributors and provide for the residue gas to flow through the entire length of the heat exchanger. The “B & C” stream layers are arranged with side distributors and provide for each of the two high pressure feed gas streams to flow through only a portion of the overall heat exchanger length.
**Brazed Aluminum Heat Exchanger Specifications**

The design and specification of a brazed aluminum heat exchanger require thermodynamic and mechanical information.

**Thermodynamic** — Heat transfer duty, operating pressure and temperature, allowable pressure drop, flow rates, compositions, and the physical, thermodynamic, and transport properties of the fluids involved must be specified. A cooling (or load) curve should be supplied to the designer/manufacturer for two-phase applications, and it may be necessary for single phase streams operating over a wide temperature range.

**Mechanical** — Specifications should include information on applicable code authorities, design pressure and temperature, and requirements for connection size, type, and orientation. Exchanger support and package requirements should also be defined.

Fig. 9-38 is a sample manufacturer’s specification sheet. This document communicates the details of the heat exchanger design between the manufacturer and user. Lines 1-28 define the minimum information required from the user. Other required information includes turndown conditions, off-design conditions, and any other special operating conditions, if applicable. Using this information, the manufacturer will design the heat exchanger and provide the information in lines 29-51.

**Heat Load Curves**

Generation of the heat load curve, commonly called the cooling curve, from a temperature-duty table is an important first step in the analysis of any heat exchanger. It illustrates the intended heat exchange process and is used to define the required heat exchanger conductance (UA). A cooling curve also shows bubble and dewpoints, regions of phase change, and close temperature approaches.

Fig. 9-39 shows the temperature-duty data and cooling curve for a three-stream gas-to-gas heat exchanger. For multistream heat exchange services (more than two streams), the cooling curve can be reduced to a classic two-stream case for purposes of calculating and corrected mean temperature difference (CMTD) and the UA required. This is called the combined cooling curve assumption and is normally used for simple sizing calculations. Fig. 9-39 shows how a three-stream exchanger cooling curve is reduced to two streams by combining the cold residue gas stream duty with the cold recycle stream duty at points of constant temperature to form a combined cold stream.

The CMTD is approximated by calculating the log mean temperature difference (LMTD) on portions of the combined cooling curve called zones. The UA required in each zone is then calculated from the zone LMTD. At this point, it is possible to make individual heat exchanger sizings for each zone, or where more approximate sizings are acceptable, to make a heat exchanger sizing based on the combined zones using the CMTD. The CMTD is approximated by adding the UA's and Q's of each zone and dividing as shown in Fig. 9-39.

Selection of the precise number and location of zones is a matter of choice for the designer. However, proper selection of
**FIG. 9-38**  
Brazed Aluminum Heat Exchanger Specifications

<p>| | | | | | | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
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<th></th>
<th></th>
<th></th>
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<td>SERVICE GAS TO GAS EXCHANGER</td>
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<td>VENDOR</td>
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<td>– 77</td>
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<td>FLUID COND. OR VAPORIZED</td>
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<td>3835 (COND.)</td>
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<td>19</td>
<td>AVG. SP. HT.-VAP/LIQ.</td>
<td>kJ/m²</td>
<td>2.7 / 2.4</td>
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</tr>
<tr>
<td>20</td>
<td>AVG. TH. COND. VAP/LIQ.</td>
<td>W/m² °C</td>
<td>0.115 / 0.223</td>
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<td>21</td>
<td>DENSIBLE HEAT TRANSFERRED</td>
<td>W</td>
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<td>22</td>
<td>LATENT HEAT TRANSFERRED</td>
<td>W</td>
<td>500,000</td>
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<td>26</td>
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<td>33</td>
<td>CORE SPEC. WIDTH</td>
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<tr>
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<td>HEIGHT</td>
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<tr>
<td>35</td>
<td>LENGTH</td>
<td>14600 mm</td>
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<td>38</td>
<td>NUMBER OF PASSAGES</td>
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<td>39</td>
<td>EFFECTIVE PASSAGE WIDTH</td>
<td>mm</td>
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<td>7.1 x 0.40</td>
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<td>FIN SPACING PER INCH</td>
<td>17</td>
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<td>43</td>
<td>EFFECTIVE PASSAGE LENGTH</td>
<td>mm</td>
<td>4120</td>
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<td>HEAT TRANSFER SURFACE</td>
<td>m²</td>
<td>550</td>
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<td>m²</td>
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<tr>
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<td>600 ANSI</td>
<td></td>
<td></td>
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<td></td>
</tr>
<tr>
<td>50</td>
<td>WEIGHTS</td>
<td>kg</td>
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<td></td>
<td></td>
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<td>51</td>
<td>REMARKS</td>
<td></td>
<td></td>
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</tr>
</tbody>
</table>

