Heat Exchanger Data Book

CHC603 Heat Transfer Operation – II
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Part I

Formulae, graphs and tables
Chapter 1

Shell and Tube Heat Exchanger

1.1 Log Mean Temperature difference

For counter current flow,

\[ \Delta T_{lm} = \frac{(T_{hin} - T_{cout}) - (T_{hout} - T_{cin})}{\ln \left( \frac{T_{hin} - T_{cout}}{T_{hout} - T_{cin}} \right)} \]  

\[ \Delta T_m = F_t \Delta T_{lm} \]  

1.1.1 Correction factor

For one shell pass and two or more even tube passes shell and tube heat exchanger,

\[ F_t = \frac{\sqrt{(R^2 + 1)\ln \left[ \frac{1 - S}{1 - RS} \right]}}{(R - 1)\ln \left[ \frac{2 - S \left( R + 1 - \sqrt{R^2 + 1} \right)}{2 - S \left( R + 1 + \sqrt{R^2 + 1} \right)} \right]} \]  

where,

\[ R = \frac{T_{in} - T_{out}}{t_{out} - t_{in}} \]  
\[ S = \frac{t_{out} - t_{in}}{T_{in} - t_{in}} \]  

\( T \) = Shell side temperature  
\( t \) = Tube side temperature  
\( T_h \) = hot fluid temperature  
\( T_c \) = cold fluid temperature
LMTD Correction Factor (1 shell pass; 2 or more tube passes)

\[
P = \frac{b_2 - t_1}{T_1 - t_1}
\]

\[
F = \frac{\text{LMTD}}{\text{CMTD}} = (\text{LMTD})/(F)
\]

LMTD Correction Factor

<table>
<thead>
<tr>
<th>1 shell pass</th>
<th>2 or more tube passes</th>
</tr>
</thead>
<tbody>
<tr>
<td>( P )</td>
<td>( F )</td>
</tr>
</tbody>
</table>

LMTD Correction Factor (2 shell passes; 4 or more tube passes)

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

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

\[
\text{CMTD} = (\text{LMTD})/(F)
\]

LMTD Correction Factor

<table>
<thead>
<tr>
<th>2 shell passes</th>
<th>4 or more tube passes</th>
</tr>
</thead>
<tbody>
<tr>
<td>( P )</td>
<td>( R )</td>
</tr>
</tbody>
</table>

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

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

\[
\text{CMTD} = (\text{LMTD})/(F)
\]
LMTD Correction Factor (3 shell passes; 6 or more tube passes)

\[ P = \frac{T_2 - T_1}{T_P - T_1} \]

\[ LMTD = \frac{(T_2 - T_1)}{(T_P - T_1)} \]

CMTD = (LMTD) \((F_3)\)

LMTD Correction Factor (4 shell passes; 8 or more tube passes)

\[ P = \frac{T_2 - T_1}{T_1 - T_1} \]

\[ R = \frac{T_1 - T_2}{T_P - T_1} \]

CMTD = (LMTD) \((F_4)\)
1.2 Heat transfer through tubes

Seider-Tate and Hausen equations,

for $Re \geq 10^4$

$$Nu = 0.023 Re^{0.8} Pr^{1/3} (\mu/\mu_w)^{0.14}$$  \hspace{1cm} (1.6)

for $2100 < Re \geq 10^4$

$$Nu = 0.116 [Re^{2/3} - 125] Pr^{1/3} (\mu/\mu_w)^{0.14} \left[ 1 - (D/L)^{2/3} \right]$$  \hspace{1cm} (1.7)

for $Re \leq 2100$

$$Nu = 1.86 [Re Pr (D/L)]^{1/3} (\mu/\mu_w)^{0.14}$$  \hspace{1cm} (1.8)

1.3 Shell-side heat-transfer coefficient (Kern’s Method)

$$j_H = 0.5 (1 + l_B/D_s) \left( 0.08 Re^{0.6821} + 0.7 Re^{0.1772} \right)$$  \hspace{1cm} (1.9)

where,

$$j_H = \frac{h_o d_e}{k} Pr^{-1/3} (\mu/\mu_w)^{-0.14}$$  \hspace{1cm} (1.10)

$l_B$ = baffle spacing  
$D_s$ = shell ID  
$d_e$ = equivalent diameter
\[ Re = \frac{G_s d_e}{\mu} \]  
(1.11)
\[ G_s = \frac{\dot{m}_s}{A_s} \]  
(1.12)
\[ A_s = \frac{(p_t - d_o) D_s l_B}{p_t} \]  
(1.13)

for a square pitch arrangement:
\[ d_e = \frac{1.27}{d_o} \left( p_t^2 - 0.785 d_o^2 \right) \]  
(1.14)

for an equilateral triangular pitch arrangement:
\[ d_e = \frac{1.10}{d_o} \left( p_t^2 - 0.917 d_o^2 \right) \]  
(1.15)

### 1.4 Bell’s method for HTC shell side

\[ h_s = h_{oc} F_n F_w F_b F_L \]  
(1.16)

- \( h_{oc} \) = heat transfer coefficient calculated for cross-flow over an ideal tube bank, no leakage or bypassing.
- \( F_n \) = correction factor to allow for the effect of the number of vertical tube rows,
- \( F_w \) = window effect correction factor,
- \( F_b \) = bypass stream correction factor,
- \( F_L \) = leakage correction factor.

#### 1.4.1 HTC cross flow

See section – (1.3)
\[ \frac{h_{oc} d_o}{k_f} = j_H P r^{1/3} \left( \frac{\mu}{\mu_w} \right)^{0.14} \]  
(1.17)

#### 1.4.2 \( F_n \), tube row correction factor

1. \( Re > 2000 \), turbulent; take \( F_n \) from Figure 12.32.
2. \( Re > 100 \) to 2000, transition region, take \( F_n = 1.0 \);
3. \( Re < 100 \), laminar region, \( F_n \propto (N_c')^{-0.18} \)
   where \( N_c' \) is the number of rows crossed in series from end to end of the shell.

#### 1.4.3 \( F_w \), window correction factor

The correction factor is shown in Figure 12.33 plotted versus \( R_w \), the ratio of the number of tubes in the window zones to the total number in the bundle. For \( R_w \) refer section – 1.5.8
1.4.4 $F_b$, bypass correction factor

$$F_b = exp \left(-\alpha \frac{A_b}{A_s} \left(1 - \left(\frac{2N_s}{N_{cv}}\right)^{1/3}\right)\right)$$

(1.18)

where,

$\alpha = 1.5$ for laminar flow, $Re < 100$,

$\alpha = 1.35$ for transitional and turbulent flow $Re > 100$,

$A_b$ = clearance area between the bundle and the shell, refer equation – (1.35),

$A_s$ = maximum area for cross-flow,

$N_s$ = number of sealing strips encountered by the bypass stream in the cross-flow zone,

$N_{cv}$ = the number of constrictions, tube rows, encountered in the cross-flow section.

1.4.5 $F_L$, Leakage correction factor

$$F_L = 1 - \beta_L \left[\frac{A_{tb} + 2A_{sb}}{A_L}\right]$$

(1.19)

where,

$\beta_L$ = a factor obtained from Figure 12.35,

$A_{tb}$ = the tube to baffle clearance area, per baffle,

$A_{sb}$ = shell-to-baffle clearance area, per baffle,

$A_L$ = total leakage area = ($A_{tb} + A_{sb}$).

1.5 Pressure Drop in shell

1.5.1 Cross-flow zones

$$\Delta P_c = \Delta P_i F_b' F_L'$$

(1.20)
1.5.2 $\Delta P_i$ ideal tube bank pressure drop

$$\Delta P_i = 8j_t N_{cv} \frac{\rho u_s^2}{2} \left( \frac{\mu}{\mu_w} \right)^{-0.14}$$  \hspace{1cm} (1.21)

where,

- $N_{cv}$ = number of tube rows crossed (in the cross-flow region),
- $u_s$ = shell side velocity, based on the clearance area at the bundle equator,
- $j_t$ = friction factor obtained from Figure 12.36, at the appropriate Reynolds number, $Re = (\rho u_s d_o/\mu)$.

1.5.3 $F'_b$, bypass correction factor for pressure drop

The correction factor is calculated from the equation used to calculate the bypass correction factor for heat transfer, equation – (1.18) but with the following values for the constant $\alpha$,

where,

- $\alpha = 5$ for laminar flow, $Re < 100$,
- $\alpha = 4$ for transitional and turbulent flow $Re > 100$

The correction factor for exchangers without sealing strips is shown in Figure 12.37.
Figure 12.34. Bypass correction factor
1.5.4 $F'_L$, leakage factor for pressure drop

The factor is calculated using the equation for the heat-transfer leakage-correction factor, with the values for the coefficient $\beta_L$ taken from Figure 12.38.