Lines 1-28: Data needed by manufacturer from user for design purposes.
FIG. 9-39
Heat Load Curve for a Three Stream Exchanger

TEMPERATURE DUTY DATA

<table>
<thead>
<tr>
<th>T (°C)</th>
<th>DUTY (MW)</th>
<th>Xv</th>
<th>T (°C)</th>
<th>DUTY (MW)</th>
<th>Xv</th>
<th>T (°C)</th>
<th>DUTY (MW)</th>
<th>Xv</th>
</tr>
</thead>
<tbody>
<tr>
<td>48.9</td>
<td>0</td>
<td>1.0</td>
<td>45</td>
<td>0</td>
<td>1.0</td>
<td>45</td>
<td>0</td>
<td>1.0</td>
</tr>
<tr>
<td>35.6</td>
<td>0.171</td>
<td>1.0</td>
<td>18.1</td>
<td>0.241</td>
<td>1.0</td>
<td>18.1</td>
<td>0.115</td>
<td>1.0</td>
</tr>
<tr>
<td>22.2</td>
<td>0.345</td>
<td>1.0</td>
<td>-8.8</td>
<td>0.479</td>
<td>1.0</td>
<td>-8.8</td>
<td>0.229</td>
<td>1.0</td>
</tr>
<tr>
<td>8.9</td>
<td>0.523</td>
<td>1.0</td>
<td>-35.7</td>
<td>0.716</td>
<td>1.0</td>
<td>-35.7</td>
<td>0.344</td>
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<td>-4.4</td>
<td>0.734</td>
<td>0.977</td>
<td>-62.6</td>
<td>0.953</td>
<td>1.0</td>
<td>-62.6</td>
<td>0.460</td>
<td>1.0</td>
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<tr>
<td>-17.8</td>
<td>0.978</td>
<td>0.931</td>
<td>-76.7</td>
<td>1.084</td>
<td>1.0</td>
<td>-76.7</td>
<td>0.527</td>
<td>1.0</td>
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<tr>
<td>-31.1</td>
<td>1.239</td>
<td>0.879</td>
<td>-62.6</td>
<td>0.953</td>
<td>1.0</td>
<td>-62.6</td>
<td>0.460</td>
<td>1.0</td>
</tr>
<tr>
<td>-47.8</td>
<td>1.612</td>
<td>0.792</td>
<td>-62.6</td>
<td>0.953</td>
<td>1.0</td>
<td>-62.6</td>
<td>0.460</td>
<td>1.0</td>
</tr>
</tbody>
</table>

HEAT EXCHANGE ZONES

<table>
<thead>
<tr>
<th>ZONE</th>
<th>LMTD (°C)</th>
<th>DUTY (MW)</th>
<th>UA (MW/°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>3.48</td>
<td>0.29</td>
<td>0.0848</td>
</tr>
<tr>
<td>2</td>
<td>3.19</td>
<td>0.15</td>
<td>0.0459</td>
</tr>
<tr>
<td>3</td>
<td>3.73</td>
<td>0.15</td>
<td>0.0362</td>
</tr>
<tr>
<td>4</td>
<td>5.32</td>
<td>0.15</td>
<td>0.0276</td>
</tr>
<tr>
<td>5</td>
<td>10.1</td>
<td>0.37</td>
<td>0.0366</td>
</tr>
<tr>
<td>6</td>
<td>20.8</td>
<td>0.51</td>
<td>0.0245</td>
</tr>
<tr>
<td>TOTAL</td>
<td>CMTD</td>
<td>1.61</td>
<td>0.2586</td>
</tr>
</tbody>
</table>

\[
\text{CMTD} = \frac{Q}{UA} = \frac{1.61}{0.2586} = 6.23^\circ\text{C}
\]

Courtesy ALTEC International, Inc.
the number and location of zones will increase the accuracy of the estimate for the corrected mean temperature difference (CMTD) and the exchanger performance. The following are some useful guidelines for proper selection of a zone:

1. The temperature difference between the warm and cold composite streams must be essentially a linear function of duty; i.e., straight line.
2. The heat transfer coefficients in each stream must be nearly constant.
3. The properties of the fluids in each stream should be nearly constant. This is usually necessary in order for the heat transfer coefficient to remain essentially constant and for achieving reasonable accuracy in estimating stream pressure drop. This is particularly true for two-phase streams; therefore, special care should be taken in selecting zones when two-phase fluids are involved.

Heat leak in cryogenic heat exchangers is another factor which will affect the cooling curve. It acts as an unwanted heat flow into the heat exchange fluids and will reduce the CMTD. Heat leak and inaccuracies in the fluid thermodynamic data used to generate the cooling curve can significantly reduce the CMTD and increase the UA required for a particular process. For well insulated exchangers, heat leak normally has a negligible effect on the CMTD. However, the amount of heat leak should always be checked and combined as another warm stream on the cooling curve to determine its effect on the CMTD.

**Design Considerations for Two-Phase Flow**

Procedures for designing brazed aluminum heat exchangers for single-phase streams are well publicized by manufacturers and by Kays and London. The design of brazed aluminum heat exchangers for two-phase streams is not as well published.

Brazed aluminum heat exchangers are often used with two-phase streams in gas processing applications. Generally, condensing is performed vertically downward and vaporization vertically upward.

Pressure drop is usually evaluated with the Lockhart and Martinelli method which has been found to be reasonable for both vaporizing and condensing streams.

Since evaluation of the heat transfer coefficients for two-phase streams in brazed aluminum heat exchangers has not been well reported in the literature, most manufacturers use their own proprietary calculation procedures.

Two-phase (liquid-vapor) heat transfer in brazed aluminum heat exchangers is usually dominated by the forced convection mechanism. This convection mechanism tends to suppress any nucleate boiling.

The two-phase forced convection heat transfer coefficients for multi-component fluids are evaluated using the method described by Bell and Ghaly. This method provides for reducing the two-phase heat transfer coefficient to account for the mass transfer resistance that is a characteristic of multi-component heat transfer.

Fluid distribution is always an important consideration when designing high effectiveness heat exchangers. Special care must be taken to ensure that the fluids maintain homogeneous flow throughout the heat exchanger. This is especially important with two-phase streams, where fluid maldistribution can significantly reduce the performance of the heat exchanger. For this reason, special distributors are available for use with fluids which enter the exchanger in a two-phase condition. Fluid distribution for such an inlet condition is often handled by separating the vapor and liquid phases in a knock-out drum. The separated vapor and liquid phases are then distributed individually into the exchanger using conventional single-phase distributor arrangements. The vapor and liquid phases are then recombined inside the exchanger in the heat transfer zone. This method is the preferred arrangement for distributing fluids which enter the heat exchanger in a two-phase condition. However, sometimes alternative simple approaches to a two-phase inlet can be used. This choice depends on a thorough analysis of the effects of the potential vapor/liquid maldistribution on the cooling curve and performance of the exchanger.

Brazed aluminum heat exchangers are well suited for thermosyphon applications such as demethanizer reboilers and propane chillers. Hydraulic design considerations are the same as for shell and tube exchangers.

**Approximate Sizing Procedure**

The following is a quick and simple method for estimating the approximate size and performance of gas-to-gas exchangers and demethanizer reboilers used for ethane recovery. This short-cut method is applicable only for services condensing up to 30% (wt.) of feed gas or for reboilers vaporizing up to 20% (wt.) of feed liquid. For services outside these limits, a plate fin design specialist should be consulted.

The following sample problem illustrates the approximate sizing procedure for a gas-to-gas exchanger. Typically, this application involves a warm feed gas operating at pressures between 3500 kPa (abs) and 7600 kPa (abs), which is cooled from...
above 38°C to below –73°C, and will partially condense up to 30% of its mass. The refrigeration is supplied by cold residue or recycle gas streams which normally operate at between 700 and 2100 kPa (abs). Warm end temperature approaches for this exchanger are typically designed from 3-6°C. This sample problem is the same example used in the heat load curve (Fig. 9-39) and in the brazed aluminum heat exchanger specification (Fig. 9-38) for an optimized selection. The results of the rough selection agree well with the optimized selection. Lines 1 through 28 of Fig. 9-38 are given data, provided by the purchaser.  

**Example 9-4** — For purposes of simplifying this quick selection procedure, the following are assumed:  

A fouling factor of 0.001 is included on each stream.  

The CMTD was calculated assuming that heat leak was negligible, as per the example of Fig. 9-39.  

The estimation of heat exchanger size by this quick procedure is reduced to a single-zone calculation. Precise selections by the manufacturers normally involve a multizone analysis of the heat exchanger.  