1.5.5 Window-zone pressure drop

$$\Delta P_w = F'_L \left(2 + 0.6N_{wv}\right) \frac{\rho u_{z}^2}{2}$$

(1.22)

where

- $u_z$ = the geometric mean velocity, $u_z = \sqrt{u_wu_s}$
- $u_w$ = the velocity in the window zone, based on the window area less the area occupied by the tubes, $u_w = \frac{W_s}{A_{w\rho}}$
- $W_s$ = shell-side fluid mass flow, kg/s,
- $N_{wv}$ = number of restrictions for cross-flow in window zone, approximately equal to the number of tube rows.

1.5.6 End zone pressure drop

$$\Delta P_e = \Delta P_i \left[\frac{N_{wv} + N_{cv}}{N_{cv}}\right] F'_b$$

(1.23)
Figure 12.37. Bypass factor for pressure drop $F'_b$

Figure 12.38. Coefficient for $F'_L$, pressure drop
### 1.5.7 Total shell-side pressure drop

\[ \Delta P_s = 2\Delta P_e + \Delta P_c (N_b - 1) + N_b \Delta P_w \]  

(1.24)

where, \( N_b = \frac{L}{l_B} - 1 \)

\( L \) = Total tube length,

\( l_B \) = baffle spacing

### 1.5.8 Shell and bundle geometry

**Bundle diameter**

\[ D_b = d_o \left( \frac{N_t}{K_1} \right)^{1/n_1} \]  

(1.25)

where

\( N_t \) = number of tubes,

\( D_b \) = bundle diameter, mm,

\( d_o \) = tube outside diameter, mm.

<table>
<thead>
<tr>
<th>Triangular pitch, ( P_T = 1.25d_o )</th>
</tr>
</thead>
<tbody>
<tr>
<td>No. of passes</td>
</tr>
<tr>
<td>( K_1 )</td>
</tr>
<tr>
<td>( n_1 )</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Square pitch, ( P_T = 1.25d_o )</th>
</tr>
</thead>
<tbody>
<tr>
<td>No. of passes</td>
</tr>
<tr>
<td>( K_1 )</td>
</tr>
<tr>
<td>( n_1 )</td>
</tr>
</tbody>
</table>

\[ H_b = \frac{D_b}{2} - D_s (0.5 - B_c) \]  

(1.26)

\[ N_{cv} = \frac{D_b - 2H_b}{p_t'} \]  

(1.27)

\[ N_{wv} = \frac{H_b}{p_t'} \]  

(1.28)

\( H_b \) = baffle cut height = \( D_s \times B_c \), where \( B_c \) is the baffle cut as a fraction,

\( H_b \) = height from the baffle chord to the top of the tube bundle,

\( B_b \) = bundle cut = \( H_b/D_b \),

\( \theta_b \) = angle subtended by the baffle chord, rads,

\( D_b \) = bundle diameter.

\( D_s \) = Shell ID.

\( p_t' = p_t \) for square pitch,

\( p_t' = 0.87p_t \) for equilateral triangular pitch.
The number of tubes in a window zone \( N_w \) is given by:

\[
N_w = N_t \times R'_a
\]  

(1.29)

where, \( R'_a \) is the ratio of the bundle cross-sectional area in the window zone to the total bundle cross-sectional area, \( R'_a \) can be obtained from Figure 12.41, for the appropriate bundle cut, \( B_b \).

The number of tubes in a cross-flow zone \( N_c \) is given by

\[
N_c = N_t - 2N_w
\]  

(1.30)

\[
R_w = \frac{2N_w}{N_t}
\]  

(1.31)

\[
A_w = \left( \frac{\pi D_s^2}{4} \times R_a \right) - \left( N_w \frac{\pi d_o^2}{4} \right)
\]  

(1.32)

\( R_a \) is obtained from Figure 12.41, for the appropriate baffle cut \( B_c \)

\[
A_{tb} = c_t \pi d_o \left( N_t - N_w \right)
\]  

(1.33)

where \( c_t \) is the diametrical tube-to-baffle clearance; the difference between the hole and tube diameter, typically 0.8 mm.

\[
A_{sb} = \frac{c_s D_s}{2} \left( 2\pi - \theta_b \right)
\]  

(1.34)

where \( c_s \) is the baffle-to-shell clearance, see Figure – 1.2.

\( \theta_b \) can be obtained from Figure 12.41, for the appropriate baffle cut, \( B_c \)

\[
A_b = l_B \left( D_s - D_b \right)
\]  

(1.35)
1.6 Wills-Johnston Method

1.6.1 Streams and flow areas

The bypass flow area

\[ S_{bp} = B (D_s - D_{ot} + N_p \delta_p) \]  \hspace{1cm} (1.36)

where,

- \( D_s \) = Shell ID
- \( B \) = central baffle spacing
- \( D_{ot} \) = outer tube limit diameter
- \( N_p \) = number of tube pass partitions aligned with the cross-flow direction
- \( \delta_p \) = pass partition clearance

Tube-to-baffle leakage flow

\[ S_t = n_t \pi D_o \delta_{tb} \]  \hspace{1cm} (1.37)

Shell-to-baffle leakage flow

\[ S_s = \pi D_s \delta_{sb} \]  \hspace{1cm} (1.38)

where

- \( n_t \) = number of tubes in bundle
- \( \delta_{tb} \) = tube-to-baffle clearance
- \( \delta_{sb} \) = shell-to-baffle clearance
Figure 1.2: Typical baffle clearances.

\[ S_{BW} = \frac{B D_{ot}^2 (\pi - \theta_{ot} + \sin \theta_{ot})}{4 D_s (1 - 2B_c)} - B N_p \delta_p \] (1.39)

where,

\[ \theta_{ot} = 2 \cos^{-1} \left[ \frac{D_s (1 - 2B_c)}{D_{ot}} \right] \] (1.40)

Here, \( B_c \) is the fractional baffle cut and \( \theta_{ot} \) is expressed in radians.

1.6.2 Flow resistances

The cross-flow resistance is given by the following equation

\[ \xi_B = \frac{4aD_o D_s D_v (1 - 2B_c) (P_T - D_o)^{-3} \left( \frac{\dot{m}_B D_o}{\mu S_{BW}} \right)^{-b}}{2\rho g c S_{BW}^2} \] (1.41)

\[ \frac{\dot{m}_B}{\dot{m}_o} = \sqrt{\frac{\xi_x \xi_o}{\xi_y \xi_B}} \] (1.42)

where

\( a = 0.061, b = 0.088 \) for square and rotated-square pitch
\( a = 0.450, b = 0.267 \) for triangular pitch

\[ D_v = \frac{\Omega_1 P_T^2 - D_o^2}{D_o} \] (1.43)

\( \Omega_1 = 1.273 \) for square and rotated-square pitch
\( = 1.103 \) for triangular pitch

The bypass flow resistance is computed as follows:

\[ \xi_{CF} = \frac{0.3164 D_s \left( \frac{1 - 2B_c}{\Omega_2 P_T} \right) \left( \frac{\dot{m}_o D_o}{\mu S_{bp}} \right)^{-0.025} + 2N_{ss}}{2\rho g c S_{bp}^2} \] (1.44)
\[
\frac{\dot{m}_{CF}}{\dot{m}_o} = \sqrt{\frac{\xi_x \xi_o}{\xi_y \xi_{CF}}} \tag{1.45}
\]

where

\[\begin{align*}
N_{ss} & = \text{number of pairs of sealing strips} \\
\Omega_2 & = 1.0 \text{ for square pitch} \\
 & = 1.414 \text{ for rotated-square pitch} \\
 & = 1.732 \text{ for triangular pitch} \\
D_e & = \text{equivalent diameter for the bypass flow}
\end{align*}\]

\[
D_e = \frac{2S_{bp}}{D_s - D_{otl} + 2B + N_p (B + \delta_p)} \tag{1.46}
\]

\[\xi_y = \xi_w + \xi_x \tag{1.47}\]

\[\xi_x = \frac{1}{\left(\frac{1}{\sqrt{\xi_B}} + \frac{1}{\sqrt{\xi_{CF}}}\right)^2} \tag{1.48}\]

**The tube-to-baffle leakage flow resistance** The flow resistance for the tube-to-baffle leakage stream is given by the following equation,

\[
\xi_A = \frac{0.036B_t/\delta_b + 2.3(B_t/\delta_b)^{-0.177}}{2\rho g_c S_t^2} \tag{1.49}\]

where \(B_t\) is the baffle thickness. Flow fraction,

\[
\frac{\dot{m}_A}{\dot{m}_o} = \sqrt{\frac{\xi_o}{\xi_A}} \tag{1.50}\]

where, \(\dot{m}_o\) is total mass flow.