**Solution Steps**  

Step 1 — Determine Exchanger Cross Section  

From Fig. 9-40, select the typical mass velocities (G) for each stream based on their respective operating pressures:  

Feed Gas [5584.8 kPa (abs)]  

\[ G_H = 90.9 \text{ kg} / (\text{m}^2 \cdot \text{s}) \]  

Residue Gas ([1413.4 kPa (abs)])  

\[ G_C = 47.4 \text{ kg} / (\text{m}^2 \cdot \text{s}) \]  

Recycle Gas 1965.0 kPa (abs)  

\[ G_R = 56.7 \text{ kg} / (\text{m}^2 \cdot \text{s}) \]  

Using these G values, the exchanger cross section can be computed from the following equations:  

For gas to gas exchangers (where feed gases condense up to 20% of their mass):  

\[ L = \frac{66500 \text{ UA}}{WHN \sqrt{G_{\text{Total}}}} + 0.65W \]  

Eq. 9-17  

Where \( G_{\text{Total}} = (G_H)_{\text{Avg}} + (G_C)_{\text{Avg}} \)  

For serrated fin exchangers —  

\[ L = \frac{99750 \text{ UA}}{WHN \sqrt{G_{\text{Total}}}} + 0.65W \]  

Eq. 9-18  

For perforated fin exchangers —  

\[ L = \frac{44300 \text{ UA}}{WHN \sqrt{G_{\text{H}}}} + 0.65W \]  

Eq. 9-19  

Step 2 — Determine Exchanger Length  

First calculate the required UA from the heat load curve (see Fig. 9-39):  

\[ UA = UA_{\text{zone 1}} + UA_{\text{zone 2}} + \ldots + UA_{\text{zone 6}} = \frac{Q}{\text{CMTD}} \]  

\[ = \frac{1620000 \text{ W}}{6.23^\circ \text{C}} \]  

\[ = 260000 \text{ W}^\circ \text{C} \]  

From one of the following equations, depending on fin preference, calculate the exchanger length. When both perforated and serrated fins are used in the heat exchanger, use the average value obtained from Equations 9-17 and 9-18 (or Equations 9-20 and 9-21 for demethanizer reboilers). Serrated fins are high performance and yield shorter exchanger lengths with higher stream pressure drops. Perforated fins are lower performers and will yield longer exchangers with lower pressure drops. Plain fins are lowest performers and are normally only used in distributors, due to their low pressure drop characteristics. Normally, serrated fins provide the most optimum selection, unless pressure drop/operating cost is the controlling parameter.  

For gas to gas exchangers (where feed gases condense up to 30% of their mass):  

For serrated fin exchangers —  

\[ L = \frac{44300 \text{ UA}}{WHN \sqrt{G_{\text{H}}}} + 0.65W \]  

Eq. 9-20  

For perforated fin exchangers —  

\[ L = \frac{66500 \text{ UA}}{WHN \sqrt{G_{\text{H}}}} + 0.65W \]  

Eq. 9-21  

For one gas to gas exchanger, 635 mm (W) x 564 mm (H), using serrated fins, and a UA required of 260 000 W/°C, the required length is:  

\[ L = \frac{(68500)(260000)}{(635)(563.5)(1) \sqrt{90.9 + (47.4 + 56.7)/2}} + 0.65(635) \]  

\[ L = 4570 \text{ mm} \]  

The heat exchanger size is now established as:  

One exchanger, 635 x 564 x 4570 mm.  

If exchanger length is too long for packaging and/or transportation considerations, lower the mass velocities (G) and return to Step 1. This will increase the core cross section and decrease the core length.  

Step 3 — Check Stream Pressure Drops  

The last step in rough sizing the heat exchanger is to verify that pressure drop for the exchanger size selected is within
allowable levels for all streams. Pressure drop can vary widely depending on type and size of distributors chosen, the amount of phase change in two-phase streams, and other factors. The following equations will yield approximate stream frictional pressure drop which includes the heat transfer zone, distributors, and nozzles.

For Serrated Fin Exchangers:

For vapor streams —

\[
\Delta P = \frac{(1.1) \times (10^{-4}) \times (L + 510) \times (G)^{1.8}}{\rho_m} \quad \text{Eq 9-22}
\]

For partially condensing streams (up to 30% of mass condensed) —

\[
\Delta P = \frac{(2.2) \times (10^{-4}) \times (L + 1020) \times (G)^{1.8}}{\rho_m} \quad \text{Eq 9-23}
\]

For Perforated Fin Exchangers:

For vapor streams —

\[
\Delta P = \frac{(2.4) \times (10^{-4}) \times (L + 2415) \times (G)^{1.8}}{\rho_m} \quad \text{Eq 9-24}
\]

For partially condensing streams (up to 30% of mass condensed) —

\[
\Delta P = \frac{(4.8) \times (10^{-5}) \times (L + 4830) \times (G)^{1.8}}{\rho_m} \quad \text{Eq 9-25}
\]

where \( \rho_m = \frac{2(\rho_m)(\rho_{out})}{\rho_m + \rho_{out}} \) \quad \text{Eq 9-26}

And when inlet or outlet conditions are two-phase:

\[
\rho_{2\phi} = \frac{1}{1 + \frac{X_V}{X_L}} \quad \text{Eq 9-27}
\]

Pressure drops for vaporizing liquids are not easily approximated by these rough estimating procedures. However, for these applications, the demethanizer liquid pressure drop will normally be within allowable levels when the exchanger is selected for feed gas mass velocities recommended in Fig. 9-40.