**The shell-to-baffle leakage flow resistance** The flow resistance for the shell-to-baffle leakage stream is given by an equation,

\[
\xi_E = \frac{0.036B_t/\delta_{sb} + 2.3(B_t/\delta_{sb})^{-0.177}}{2\rho g_c S_s^2} \tag{1.51}\]

\[
\frac{\dot{m}_E}{\dot{m}_o} = \sqrt{\frac{\xi_o}{\xi_E}} \tag{1.52}\]

**The window flow resistance** The window flow resistance is given by

\[
\xi_w = \frac{1.9 \exp \left(0.6856S_w/S_m\right)}{2\rho g_c S_w^2} \tag{1.53}\]

\[
\frac{\dot{m}_w}{\dot{m}_o} = \sqrt{\frac{\xi_o}{\xi_y}} \tag{1.54}\]
**Inlet and outlet baffle spaces** The cross-flow resistance in the end spaces is estimated by the following equation

\[ \xi_e = 0.5 \xi_x \left( \frac{B}{B_e} \right)^2 \left[ 1 + \frac{D_{otl}}{D_s (1 - 2B_c)} \right] \]  

(1.55)

The end baffle spaces. The flow resistance in the end windows is calculated as follows

\[ \xi_{we} = \frac{1.9 \exp \left[ 0.6856 S_w B / (S_m B_e) \right]}{2 \rho g_c S_w^2} \]  

(1.56)

The pressure drop in the inlet or outlet baffle space is then given by:

\[ \Delta P_e = \xi_e \dot{m}_e^2 + 0.5 \xi_{we} \dot{m}_w^2 \]  

(1.57)

\[ \Delta P_j = \xi_j \dot{m}_j^2 \quad j = A, B, CF, E \]  

(1.58)

### 1.6.3 Total shell-side pressure drop

\[ \Delta P_o = \psi [ (n_b - 1) \Delta P_y + \Delta P_{in} + \Delta P_{out} ] + \Delta P_n \]  

(1.59)

\[ \psi = 3.646 Re_B^{-0.1934} \quad Re_B < 1000 \]

\[ = 1.0 \quad Re_B \geq 1000 \]

\[ Re_B = \frac{D_o \dot{m}_B}{\mu S_m} \]

\[ \Delta P_y = \xi_y \dot{m}_w^2 \]

where,

\[ n_b = \text{number of baffles.} \]

\[ \Delta P_{in}, \Delta P_{out} = \text{the pressure drops in the inlet and outlet baffle spaces.} \]

\[ \Delta P_n = \text{the pressure drops in the nozzles.} \]

### 1.7 Fouling factor
### Fluid Coefficient (W.m\(^{-2}.\)°C\(^{-1}\)) Resistance (m\(^2.\)°C.W\(^{-1}\))

<table>
<thead>
<tr>
<th>Fluid</th>
<th>Coefficient (W.m(^{-2}.)°C(^{-1}))</th>
<th>Resistance (m(^2.)°C.W(^{-1}))</th>
</tr>
</thead>
<tbody>
<tr>
<td>River water</td>
<td>3000-12,000</td>
<td>0.0003-0.0001</td>
</tr>
<tr>
<td>Sea water</td>
<td>1000-3000</td>
<td>0.001-0.0003</td>
</tr>
<tr>
<td>Cooling water (towers)</td>
<td>3000-6000</td>
<td>0.0003-0.00017</td>
</tr>
<tr>
<td>Towns water (soft)</td>
<td>3000-5000</td>
<td>0.0003-0.0002</td>
</tr>
<tr>
<td>Towns water (hard)</td>
<td>1000-2000</td>
<td>0.001-0.0005</td>
</tr>
<tr>
<td>Steam condensate</td>
<td>1500-5000</td>
<td>0.00067-0.0002</td>
</tr>
<tr>
<td>Steam (oil free)</td>
<td>4000-10,000</td>
<td>0.0025-0.0001</td>
</tr>
<tr>
<td>Steam (oil traces)</td>
<td>2000-5000</td>
<td>0.0005-0.0002</td>
</tr>
<tr>
<td>Refrigerated brine</td>
<td>3000-5000</td>
<td>0.0003-0.0002</td>
</tr>
<tr>
<td>Air and industrial gases</td>
<td>5000-10,000</td>
<td>0.0002-0.0001-1</td>
</tr>
<tr>
<td>Flue gases</td>
<td>2000-5000</td>
<td>0.0005-0.0002</td>
</tr>
<tr>
<td>Organic vapors</td>
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<tr>
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<tr>
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<tr>
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<tr>
<td>Condensing organics</td>
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<td>0.0002</td>
</tr>
<tr>
<td>Heat transfer fluids</td>
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<td>0.0002</td>
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<tr>
<td>Aqueous salt solutions</td>
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<td>0.0003-0.0002</td>
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Figure 1.3: Typical values of fouling coefficients and resistances

### 1.8 Tube data

Standard tube data:

<table>
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<tr>
<th>Tube Size</th>
<th>Outside diameter</th>
<th>Wall thickness</th>
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<td>inch</td>
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<td>10</td>
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<td>18</td>
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<td>3/16</td>
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<td>11/32</td>
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</table>
Chapter 2
Plate Heat Exchanger

2.1 Plate

![Diagram of a plate heat exchanger](image)

Figure 2.1: Plate

2.2 Heat transfer coefficient

\[
\frac{h_p d_e}{k} = 0.26 Re^{0.65} Pr^{0.4} \left( \frac{\mu}{\mu_w} \right)^{0.14}
\]  

(2.1)

\[
Re = \frac{\rho u_p d_e}{\mu}
\]

\[
u_p = \frac{(\dot{m}/n_c)}{\rho A_f}
\]
$A_f = \text{flow area through plates} = b \times L_w,$
$L_w = \text{effective plate width},$
$de = \text{equivalent diameter} = 2b,$
$b = \text{plate gap} = p - t,$
$p = \text{pitch},$
$t = \text{plate thickness},$
$n_c = \text{number of channels}$

### Table 2.1: Heat transfer coefficient

<table>
<thead>
<tr>
<th>Fluid</th>
<th>Coefficient (W/m²·°C)</th>
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<td>River water</td>
<td>3000 – 12,000</td>
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<tr>
<td>Sea water</td>
<td>1000 – 3000</td>
</tr>
<tr>
<td>Cooling water (towers)</td>
<td>3000 – 6000</td>
</tr>
<tr>
<td>Towns water (soft)</td>
<td>3000 – 5000</td>
</tr>
<tr>
<td>Towns water (hard)</td>
<td>1000 – 2000</td>
</tr>
<tr>
<td>Steam condensate</td>
<td>1500 – 5000</td>
</tr>
<tr>
<td>Steam (oil free)</td>
<td>4000 – 10,000</td>
</tr>
<tr>
<td>Steam (oil traces)</td>
<td>2000 – 5000</td>
</tr>
<tr>
<td>Refrigerated brine</td>
<td>3000 – 5000</td>
</tr>
<tr>
<td>Air and industrial gases</td>
<td>5000 – 10,000</td>
</tr>
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<td>Flue gases</td>
<td>2000 – 5000</td>
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<td>Organic vapours</td>
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<td>Organic liquids</td>
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<td>Light hydrocarbons</td>
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<td>Heavy hydrocarbons</td>
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<td>Aqueous salt solutions</td>
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### Table 2.2: Fouling factor in PHE

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<td>Town water (hard)</td>
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<tr>
<td>Cooling water (treated)</td>
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</tr>
<tr>
<td>Sea water</td>
<td>0.00017</td>
</tr>
<tr>
<td>Lubricating oil</td>
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<td>Light organics</td>
<td>0.00010</td>
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<tr>
<td>Process fluids</td>
<td>0.0002 – 0.00005</td>
</tr>
</tbody>
</table>
2.2.1 Number of Transfer Units (NTU)

\[ NTU = \frac{t_o - t_i}{\Delta T_{lm}} \]  

(2.2)

Corrected mean temperature difference,

\[ \Delta T_m = F_t \Delta T_{lm} \]  

(2.3)

2.2.2 Correction factor, \( F_t \)

Refer figure – 2.2

![Figure 2.2: Log mean temperature correction factor for plate heat exchangers](image)

2.3 Pressure drop

2.3.1 Pressure drop in flow through plates

\[ \Delta P_p = 8j_f \left( \frac{L_p}{d_e} \right) \frac{\rho u_p^2}{2} \]  

(2.4)

\( L_p = \) effective plate length, \( = L_v - d_{pt} \)

\( j_f = 0.6Re^{-0.3} \) for turbulent flow.
2.3.2 Pressure drop in flow through port

\[ \Delta P_{pt} = 1.3 \left( \frac{\rho u_{pt}^2}{2} \right) N_P \]  

\( N_P \) = number of passes,
\( u_{pt} \) = velocity in port = \( \frac{\dot{m}}{\rho A_p} \), where \( A_p = \frac{\pi d_{pt}^2}{4} \)