For example problem 9-4:

Feed Gas:

\[
\rho_{out} = \rho_{2\phi} = \left(1 \frac{0.792}{0.208} \frac{77.5}{448.5}\right) = 93.6 \text{ kg/m}^3
\]

\[
\rho_m = \frac{2(44.7)(93.6)}{44.7 + 93.6} = 60.5
\]

\[
\Delta P = \frac{(2.2) \times (10^{-4}) \times (4570 + 1020) \times (90.7)^{1.8}}{60.5}
\]

\[
= 67.9 \text{ kPa (vs. 69 kPa allowed)}
\]

Residue Gas:

\[
\rho_m = \frac{2(16.3)(8.97)}{16.3 + 8.97} = 11.6 \text{ kg / m}^3
\]

\[
\Delta P = \frac{(1.1) \times (10^{-4}) \times (4570 + 510) \times (47.4)^{1.8}}{11.6}
\]

\[
= 50.0 \text{ kPa (vs. 48 kPa allowed)}
\]

Recycle Gas:

\[
\rho_m = \frac{2(21.1)(12.8)}{21.1 + 12.8} = 15.9 \text{ kg/m}^3
\]

\[
\Delta P = \frac{(1.1) \times (10^{-4}) \times (4570 + 510) \times (56.7)^{1.8}}{15.9}
\]

\[
= 50.4 \text{ kPa (vs. 48 kPa allowed)}
\]

For selections requiring more than one heat exchanger, the pressure drop for the manifold piping which interconnects the individual heat exchangers can be estimated according to the method developed by F. A. Zenz, and must be added to the pressure drops for the individual exchanger(s) calculated by the above procedures to arrive at the total unit pressure drop.

For the single core Example 9-4, this is not required, and the pressure drops are only slightly over allowable. If pressure drops were too excessive, it would be necessary to return to Step 1 (Equation 9-15) and to lower the mass velocities (G). This has the effect of increasing exchanger cross section until the desired pressure drop is achieved. Use the following equation for approximating a new mass velocity which will yield the allowable pressure drop.

\[
G_{\text{new}} = G_{\text{old}} \left[ \frac{\Delta P_{\text{allowed}}}{\Delta P_{\text{old}}} \right]^{0.56} \quad \text{Eq 9-28}
\]

The above sizing procedure should produce estimates of exchanger size which will be within plus or minus 15% and pressure drops which should be within plus or minus 25% of the final design.

**INSTALLATION-OPERATION-MAINTENANCE**

**Mounting**

Brazed aluminum heat exchangers are normally installed in a vertical orientation with the operational cold end down, and are supported with either aluminum support angles or an aluminum pedestal base supplied by the heat exchanger manufacturer. This type of support system is the most common for mounting the exchanger to steel framework or onto a platform. Other orientations of the heat exchangers and other support systems are sometimes permissible, but only when designed for special service by the manufacturer.

External loads on the heat exchanger can be imposed through the connecting piping due to mechanical or thermal loading or both. All support systems should be designed to minimize these loads and their effect on the heat exchanger.

This is accomplished by providing sufficient pipe flexibility and by providing allowance for movement at the heat exchanger support member by using slotted bolt holes and bolts that are only finger tight. All support systems should be additionally safeguarded by use of sway bracing on the end of the exchanger opposite the main support system whenever the total external pipe loads on the exchanger will produce reaction forces at the main support members which exceed the actual weight of the exchanger.

**Insulation**

Since the exchangers are usually operating at cryogenic temperatures, highly efficient insulation is required to minimize heat leak. Typically, the exchanger is mounted in a cold box which is filled with perlite or rock wool. When the exchanger is not mounted inside a cold box, its exterior is nor-
mally insulated with rigid polyurethane foam. An alternative is Foamglas® insulation. These insulations are positioned and fastened around the exchanger and covered with a vapor barrier. Protective metal coverings or flashing can be used for this purpose. Some form of insulation (such as micarta spacers) should be used between the heat exchanger support member and the supporting beam or platform.

Field Testing and Repair

Maximum working pressures and temperatures are always specified on the manufacturer’s nameplate. These values should not be exceeded during field testing or operation. Since it is extremely difficult to dry brazed aluminum heat exchangers in the field, only a clean dry gas should be used for leak testing.