2.4 Plate sizes
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<th>Plate No.</th>
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<th>A (mm)</th>
<th>B (mm)</th>
<th>C (mm)</th>
<th>D (mm)</th>
<th>E (mm)</th>
<th>L2 (mm)</th>
<th>PP (mm)</th>
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<td></td>
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<td>250</td>
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<td>0.3650</td>
<td>10.0200</td>
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<td>12.75</td>
<td>0.4060</td>
<td>11.9380</td>
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<tr>
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<td>13.1240</td>
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<tr>
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<td>15.0000</td>
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<td>16.8760</td>
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<td></td>
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<tr>
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<td>20</td>
<td>0.5940</td>
<td>18.8120</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Chapter 3
Condenser Design

3.1 HTC in Vertical condenser

Heat transfer coefficient for condensation on vertical tubes is given by Nusselt theory,

\[ h = 1.47 \left[ \frac{k_L^3 \rho_L (\rho_L - \rho_v) g}{\mu_L^2 Re} \right]^{1/3} \] for \( Re \leq 30 \) \hspace{1cm} (3.1)

\[ h = \frac{Re \left[ k_L^3 \rho_L (\rho_L - \rho_v) g/\mu_L^2 \right]^{1/3}}{1.08 Re^{1.22} - 5.2} \] for \( 30 \leq Re \leq 1600 \) \hspace{1cm} (3.2)

\[ h = \frac{Re \left[ k_L^3 \rho_L (\rho_L - \rho_v) g/\mu_L^2 \right]^{1/3}}{8750 + 58 Pr_L^{-0.5} (Re^{0.75} - 253)} \] for \( Re \geq 1600 \) and \( Pr \leq 10 \) \hspace{1cm} (3.3)

\[ Re = \frac{4\Gamma}{\mu_L} \]

where

\[ \Gamma = \frac{\dot{m}}{n_t \pi D} \]

3.2 HTC in Horizontal Condenser

Heat transfer coefficient for condensation on horizontal single tube or a single row of tubes is given by Nusselt theory,

\[ h = 1.52 \left[ \frac{k_L^3 \rho_L (\rho_L - \rho_v) g}{\mu_L^2 Re} \right]^{1/3} \] for \( Re \leq 3200 \) \hspace{1cm} (3.4)

\[ Re = \frac{4\Gamma}{\mu_L} \]

where

\[ \Gamma = \frac{\dot{m}}{n_t L} \]

\( k_L \) = thermal conductivity of condensate at average film temperature
\( \rho_L \) = density of condensate at average film temperature
\( \rho_v \) = density of vapour
\[ \mu_L = \text{viscosity of condensate at average film temperature} \]
\[ \dot{m} = \text{rate of condensation at average film temperature} \]
\[ n_t = \text{number of tubes in tube bank} \]
\[ L = \text{tube length} \]

for \( N_r \) tube rows stacked vertically,
\[ h_{N_r} = \frac{h}{N_r^{1/6}} \tag{3.5} \]

for circular tube bundles used in shell-and-tube condensers,
\[ h = 1.52 \left[ \frac{k_L^3 \mu_L (\rho_L - \rho_v) g}{4 \mu_L \Gamma^*} \right]^{1/3} \tag{3.6} \]

where,
\[ \Gamma^* = \frac{\dot{m}}{n_t^{2/3} L} \]

Average film temperature,
\[ T_f = 0.75 T_w + 0.25 T_{sat} \tag{3.7} \]

### 3.3 Condensation with subcooling

Sadisivan and Lienhard equation,
\[ \frac{h}{h_{Nu}} = \left[ 1 + \left( 0.683 - 0.228 \frac{P_{RL}}{P_{RL}} \right) \varepsilon \right]^{1/4} \quad \text{for } P_{RL} \geq 0.6 \tag{3.8} \]

where,
\[ P_{RL} = \frac{C_p L \mu_L}{k_L} \]
\[ \varepsilon = \frac{C_p L (T_{sat} - T_w)}{\lambda} \]

where
\[ h_{Nu} = \text{is the heat-transfer coefficient given by the basic Nusselt theory.} \]
\[ T_{sat} = \text{condensation temperature} \]
\[ T_w = \text{tube wall temperature} \]
\[ \lambda = \text{latent heat of condensation.} \]

### 3.4 Condensation with desuperheating

\[ \frac{h}{h_{Nu}} = \left[ 1 + \frac{C_p v (T_v - T_{sat})}{\lambda} \right]^{1/4} \tag{3.9} \]

where
\[ h_{Nu} = \text{is the heat-transfer coefficient given by the basic Nusselt theory.} \]
\[ C_p v = \text{specific heat of vapour} \]
\[ T_{sat} = \text{condensation temperature} \]
\[ T_v = \text{vapour temperature} \]
\[ \lambda = \text{latent heat of condensation.} \]
3.5 Condensation in vertical tubes with vapour down-flow

The correlation of Boyko and Kruzhilin,

\[ h = h_{Lo} \left[ 1 + x \left( \rho_L - \rho_v \right) / \rho_v \right]^{0.5} \]  

(3.10)

where

\[ x \] = vapour weight fraction

\[ h_{Lo} \] = heat-transfer coefficient for total flow as liquid

3.6 Condensation outside horizontal tubes

McNaught developed the following simple correlation for shear-controlled condensation in tube bundles:

\[ \frac{h}{h_L} = 1.26 X_{tt}^{-0.78} \]  

(3.11)

where,

\[ X_{tt} \] = Lockhart-Martinelli parameter, (refer section – 4.3)

\[ h_L \] = heat-transfer coefficient for the liquid phase flowing alone through the bundle.  
  (refer chapter – 1)
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Chapter 4
Reboiler Design

4.1 Nucleate Boiling

4.1.1 The Forster-Zuber correlation

\[
h_{nb} = 0.00122 \frac{k_L^{0.79} C_p L^{0.45} \rho_L^{0.49} g \Delta T_e^{0.25} \Delta P_{sat}^{0.75}}{\mu_L^{0.29} \lambda^{0.24} \rho_v^{0.24}}
\]  (4.1)

where,

- \(h_{nb}\) = nucleate boiling heat-transfer coefficient, Btu/h \cdot ft\(^2\)\(\cdot\)\(^\circ\)F (W/m\(^2\)\(\cdot\)K)
- \(k_L\) = liquid thermal conductivity, Btu/h \cdot ft\(\cdot\)\(^\circ\)F (W/m\(\cdot\)K)
- \(C_p L\) = liquid heat capacity, Btu/lbm \(\cdot\)\(^\circ\)F (J/kg \(\cdot\)K)
- \(\rho_L\) = liquid density, lbm/ft\(^3\) (kg/m\(^3\))
- \(\mu_L\) = liquid viscosity, lbm/ft\(\cdot\)h (kg/m\(\cdot\)s)
- \(\sigma\) = surface tension, lbf/ft (N/m)
- \(\lambda\) = latent heat of vaporization, Btu/lbm (J/kg)
- \(g\) = unit conversion factor = \(4.17 \times 10^8\) lbm\(\cdot\)ft/\(\cdot\)lbf\(\cdot\)h\(^2\) (1.0 kg\(\cdot\)m\(\cdot\)N\(\cdot\)s\(^2\))
- \(\Delta T_e\) = \(T_w - T_{sat}\), \(^\circ\)F (K)
- \(T_w\) = tube-wall temperature, \(^\circ\)F (K)
- \(T_{sat}\) = saturation temperature at system pressure, \(^\circ\)F (K)
- \(\Delta P_{sat}\) = \(P_{sat}(T_w) - P_{sat}(T_{sat})\), lbf/ft\(^2\) (Pa)
- \(P_{sat}(T)\) = vapor pressure of fluid at temperature T, lbf/ft\(^2\) (Pa)

Any consistent set of units can be used with Equation - 4.1, including the English and SI units shown above.

4.1.2 The Mostinski correlation

In English unit,

\[
h_{nb} = 0.00622 P_c^{0.69} \hat{q}^{0.7} Fp
\]  (4.2)

where,

- \(h_{nb}\) = nucleate boiling heat-transfer coefficient, Btu/h \cdot ft\(^2\)\(\cdot\)\(^\circ\)F
- \(P_c\) = fluid critical pressure, psia
- \(\hat{q}\) = heat flux, Btu/h \cdot ft\(^2\) = \(h_{nb} \Delta T_e\)
- \(Fp\) = pressure correction factor, dimensionless
In SI units,

\[ h_{nb} = 0.00417 P_c^{0.69} \hat{q}^{0.7} F_p \]  

(4.3)

where,

- \( h_{nb} \) = nucleate boiling heat-transfer coefficient, W/m\(^2\)-K
- \( P_c \) = fluid critical pressure, kPa
- \( \hat{q} \) = heat flux, W/m\(^2\) = \( h_{nb} \Delta T_c \)

The pressure correction factor given by:

\[ F_p = 2.1 P_r^{0.27} + \left[ 9 + (1 - P_r^2)^{-1} \right] P_r^2 \]  

(4.4)

where, \( P_r = P/P_c \) = reduced pressure.