Internal leaks in a brazed aluminum heat exchanger are generally indicated by a change of purity in any of the fluid streams. External leaks can be determined by sight, smell, audible sounds of leaking fluid, external gas monitoring equipment, or localized cold spots appearing on the external insulation. External leaks in exchangers mounted in a cold box are generally indicated by excessive venting through the cold box breather valves.

Several tests are available for locating external or internal leaks. An air-soap test is effective for locating external leaks. An air test with soap applied to nozzle connections or a nitrogen-freon test can be used to identify the streams involved in an internal or cross pass leak. Internal and external leaks usually can be repaired by blocking layers, making localized external welds, etc. Qualified manufacturer’s representatives are usually required to establish the exact location of an internal leak and to make any repairs.

Hydrate Suppression

During start-up, upset, or even normal operating conditions, hydrates and/or heavy hydrocarbons may freeze out and block sections of the heat exchanger.

Injection sparge systems (see Fig. 9-41) are designed for injecting either methanol or glycol into the feed gas entering the exchanger. This method of hydrate suppression has proven effective.

Cleaning

Only clean, dry fluids which are non-corrosive to aluminum should be used in brazed aluminum heat exchangers. The presence of particulates in the fluid resulting from start-up or mal-operation may not only lead to exchanger fouling but may also cause erosion in the high velocity areas of the exchanger. This can be prevented with proper filtering (177 micron screen-80 mesh Tyler standard, or finer) upstream of the brazed aluminum heat exchanger. A heavy duty, cleanable filter or strainer is strongly recommended on the inlet of all streams entering the exchanger.

Fouling which is caused by hydrate formation can be removed by warming the exchanger to ambient conditions. Deposits of heavy hydrocarbons, waxy materials, or compressor oils can be removed by a combination of warming and a solvent rinse. Solvents such as trichloroethane, toluene, or propylene are effective.

If plugging occurs, reverse gas flow, called puffing, is an effective method of removing particulate matter such as adsorbents, pipe scale, sand, or other solid debris. It involves the use of a calibrated rupture disk on the inlet nozzle of the plugged stream and one or more charges and ruptures to es-

![Typical Methanol or Glycol Injection Sparge System](image1)

![Plate and Frame Heat Exchanger](image2)
exposure of personnel or equipment to explosive or toxic fluids

Plate Frame Heat Exchangers

A typical plate and frame heat exchanger “PHE” (sometimes referred to as a gasketed plate heat exchanger) is shown in an exploded view in Fig. 9-42. The PHE consists of an arrangement of gasketed pressed metal plates (heat transfer surface), aligned on two carrying bars, secured between two covers by compression bolts. Inlet and outlet ports for both hot and cold fluids are stamped into the corners of each plate. The ports are lined up to form distribution headers through the plate pack. All four fluid connections are usually located in the fixed end cover. This permits opening the exchanger without disconnecting any piping. Plates can be added and removed in the field should service requirements change. The plates are pressed into one of a number of available patterns and may be constructed of any material which can be cold formed to the desired pattern. The welding characteristics of the plate material are not of prime importance since very little or no welding is involved in plate construction.

Gasket grooves are pressed into the plates as they are formed. The gaskets are generally made of elastomers such as natural rubber, nitrile, butyl, neoprene, etc. The gasket material chosen depends on the temperature, pressure, and chemical characteristics of the fluid to which it will be exposed. The gasket cross-section varies with different plate designs and sizes. Rectangular, trapezoidal, or oval cross-sections are the most common. The width is generally 5-15 mm, depending on spacing. The height of the gasket before it is compressed is 15 to 50% higher than the spacing, depending on material, cross-section of gasket, gasket track, and gasket hardness. When the plate stack is compressed, the exposed surface of the gasket is very small. The gaskets are generally arranged in such a way that the through pass portal is sealed independently of the boundary gasket. Leaks from one fluid to the other cannot take place unless a plate develops a hole. Any leakage from the gaskets is to the outside of the exchanger where it is easily detected.

Since the plates are generally designed to form channels giving highly turbulent flow, the PHE produces higher heat transfer coefficients for liquid flow than most other types. The high heat transfer coefficients are developed through the efficient use of pressure drop.

Advantages

The PHE has the following advantages over conventional shell and tube heat exchangers:

1. It can easily be disassembled for cleaning.
2. The plates can be rearranged, added to, or removed from the plate rack for difference service conditions.
3. The fluid residence time is short (low fluid volume to surface area ratio).
4. No hot or cold spots exist which could damage temperature sensitive fluids.
5. Fluid leakage between streams cannot occur unless plate material fails.
6. Fluid leakage due to a defective or damaged gasket is external and easily detected.
7. Low fouling is encountered due to the high turbulence created by the plates.
8. A very small plot area is required relative to a shell and tube type heat exchanger for the same service.
9. The maintenance service area required is within the frame size of the exchanger.

Disadvantages

1. Care must be taken by maintenance personnel to prevent damage to the gaskets during disassembly, cleaning, and reassembly.
2. A relatively low upper design temperature limitation exists.
3. A relatively low upper design pressure limitation exists.
4. Gasket materials are not compatible with all fluids.

Applications

The PHE is normally used in liquid services. This type of heat exchanger is considered to be a high heat transfer, high pressure drop device, but it can be used for services requiring a low pressure drop with the associated reduction in heat transfer coefficients.

Since the plates are thin, the PHE gives a relatively high heat transfer coefficient for the mass of material required. When alloy materials are required, the PHE is competitive with more conventional heat exchanger designs.

Materials of Construction

The frames are usually fabricated from carbon steel while the tension bolts are high tensile strength steel. Common plate materials include 304 and 316 stainless steel, titanium, In-

Typical Gasket Material Temperature Limitations

<table>
<thead>
<tr>
<th>Gasket Material</th>
<th>Temperature Limitation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Natural rubber, styrene, neoprene</td>
<td>70°C</td>
</tr>
<tr>
<td>Nitrile</td>
<td>100°C</td>
</tr>
<tr>
<td>Resin-cured butyl, viton</td>
<td>150°C</td>
</tr>
<tr>
<td>Ethylene/propylene, silicone</td>
<td>150°C</td>
</tr>
<tr>
<td>Compressed non-asbestos fiber</td>
<td>200°C</td>
</tr>
</tbody>
</table>

Typical Fouling Factors for PHEs

<table>
<thead>
<tr>
<th>Fluid</th>
<th>Fouling Factor K * m²/W</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water</td>
<td>0.000 002</td>
</tr>
<tr>
<td>Demineralized or distilled</td>
<td>0.000 004</td>
</tr>
<tr>
<td>Municipal supply (soft)</td>
<td>0.000 009</td>
</tr>
<tr>
<td>Municipal supply (hard)</td>
<td>0.000 007</td>
</tr>
<tr>
<td>Cooling tower (treated)</td>
<td>0.000 009</td>
</tr>
<tr>
<td>Sea (coastal) or estuary</td>
<td>0.000 005</td>
</tr>
<tr>
<td>Sea (ocean)</td>
<td>0.000 008</td>
</tr>
<tr>
<td>River, canal, borehole, etc.</td>
<td>0.000 009</td>
</tr>
<tr>
<td>Engine jacket</td>
<td>0.000 01</td>
</tr>
<tr>
<td>Oils, lubricating</td>
<td>0.000 004 to 0.000 009</td>
</tr>
<tr>
<td>Solvents, organic</td>
<td>0.000 002 to 0.000 005</td>
</tr>
<tr>
<td>Steam</td>
<td>0.000 002</td>
</tr>
<tr>
<td>Process fluids, general</td>
<td>0.000 002 to 0.000 01</td>
</tr>
</tbody>
</table>
coloy 825®, Hastelloy®, aluminum-bronze, tantalum, copper-nickel, aluminum, and palladium stabilized titanium. The PHE can be fabricated and stamped to the ASME Code.

**Maximum Pressure and Temperature Ratings**

Maximum allowable working pressure may be determined by frame strength, gasket retainment, or plate deformation resistance. Of these, it is often the frame that limits operating pressure, so that many manufacturers produce a low cost frame for low pressure duties (typically up to 85 psig) and a more substantial frame for higher pressures, for the same plate size. Maximum operating pressure for a typical PHE is normally 1000-1600 kPa (ga), but some types are designed to operate over 2100 kPa.

Normally, it is gaskets that limit the maximum operating temperature for a plate heat exchanger. Fig. 9-43 provides typical temperature limitations for common gasket materials not subject to chemical attack.

**Size Limitations**

The surface area per plate and number of plates per frame varies depending upon the manufacturer. Typically, surface area ranges between 0.04 and 4 m² per plate. Frame sizes have been manufactured to contain up to 600 plates. When larger surface areas are required, multiple units are supplied.

**Fouling Factors**

Fouling factors required in the PHE are small compared with those commonly used in shell and tube designs for the following reasons:

1. High turbulence maintains solids in suspension.
2. Heat transfer surfaces are smooth. For some applications, a mirror finish may be available.

**Printed Circuit Heat Exchangers**

**General**

Printed Circuit Heat Exchangers (PCHEs) are highly compact, corrosion resistant heat exchangers capable of operating at pressures of several hundred atmospheres and temperatures ranging from cryogenic to several hundred degrees Celsius.