### 4.1.3 The Cooper correlation

In English unit same as that of equation – 4.2,

\[ h_{nb} = 21 \hat{q}^{0.67} P_r^{0.12} (- \log_{10} P_r)^{-0.55} M^{-0.5} \]  

(4.5)

In SI unit same as that of equation – 4.3,

\[ h_{nb} = 55 \hat{q}^{0.67} P_r^{0.12} (- \log_{10} P_r)^{-0.55} M^{-0.5} \]  

(4.6)

where \( M \) is the molecular weight of the fluid.

### 4.1.4 The Stephan-Abdelsalam correlation

\[ Z_1 = \frac{\hat{q} d_B}{k_L T_{sat}} \]  

(4.7)

\[ Z_2 = \frac{\alpha^2 P_r}{g_c \sigma d_B} \]  

(4.8)

\[ Z_3 = g_c \lambda d_B^2 \]  

(4.9)

\[ Z_4 = \frac{\rho_v}{\rho_L} \]  

(4.10)

\[ Z_5 = \frac{\rho_L - \rho_v}{\rho_L} \]  

(4.11)

\[ d_B = 0.01466 c \left[ \frac{2 g_c \sigma}{g (\rho_L - \rho_v)} \right]^{0.5} \]  

(4.12)

where,

- \( d_B \) = theoretical diameter of bubbles leaving surface, ft(m)
- \( \theta_c \) = contact angle in degrees
- \( g \) = gravitational acceleration, ft/h\(^2\) (m/s\(^2\))
- \( g_c \) = 4.17 \times 10^8 lbm-ft/lbf-h\(^2\) (1.0 kg-m/N-s\(^2\))
\[ \alpha_L = \text{liquid thermal diffusivity, ft}^2/\text{s (m}^2/\text{s)} \]
\[ \dot{q} \propto \text{Btu/h-ft}^2/\text{W/m}^2 \]
\[ k_L \propto \text{Btu/h-ft} \cdot ^\circ \text{F (W/m-K)} \]
\[ T_{\text{sat}} \propto ^\circ \text{R (K)} \]
\[ \sigma \propto \text{lbf/ft (N/m)} \]
\[ \rho_L, \rho_v \propto \text{lbm/ft}^3/\text{kg/m}^3 \]
\[ \lambda \propto \text{ft-lbf/lbm (J/kg)} \]

1 Btu = 778 ft-lbf

The heat-transfer coefficient is given by the following equation:

\[ \frac{h_{nb}d_B}{k_L} = 0.23Z^{0.674}_1Z^{0.35}_2Z^{0.371}_3Z^{0.297}_4Z^{-1.73}_5 \quad \text{(4.13)} \]

<table>
<thead>
<tr>
<th>Fluid group</th>
<th>Contact angle ((\theta_c)) in (^\circ)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water</td>
<td>45</td>
</tr>
<tr>
<td>Hydrocarbons (including alcohols)</td>
<td>35</td>
</tr>
<tr>
<td>Refrigerants (including CO(_2), propane, n-butane)</td>
<td>35</td>
</tr>
<tr>
<td>Cryogenic fluids (including methane, ethane)</td>
<td>1</td>
</tr>
</tbody>
</table>

### 4.1.5 Boiling mixtures

The coefficient, \(h_{\text{ideal}}\), is an average of the pure component values that is calculated as follows:

\[ h_{\text{ideal}} = \left[ \sum_{i=1}^{n} \frac{x_i}{h_{nb,i}} \right] \quad \text{(4.14)} \]

where \(h_{nb,i}\) is the heat-transfer coefficient for pure component \(i\). So heat transfer coefficient for mixture is,

\[ h_{nb} = h_{\text{ideal}} \left\{ 1 + \left( \frac{\text{BR} \cdot h_{\text{ideal}}}{\dot{q}} \right) \left[ 1 - \exp \left( \frac{-\dot{q}}{\rho_L \lambda \beta} \right) \right] \right\}^{-1} \quad \text{(4.15)} \]

where

\[ \text{BR} = T_D - T_B = \text{boiling range} \]
\[ T_D = \text{dew-point temperature} \]
\[ T_B = \text{bubble-point temperature} \]
\[ \beta = 0.0003 \text{ m/s (SI units)} = 3.54 \text{ ft/h (English units)} \]

### 4.1.6 Convective effects in tube bundles

The average boiling heat-transfer coefficient, \(h_b\), is expressed as follows:

\[ h_b = h_{nb}F_b + h_{nc} \quad \text{(4.16)} \]

where \(h_{nc}\) is a heat-transfer coefficient for liquid-phase natural convection and \(F_b\) is a factor that accounts for the effect of the thermosyphon-type circulation in the tube bundle. The
bundle convection factor is correlated in terms of bundle geometry by the following empirical equation,

\[
F_b = 1.0 + 0.1 \left[ \frac{0.785D_b}{C_1 (P_T/D_o)^2 D_o} - 1.0 \right]^{0.75} 
\]  

(4.17)

where,

\( D_b \) = bundle diameter (outer tube-limit diameter)  
\( D_o \) = tube OD  
\( P_T \) = tube pitch  
\( C_1 \) = 1.0 for square and rotated square layouts  
\( = 0.866 \) for triangular layouts

For larger temperature differences, therefore, Palen suggests using a rough approximation for \( h_{nc} \) of 250 W/m\(^2\). K (44 Btu/h·ft\(^2\)·°F) for hydrocarbons and 1000 W/m\(^2\). K (176 Btu/h·ft\(^2\)·°F) for water and aqueous solutions.

### 4.2 Critical heat flux

The equation for critical heat flux is generally used in the following form:

\[
\hat{q}_c = 0.149\lambda\sqrt{\rho_c} [\sigma g g_c (\rho_L - \rho_v)]^{0.25}
\]  

(4.18)

#### 4.2.1 Mostinski correlation

Boiling on single tube. For English unit,

\[
\hat{q}_c = 803P_c P_{cr}^{0.35} (1 - P_r)^{0.9}
\]  

(4.19)

where \( \hat{q}_c \) Btu/h·ft\(^2\) and \( P_c \) in psia.

For SI unit,

\[
\hat{q}_c = 367P_c P_{cr}^{0.35} (1 - P_r)^{0.9}
\]  

(4.20)

where \( \hat{q}_c \) W/m\(^2\) and \( P_c \) in kPa.

For tube bundles, Palen presented the following correlation:

\[
\hat{q}_{c,bundle} = \hat{q}_{c,tube} \phi_b
\]  

(4.21)

where

\( \hat{q}_{c,bundle} \) = critical heat flux for tube bundle  
\( \hat{q}_{c,tube} \) = critical heat flux for a single tube  
\( \phi_b \) = bundle correction factor  
\( = 3.1\psi_b \) for \( \psi_b < 1.0/3.1 \approx 0.323 \)  
\( = 1.0 \) otherwise  
\( \psi_b \) = dimensionless bundle geometry parameter = \( \frac{\pi D_b L}{A} \)  
\( D_b \) = bundle diameter  
\( A \) = bundle surface area = \( n_t \pi D_o L \) for plain tubes  
\( D_o \) = tube OD  
\( L \) = tube length  
\( n_t \) = number of tubes in bundle
4.3 Two Phase Flow

4.3.1 Pressure drop correlations

The Lockhart-Martinelli correlation

The two-phase pressure gradient is expressed as,

\[
\frac{\Delta P_f}{L}_{tp} = \phi_2^2 \left( \frac{\Delta P_f}{L} \right)_L
\]  

(4.22)

where

\[
\phi_2^2 = \text{two-phase multiplier}
\]

\[
\left( \frac{\Delta P_f}{L} \right)_L = \text{negative pressure gradient for liquid alone}
\]

\[
\left( \frac{\Delta P_f}{L} \right)_{tp} = \text{negative two-phase pressure gradient}
\]

The two-phase multiplier is a function of the parameter, \( X \), which is defined as follows:

\[
X = \left[ \frac{\left( \frac{\Delta P_f}{L} \right)_L}{\left( \frac{\Delta P_f}{L} \right)_v} \right]^{0.5}
\]  

(4.23)

where \( \left( \frac{\Delta P_f}{L} \right)_v \) is the pressure gradient that would occur if the vapor phase flowed alone in the conduit. The relationship between \( \phi_2^2 \) and \( X \) was given in graphical form by Lockhart and Martinelli, and subsequently expressed analytically by Chisholm as follows:

\[
\phi_2^2 = 1 + \frac{C}{X} + \frac{1}{X^2}
\]  

(4.24)

The constant, \( C \), depends on whether the flow in each phase is laminar or turbulent, as shown in Table – 4.1.

<table>
<thead>
<tr>
<th>Liquid Notation</th>
<th>Vapour Notation</th>
<th>( Re_L )</th>
<th>( Re_V )</th>
<th>( C )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Turbulent ( tt )</td>
<td>Turbulent ( tt )</td>
<td>&gt; 2000</td>
<td>&gt; 2000</td>
<td>20</td>
</tr>
<tr>
<td>Viscous (laminar) ( vt )</td>
<td>Turbulent ( vt )</td>
<td>&lt; 1000</td>
<td>&gt; 2000</td>
<td>12</td>
</tr>
<tr>
<td>Turbulent ( tv )</td>
<td>Viscous (laminar) ( tv )</td>
<td>&gt; 2000</td>
<td>&lt; 1000</td>
<td>10</td>
</tr>
<tr>
<td>Viscous (laminar)</td>
<td>Viscous (laminar)</td>
<td>&lt; 1000</td>
<td>&lt; 1000</td>
<td>5</td>
</tr>
</tbody>
</table>

Table 4.1: Values of the Constant in Equation – 4.24

The Chisholm correlation

The two-phase pressure gradient is expressed as,

\[
\frac{\Delta P_f}{L}_{tp} = \phi_{LO}^2 \left( \frac{\Delta P_f}{L} \right)_{LO}
\]  

(4.26)

where,

\[
\phi_{LO}^2 = \text{two-phase multiplier for \( \Delta P_f \) alone}
\]

\[
\left( \frac{\Delta P_f}{L} \right)_{LO} = \text{negative \( \Delta P_f \) pressure gradient for liquid alone}
\]
\( \phi_{LO}^2 = \) two-phase multiplier

\( (\Delta P_f/L)_{LO} = \) negative pressure gradient for liquid alone

\( (\Delta P_f/L)_{tp} = \) negative two-phase pressure gradient

The correlation for the two-phase multiplier is the following:

\[
\phi_{LO}^2 = 1 + (Y^2 - 1) \left\{ B \left[ x (1 - x) \right]^{(2-n)/2} + x^{2-n} \right\} \tag{4.27}
\]

\[
Y = \left( \frac{\rho_L}{\rho_v} \right)^{0.5} \left( \frac{\mu_v}{\mu_L} \right)^{n/2} \tag{4.28}
\]

where \( n = 0.2314 \)

For English unit with \( G \) in units of lbm/h · ft²:

\[
B = \frac{1500}{\sqrt{G}} \quad (0 < Y \leq 9.5)
\]

\[
= \frac{14250}{\left( Y \sqrt{G} \right)} \quad (9.5 < Y \leq 28) \tag{4.29}
\]

\[
= \frac{399000}{\left( Y^2 \sqrt{G} \right)} \quad (Y > 28)
\]

For calculations in SI units, the following conversion can be used:

\[
G (\text{lbm/h · ft}^2) \equiv 737.35G (\text{kg/s · m}^2)
\]

The Friedel correlation

The two-phase pressure gradient is expressed in the same manner as the Chisholm method, Equation – 4.26 with the following correlation for the two-phase multiplier:

\[
\phi_{LO}^2 = E + \frac{3.24FH}{Fr^{0.045}We^{0.035}} \tag{4.30}
\]

where,

\[
E = (1 - x)^2 + x^2 (\mu_v/\mu_L)^{0.2314} (\rho_L/\rho_v)
\]

\[
F = x^{0.78} (1 - x)^{0.24}
\]

\[
H = (\rho_L/\rho_v)^{0.91} (\mu_v/\mu_L)^{0.19} (1 - \mu_v/\mu_L)^{0.7}
\]

\[
Fr = \frac{G^2}{gD_i\rho_{tp}^2} = \text{Froude number}
\]

\[
We = \frac{G^2D_i} {g_v\rho_{tp}\sigma} = \text{Weber number}
\]

\[
D_i = \text{internal diameter of conduit}
\]

\[
\rho_{tp} = \text{two-phase density}
\]

For the purpose of this correlation, the two-phase density is calculated as follows:

\[
\rho_{tp} = \left[ x/\rho_v + (1 - x)/\rho_L \right]^{-1} \tag{4.31}
\]

where, \( x \) is vapour mass fraction.
Slip ratio,

$$\text{SR} = \sqrt{\frac{\rho_L}{\rho_{tp}}} \quad (4.32)$$

void fraction is computed

$$\varepsilon_v = \frac{x}{x + \text{SR} (1 - x) \frac{\rho_v}{\rho_L}} \quad (4.33)$$

Finally, the average two-phase density is computed as,

$$\bar{\rho}_{tp} = \varepsilon_v \rho_v + (1 - \varepsilon_v) \rho_L \quad (4.34)$$

The Müller-Steinhagen and Heck (MSH) correlation

The correlation is reformulated in the Chisholm format of Equation – 4.26 with the two-phase multiplier given by the following equation:

$$\phi_{LO}^2 = Y^2 x^3 + [1 + 2x (Y^2 - 1)] (1 - x)^{1/3} \quad (4.35)$$

where \(x\) is the vapor mass fraction and \(Y\) is the Chisholm parameter (equation – 4.28).

### 4.4 Convective Boiling in Tubes

#### 4.4.1 Heat transfer coefficient

The Chen correlation

$$h_b = S_{CH} h_{nb} + F_x h_L \quad (4.36)$$

where

\(h_b\) = convective boiling heat-transfer coefficient  
\(h_{nb}\) = nucleate boiling heat-transfer coefficient  
\(S_{CH} = (1 + 2.53 \times 10^{-6} R e^{1.17})^{-1}\)  
\(F_x = 2.35 (X_{tt}^{-1} + 0.213)^{0.736}\) for \(X_{tt} < 10\)  
\(= 1.0\) for \(X_{tt} \geq 10\)  
\(R e = R e_L (F_x)^{1.25}\)

The heat flux is calculated as follows:

$$\dot{q} = S_{CH} h_{nb} (T_w - T_{sat}) + h_L (T_w - T_b) \quad (4.37)$$

whereas convective heat transfer coefficient, \(h_L\), can be calculated using Dittus-Boelter equation,

$$h_L = 0.023 \left(\frac{k_L}{D_i}\right) R e_L^{0.8} P r_L^{0.4} \quad (4.38)$$
4.4.2 Critical heat flux

The following simple correlation for vertical thermosyphon reboilers was given by Palen

\[
\dot{q}_c = 16070 \left( \frac{D^2}{L} \right)^{0.35} P_r^{0.61} P_r^{0.25} (1 - P_r) \text{ Btu/h \cdot ft}^2 (W/m^2) \quad \text{English Unit} \tag{4.39}
\]

\[
\dot{q}_c = 23660 \left( \frac{D^2}{L} \right)^{0.35} P_r^{0.61} P_r^{0.25} (1 - P_r) \text{ SI units} \tag{4.40}
\]

where

\[\dot{q}_c = \text{critical heat flux}, \text{ Btu/h \cdot ft}^2 (W/m^2)\]

\[D = \text{tube ID, ft (m)}\]

\[L = \text{tube length, ft(m)}\]

\[P_r = \text{reduced pressure in tube}\]

\[P_c = \text{critical pressure of fluid, psia (kPa)}\]

For flow in horizontal tubes, the dimensionless correlation of Merilo is recommended by Hewitt et al.

\[
\frac{\dot{q}_c}{G\lambda} = 575 \gamma_H^{-0.34} \left( \frac{L}{D} \right)^{-0.511} \left( \frac{\rho_L - \rho_v}{\rho_v} \right)^{1.27} (1 + \Delta H_{in}/\lambda)^{1.64} \tag{4.41}
\]

where

\[\gamma_H = \left( \frac{GD}{\mu_L} \right) \left( \frac{\mu_L^2}{g_{c} \sigma D \rho_L} \right)^{-1.58} \left[ \left( \frac{\rho_L - \rho_v}{g_{c} \sigma} \right)^2 \right]^{-1.05} \left( \frac{\mu_L}{\mu_v} \right)^{6.41} \tag{4.42}\]

The correlation cover the ranges \(5.3 \leq D \leq 19.1 \text{ mm}, 700 \leq G \leq 8100 \text{ kg/s} \cdot \text{m}^2, 13 \leq \rho_L/\rho_v \leq 21\).