PCHEs are constructed from flat alloy plates which have fluid flow passages photo-chemically machined (etched) into them. This process is similar to manufacturing electronic printed circuit boards, and gives rise to the name of the exchangers. In the case of PCHEs, it is fluid circuits which are formed by etching.

Stacks of etched plates, carrying flow passage designs tailored for each fluid, are interleaved and diffusion bonded together into solid blocks. Diffusion bonding is a solid state welding process in which the flat and clean metal surfaces are held together at high temperatures, resulting in interfacial crystal growth between the touching surfaces which gives rise to a bond strength equal to that of the bulk metal.

The thermal capacity of the exchanger is built to the required level by welding together diffusion bonded blocks to form the complete heat exchange core. Headers and nozzles are welded on to the core in order to direct the fluids to the appropriate sets of passages. Fig. 9-45 shows the complete construction of a two-fluid exchanger.

PCHEs are all-welded — there is no braze material employed in construction, and no gaskets are required. Hence the potential for leakage and fluid-compatibility is reduced. In fact the high level of constructional integrity renders PCHEs exceptionally well suited to critical high pressure applications, such as high-pressure gas exchangers on offshore platforms.

The thermal design of PCHEs is subject to very few constraints. Fluids may be liquid, gas or two-phase, and they can exchange heat in countercflow, crossflow or coflow at any required pressure drop. Where energy is expensive, high heat exchange effectiveness can be achieved through very close temperature approaches in countercflow. To simplify control, or to further maximize energy efficiency, more than two fluids can exchange heat in a single core. Heat loads can vary from a few watts to many megawatts, in exchangers weighing from a few kilograms to thousands of kilograms.

Mechanical design is also flexible. Etching patterns can be adjusted to provide high pressure containment where required — design pressures may be several hundred atmospheres. The all-welded construction is compatible with very high temperature operation, and the use of austenitic stainless steels allows cryogenic application. It is worthy of note that vibration is absent from PCHEs, as this can be an important source of failure in shell-and-tube exchangers.
Materials of construction include stainless steel and titanium as standard, with nickel and nickel alloys also commonly used.

Passages are typically of the order of 2 mm semi-circular cross-section — that is, 2 mm across and 1 mm deep — for reasonably clean applications, although there is no absolute limit on passage size. The corrosion resistant materials of construction for PCHEs, the high wall shear stresses, and the absence of dead-spots assist in resisting fouling deposition.

Prime heat transfer surface densities, expressed in terms of effective heat transfer area per unit volume, can be up to 2500 m²/m³. This is higher than prime surface densities in gasketed plate exchangers, and an order of magnitude higher than normal prime surface densities in shell-and-tube exchangers.

**Design**

Detailed thermal design of PCHEs is supported by proprietary design software developed by the manufacturers which allows for infinite geometric variation to passage arrangements in design optimization. Variations to passage geometry have negligible production cost impact since the only tooling required for each variation is a photographic transparency for the photo-chemical machining process.

Although the scope of PCHE capabilities is much wider, as a sizing guide it is safe to assume that channel patterns can be developed to mimic any y- and f-factors characteristics found in publications such as "Compact Heat Exchangers" by Kays and London for aluminum surfaces, or data presented by gsketed plate exchanger manufacturers.

It is rarely necessary to apply a correction factor substantially less than 1 to the LMTD calculated for an exchange, no matter how high the effectiveness required, because of the PCHE counterflow capabilities. Pressure drops can be specified at will; however, as with all heat exchangers, lower allowable pressure drops will result in lower heat transfer coefficients and hence larger exchangers.

**Applications**

PCHEs extend the benefits of compact exchangers into applications where pressure, temperature or corrosivity prevent the use of conventional plate exchangers.

In hydrocarbons processing, PCHEs are employed with gas streams in such areas as:

- compression aftercooling,
- gas/gas counterflow exchange for dewpoint control,
- cryogenic inerts removal and
- liquefaction.

The use of multi-stream contact in these duties is common. In refineries, suitable applications are to be found in light ends processing and feed-effluent exchange for platforming and HDS units.

Chemicals applications include duties requiring:

- high pressure capability, such as ammonia and methanol production,
- corrosion resistant materials, such as pure nickel for caustic soda and titanium for chloride environments, and
- high effectiveness counterflow contact, including heat recovery.

In power production, PCHEs are applicable as feedwater heaters, fuel gas heaters, water/water exchangers and in various roles in non-conventional power production systems such as geothermal and solar.

**REFERENCES**


**BIBLIOGRAPHY**