4.5 Film Boiling

4.5.1 Heat transfer coefficient

A combined heat-transfer coefficient, \(h_t\), for both convection and radiation can be calculated from the following equation:

\[h_t^{4/3} = h_{fb}^{4/3} + h_r^{1/3} \tag{4.43}\]

For saturated film boiling on the outside of a single horizontal tube,

\[
\frac{h_{fb} D_o}{k_v} = 0.62 \left[ \frac{g \rho_v (\rho_L - \rho_v) D_o^3 (\lambda + 0.76 C_{p,v} \Delta T_e)}{k_v \mu_v \Delta T_e} \right]^{0.25} \tag{4.44}
\]

Here, \(D_o\) is the tube OD. \(h_r\) is the radiative heat-transfer coefficient calculated from the following equation:

\[h_r = \frac{\varepsilon \sigma_{SB} (T_w^4 - T_{sat}^4)}{T_w - T_{sat}} \tag{4.45}\]

where,

\[\varepsilon = \text{emissivity of tube wall}\]

\[\sigma_{SB} = \text{Stefan-Boltzmann constant}\]

\[= 5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4 = 1.714 \times 10^{-9} \text{ Btu/h \cdot ft}^2 \cdot \circ \text{R}^4\]

If \(h_r < h_{fb}\), Equation – 4.43 can be approximated by the following explicit formula for \(h_t\):

\[h_t = h_{fb} + 0.75 h_r \tag{4.46}\]
4.6 Design equations

4.6.1 Number of nozzles
For a tube bundle of length $L$ and diameter $D_b$, the number, $N_n$, of nozzle pairs (feed and return) is determined from the following empirical equation,

$$N_n = \frac{L}{5D_b} \quad (4.47)$$

4.6.2 Shell diameter
Vapour loading,

$$VL = 2290 \rho_v \left( \frac{\sigma}{\rho_L - \rho_v} \right)^{0.5} \quad (4.48)$$

where

- $VL$ = vapor loading ($lbm/h \cdot ft^3$)
- $\rho_v, \rho_L$ = vapor and liquid densities ($lbm/ft^3$)
- $\sigma$ = surface tension ($dyne/cm$)

The dome segment area, $SA$, is calculated from the vapour loading as follows:

$$SA = \frac{\dot{m}_V \times VL}{L} \quad (4.49)$$

The segment area till semicircle is given by,

$$SA = \frac{D_s^2}{8} (\theta - \sin \theta) \quad (4.50)$$

$$\theta = 2 \cos^{-1} \left( 1 - \frac{2h}{D_s} \right) \quad (4.51)$$

where, $D_s$ Shell ID and $h$ is height of the segment. When segment exceeds semicircle the segment area is area of circle minus area of segment whose height in the circle diameter minus height of the given segment.

4.7 Frictional losses in pipe

4.7.1 Friction factor
Reynold’s number

$$N_{Re} = \frac{dv\rho}{\mu}$$

Friction factor in Laminar flow:

$$f = \frac{16}{N_{Re}}$$
Friction factor in turbulent flow:

A. For smooth pipe/tubes (Turbulent)

i) \( f = 0.046N_{Re}^{-0.2} \) for \( 50000 < N_{Re} < 1 \times 10^6 \)

ii) \( f = 0.0014 + \frac{0.125}{N_{Re}^{0.32}} \) for \( 3000 < N_{Re} < 3 \times 10^6 \)

iii) von-Karman equation

\[
\frac{1}{\sqrt{f/2}} = 2.5 \ln \left( N_{Re} \sqrt{f/8} \right) + 1.75
\]

B. For Commercial pipes (Turbulent)

i) Colebrook equation

\[
\frac{1}{\sqrt{4f}} = 2 \log \left( \frac{D}{2K} \right) + 1.74
\]

ii) Generalised equation

\( f = 0.3673N_{Re}^{-0.2314} \)

\[4.7.2\] Pressure drop in pipe

\[
\Delta P = 4f \left( \frac{L}{d} \right) \frac{\rho v^2}{2}
\]

\[4.7.3\] Maximum gas/vapour velocity in tubes

For plain carbon steel tube in English unit,

\[
v_{max} = \frac{1800}{\sqrt{PM}}
\]

in SI units,

\[
v_{max} = \frac{1440}{\sqrt{PM}}
\]

where,

- \( v_{max} \) = maximum velocity, ft/s (m/s)
- \( P \) = gas pressure, psia(kPa)
- \( M \) = molecular weight of gas

Multiply equation – (4.52) or (4.53) with 1.5 for stainless steel and 0.6 for copper tube.

\[4.7.4\] Maximum velocity of liquids in tubes

i. Maximum recommended velocity of water in plain carbon steel tube is 10 ft/s (3 m/s).

ii. Multiply above value with 1.5 for stainless steel and 0.6 for copper tube.

iii. Multiply above value with the factor \( \sqrt{\rho_{water}/\rho_{liquid}} \) if liquid is other than water.
4.7.5 Maximum velocity of two-phase flow in tubes/pipe

English unit,

\[ v_{\text{max}} = \sqrt{\frac{4000}{\rho_{tp}}} \]  \hspace{1cm} (4.54)

SI unit,

\[ v_{\text{max}} = \sqrt{\frac{5924}{\rho_{tp}}} \]  \hspace{1cm} (4.55)

\( v_{\text{max}} \) = maximum velocity, ft/s (m/s)
\( \rho_{tp} \) = density of two-phase mixture, lbm/ft\(^3\) (kg/m\(^3\))

4.8 Design of Vertical Thermosyphon Reboiler

![Diagram of vertical thermosyphon reboiler system]

Figure 4.1: Configuration of vertical thermosyphon reboiler system.

4.8.1 Pressure balance

\[ P_B - P_A = \rho_L \left( \frac{g}{g_c} \right) (z_A - z_B) - 4f \left( \frac{L_{\text{in}}}{D_{\text{in}}} \right) \frac{G_{\text{in}}^2}{2\rho_L} \]  \hspace{1cm} (4.56)

The subscript in refers to the inlet line to the reboiler.

\[ P_C - P_B = \rho_L \left( \frac{g}{g_c} \right) L_{BC} - 4f \left( \frac{L_{BC}}{D_l} \right) \frac{G_l^2}{2\rho_L} \]  \hspace{1cm} (4.57)
The subscript $t$ in this equation refers to the reboiler tubes.

$$P_D - P_C = -\Delta P_{\text{static}CD} - \Delta P_{f,CD} - \Delta P_{\text{acc}CD}$$  \hspace{1cm} (4.58)

$$\Delta P_{\text{static}CD} = \bar{\rho}_t (g/g_c) L_{CD}$$  \hspace{1cm} (4.59)

$$\Delta P_{f,CD} = 4f \left( \frac{L_{CD}}{D_t} \right) \frac{G_t^2 \phi_{LO}^2}{2 \rho_L}$$  \hspace{1cm} (4.60)

$$\Delta P_{\text{acc}CD} = \frac{G_t^2 \gamma}{\rho_L}$$  \hspace{1cm} (4.61)

Fair recommends calculating $\bar{\rho}_t$ at a vapour weight fraction equal to one-thirds the value at the reboiler exit using equation – (4.34).

$$\gamma = \frac{(1 - x_e)^2}{1 - \varepsilon_v,e} + \frac{\rho_L x_e^2}{\rho_v \varepsilon_v,e} - 1$$

In this equation, $x_e$ and $\varepsilon_v,e$ are the vapour mass fraction and the void fraction at the reboiler exit.

Fair recommends calculating $\bar{\phi}_{LO}$ at a vapour weight fraction equal to two-thirds the value at the reboiler exit.

$$P_A - P_D = (G_t^2 - G_{ex}^2) \left( \frac{\gamma + 1}{\rho_L} \right) - \frac{4f_{ex} L_{ex} G_{ex}^2 \phi_{LO,ex}^2}{2 \rho_L D_{ex}}$$  \hspace{1cm} (4.62)

In this equation, the subscript $ex$ designates conditions in the exit line from the reboiler.

The relationship between the circulation rate and the exit vapour fraction in the reboiler in SI units is,

$$\dot{m}_i^2 = \frac{1.234 D_t^5 \rho_L (g/g_c) (\rho_L L_{AC} - \bar{\rho}_t L_{CD})}{2D_t \left( \frac{\gamma + 1}{D_{ex}} \right)^4 \left( \frac{1}{\dot{m}_t^2} \right) + f_{in} L_{in} \left( \frac{D_t}{D_{in}} \right)^5 + \left( \frac{f_t}{\dot{m}_t^2} \right) (L_{BC} + L_{CD} \bar{\phi}_{LO}^2) + f_{ex} L_{ex} \phi_{LO,ex}^2 \left( \frac{D_t}{D_{ex}} \right)^5}$$  \hspace{1cm} (4.63)

where,

$\dot{m}_i$ = tube-side mass flow rate (kg/s)

$n_t$ = number of tubes in reboiler

### 4.8.2 Sensible heating zone

$$\frac{T_C - T_B}{P_C - P_B} = \frac{(\Delta T/L)}{(\Delta P/L)}$$  \hspace{1cm} (4.64)

$$\frac{T_{sat} - T_A}{P_{sat} - P_A} = \frac{(\Delta T/\Delta P)_{sat}}{(\Delta T)}$$  \hspace{1cm} (4.65)
The pressure gradient in the sensible heating zone is calculated as follows:

\[ -(\Delta P/L) = \rho_L (g/g_c) + \Delta P_{f,BC}/L \]  

(4.67)

The temperature gradient in the sensible heating zone is estimated as follows:

\[ \Delta T/L = \frac{n_t \pi D_o U_D \Delta T_m}{\dot{m}_t C_pL} \]  

(4.68)

Here, \( U_D \) and \( \Delta T_m \) are the overall coefficient and mean driving force, respectively, for the sensible heating zone.

### 4.8.3 Mist flow limit

Tube-side mass flux at onset of mist flow,

\[ G_{t,\text{mist}} = 1.8 \times 10^6 X_{tt} \quad \text{lbm/h} \cdot \text{ft}^2 \]  

(4.69)

\[ G_{t,\text{mist}} = 2.44 \times 10^3 X_{tt} \quad \text{kg/s} \cdot \text{m}^2 \]  

(4.70)

***********************
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Part II

Data Sheet
# Heat Exchanger Specification Sheet

**Company:**

**Location:**

**Service of Unit:**

**Item No.:**

**Prepared by:**

**Date:**

**Rev No.:**

**Job No.:**

## 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></td>
<td></td>
</tr>
<tr>
<td>Fluid quantity, Total</td>
<td>kg/h</td>
<td></td>
</tr>
<tr>
<td>Vapor (In/Out)</td>
<td>kg/h</td>
<td></td>
</tr>
<tr>
<td>Liquid</td>
<td>kg/h</td>
<td></td>
</tr>
<tr>
<td>Noncondensable</td>
<td>kg/h</td>
<td></td>
</tr>
<tr>
<td>Temperature (In/Out)</td>
<td>°C</td>
<td></td>
</tr>
<tr>
<td>Dew / Bubble point</td>
<td>°C</td>
<td></td>
</tr>
<tr>
<td>Density</td>
<td>kg/m³</td>
<td></td>
</tr>
<tr>
<td>Viscosity</td>
<td>cp</td>
<td></td>
</tr>
<tr>
<td>Molecular wt, Vap</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Molecular wt, NC</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Specific heat</td>
<td>kJ/(kg*K)</td>
<td></td>
</tr>
<tr>
<td>Thermal conductivity</td>
<td>W/(m²*K)</td>
<td></td>
</tr>
<tr>
<td>Latent heat</td>
<td>kJ/kg</td>
<td></td>
</tr>
<tr>
<td>Pressure</td>
<td>mmH₂O(g)</td>
<td></td>
</tr>
<tr>
<td>Velocity</td>
<td>m/s</td>
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<tr>
<td>Pressure drop, allow./calc.</td>
<td>mmH₂O</td>
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<tr>
<td>Fouling resist. (min)</td>
<td>m²*K/W</td>
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<td>Heat exchanged</td>
<td>kcal/h</td>
<td>MTD corrected °C</td>
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## Transfer rate, Service

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<th>W/(m²*K)</th>
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## CONSTRUCTION OF ONE SHELL

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<th>kgf/cm²</th>
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<td>Number passes per shell</td>
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<tr>
<td>Corrosion allowance</td>
<td>mm</td>
</tr>
<tr>
<td>Connections</td>
<td>In</td>
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<tr>
<td>Size/rating</td>
<td>Out</td>
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<td>mm</td>
<td>Intermediate</td>
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<table>
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<th>Tks- avg</th>
<th>mm</th>
<th>Length</th>
<th>mm</th>
<th>Pitch</th>
<th>mm</th>
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<td>mm</td>
<td>Shell cover</td>
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<tr>
<td>Channel or bonnet</td>
<td>Channel cover</td>
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<tr>
<td>Tubesheet-stationary</td>
<td>Tubesheet-floating</td>
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<tr>
<td>Floating head cover</td>
<td>Impingement protection</td>
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<tr>
<td>Baffle-crossing</td>
<td>Type</td>
<td>Cut(%d)</td>
<td>Spacing: c/c</td>
<td>mm</td>
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<td>Seal type</td>
<td>Inlet</td>
<td>mm</td>
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<td>U-bend</td>
<td>Type</td>
<td></td>
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</tr>
<tr>
<td>Tube type</td>
<td>Material</td>
<td>Tube pattern</td>
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| Bypass seal | Tube-tubesheet joint | groove/expand |
| Expansion joint | Type | |
| RhoV2-Inlet nozzle | Bundle entrance | Bundle exit | kg/(m²s²) |
| Gaskets - Shell side | Tube Side | |
| Floating head | Type | |
| Code requirements | TEMA class | |
| Weight/Shell | Filled with water | Bundle | kg |
| Remarks | |

| Weight/Shell | |

**Remarks:**

---

**Notes:**

- **Job No.:**
- **Rev No.:**
- **Prepared by:**
- **Date:**
- **Size:** mm
- **Type:** mm²
- **Surf/unit(eff.):** m²
- **Connected in:** parallel
- **Series:**
- **Surf/shell (eff.):** m²
- **Fluid allocation:** Shell Side | Tube Side
- **Fluid name:**
- **Fluid quantity, Total:** kg/h
- **Vapor (In/Out):** kg/h
- **Liquid:** kg/h
- **Noncondensable:** kg/h
- **Temperature (In/Out):** °C
- **Dew / Bubble point:** °C
- **Density:** kg/m³
- **Viscosity:** cp
- **Molecular wt, Vap:**
- **Molecular wt, NC:**
- **Specific heat:** kJ/(kg*K)
- **Thermal conductivity:** W/(m²*K)
- **Latent heat:** kJ/kg
- **Pressure:** mmH₂O(g)
- **Velocity:** m/s
- **Pressure drop, allow./calc.:** mmH₂O
- **Fouling resist. (min):** m²*K/W
- **Heat exchanged:** kcal/h
- **MTD corrected:** °C
- **Transfer rate, Service:** Dirty | Clean
- **W/(m²*K):**
- **Design/Test pressure:** kgf/cm²
- **Design temperature:** °C
- **Number passes per shell:** mm
- **Corrosion allowance:** mm
- **Connections:** In
- **Size/rating:** Out
- **mm:** Intermediate
- **Tube No.:** OD | Tks- avg | mm | Length | mm | Pitch | mm
- **Shell ID:**
- **OD:** mm
- **Shell cover:**
- **Channel or bonnet:** Channel cover
- **Tubesheet-stationary:** Tubesheet-floating
- **Floating head cover:** Impingement protection
- **Baffle-crossing:** Type | Cut(%d) | Spacing: c/c | mm
- **Baffle-long:** Seal type | Inlet | mm
- **Supports-tube:** U-bend | Type
- **Tube type:** Material | Tube pattern
- **Bypass seal:** Tube-tubesheet joint | groove/expand
- **Expansion joint:** Type
- **RhoV2-Inlet nozzle:** Bundle entrance | Bundle exit | kg/(m²s²)
- **Gaskets - Shell side:** Tube Side
- **Floating head:** Type
- **Code requirements:** TEMA class
- **Weight/Shell:** Filled with water | Bundle | kg
- **Remarks:**

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<td>03 Total flow</td>
<td>(kg/s)</td>
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<td>04 Flow per exchanger</td>
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<td>[kPa (ga)]</td>
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<td>/</td>
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<td>10 Fouling margin a</td>
<td>(%)</td>
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<td>11 OPERATING DATA</td>
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<td>OUTLET</td>
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<td>16 Operating pressure</td>
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<td>17 LIQUID PROPERTIES</td>
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<td>18 Density</td>
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<td>19 Specific heat capacity</td>
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<td>25 Specific heat capacity</td>
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<td>27 Thermal conductivity</td>
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<td>28 Relative molecular mass</td>
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<td>30 Dew point/bubble point</td>
<td>(°C)</td>
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<tr>
<td>31 Solids maximum size</td>
<td>(mm)</td>
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<td>32 Solids concentration (% volume)</td>
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<tr>
<td>33 Latent heat</td>
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<td>34 Critical pressure</td>
<td>[kPa (abs)]</td>
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<td>35 Critical temperature</td>
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<td>36</td>
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<tr>
<td>37 Total heat exchanged</td>
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</tr>
<tr>
<td>38 $U^a$</td>
<td>(W/m²·K)</td>
<td>Clean condition:</td>
<td>Service:</td>
<td></td>
</tr>
<tr>
<td>39 LMTD</td>
<td>(°C)</td>
<td>/</td>
<td></td>
<td></td>
</tr>
<tr>
<td>40 Heat transfer area</td>
<td>(m²)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>41 Stream heat transfer coefficient</td>
<td>(W/m²·K)</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Fouling margin $a$ = $[\left(U_{\text{clean}}/U_{\text{service}}\right) - 1] \times 100$ % where $U$ = Overall heat transfer coefficient (thermal transmittance).

| Rev. No. | Revision Date | Prepared by | Reviewed by |