Chapter 5
Compact Heat Exchangers (Part III)

5.8 Plate-Fin Heat Exchangers

Plate-fin exchangers have various geometries of fins to compensate the high thermal resistance by increasing the heat transfer area particularly if one of fluids is air or gas. This type of exchanger has corrugated fins sandwiched between parallel plates or formed tubes. The plate fins are categorized as triangular fin, rectangular fin, wavy fin, offset strip fin (OSF), louvered fin, and perforated fin. One of the most widely used enhanced fin geometries is the offset strip fin type, which is shown in Figure 5.40.

Plate-fin heat exchangers have been introduced since the 1910s in the auto industry, since the 1940s in the aerospace industry. They are now used widely in many industries for aircraft, cryogenics, gas turbine, nuclear, and fuel cell.

Plate-fin heat exchangers are generally designed for moderate operating pressures less than 700 kPa (gauge pressure) and have been built with a surface area density of up to 5900 m²/m³. Common fin thickness ranges between 0.05 and 0.25 mm. Fin heights may range from 2 to 25 mm. Although typical fin densities are 120 to 700 fins/m, applications exist for as many as 2100 fins/m [17].

Figure 5.40 Plate-fin heat exchanger, employing offset strip fin.
5.8.1 Geometric Characteristics

A schematic of a single-pass crossflow plate-fin heat exchanger, employing offset strip fins, is shown in Figure 5.40. The idealized fin geometry is shown in Figure 5.41. Defining the total heat transfer area for each fluid is important in the analysis. The total area consists of the primary area and the fin area. The primary area consists of the plate area except the fin base area, multi-passage side walls, and multi-passage front and back walls. It is practical to assume that the numbers of passages for the hot fluid side and the cold fluid side are $N_p$ and $N_p + 1$ to minimize the heat loss to the ambient. The top and bottom passages in Figure 5.40 are designated to be cold fluid. The number of passages (don’t confuse with the number of passes) can be obtained from an expression for $L_3$ as

$$L_3 = \frac{N_p b_1}{\text{length of fluid 1}} + \frac{(N_p + 1)b_2}{\text{length of fluid 2}} + \frac{2(N_p + 1)\delta_w}{\text{thickness of total plates}}$$  \hspace{1cm} (5.243)

Solving for $N_p$ gives the number of passages for hot fluid

$$N_p = \frac{L_3 - b_2 - 2\delta_w}{b_1 + b_2 + 2\delta_w}$$  \hspace{1cm} (5.244)

By definition, the number of passages counts the number based on one flow passage between two plates, not all the individual channels between the plates. The total number of fins for fluid 1 (hot) is calculated by

$$n_{f1} = \frac{L_1}{p_{f1}} N_p$$  \hspace{1cm} (5.245)

For fluid 2 (cold),

$$n_{f2} = \frac{L_2}{p_{f2}} (N_p + 1)$$  \hspace{1cm} (5.245a)
where \( p_{f1} \) is the fin pitch that is usually obtained by making inverse of the fin density. The total number of fins \( n_{f1} \) is based on the shaded area (a-c-d-e-f-g-h-j-a) in Figure 5.41 counting as the unit fin. Since total primary area = total plate areas – fin base areas + passage side wall areas + passage front and back wall areas, the primary area for fluid 1 is expressed by

\[
A_{p1} = \frac{2L_1L_2N_p}{\text{total plate area}} - \frac{2\delta_2 n_{f1}}{\text{fin base area}} + \frac{2b_1L_2N_p}{\text{passage side wall area}} + \frac{2(b_2 + 2\delta_w)L_1(N_p + 1)}{\text{passage front and back wall area}}
\]  

(5.246)

The number of offset strip fins \( n_{off1} \) per the number of fins is obtained by

\[
n_{off1} = \frac{L_1}{\lambda_2}
\]

(5.247)

\[
n_{off2} = \frac{L_2}{\lambda_1}
\]

(5.247a)

where \( \lambda_{1,2} \) is the offset strip fin length for fluid 1 and 2. The total fin area \( A_{f1} \) consists of the fin area and offset-strip edge areas.

\[
A_{f1} = \underbrace{2(b_1 - \delta)L_2 n_{f1}}_{\text{fin surface areas}} + \underbrace{2(b_1 - \delta)\delta n_{off1} n_{f1}}_{\text{offset-strip edge areas}} + \underbrace{(p_{f1} - \delta)\delta(n_{off1} - 1)n_{f1}}_{\text{internal offset-strip edge area}} + \underbrace{2p_{f1} \delta n_{f1}}_{\text{1st & last offset-strip edge area}}
\]

(5.248)

Note that the cross-shaded area (a-b-l-k-a) was not included in the offset-strip edge area because the area is closely blocked by the next strip fin as shown in Figure 5.41 so that no heat transfer at the area is expected. The total heat transfer area \( A_{tl} \) is the sum of the primary area and the fin area as

\[
A_{tl} = A_{p1} + A_{f1}
\]

(5.249)

The free-flow (cross sectional) area \( A_{c1} \) is obtained by

\[
A_{c1} = (b_1 - \delta)(p_{f1} - \delta)n_{f1}
\]

(5.250)

It is assumed in Equation (5.250) that there exists a small gap between the offset strip fins, whereby the next strip fin shown in the unit fin in Figure 5.41 is not considered as an obstructing structure to the free flow. The frontal area \( A_{fr1} \) for fluid 1 where fluid 1 is entering is defined by

\[
A_{fr1} = L_1L_3
\]

(5.251)
The hydraulic diameter for fluid 1 is generally defined by

\[ D_{h1} = \frac{4A_{1}L_{2}}{A_{1}} \] (5.252)

For the fin efficiency \( \eta_f \) of the offset strip fin, it is assumed that the heat flow from both plates is uniform and the adiabatic plane occurs at the middle of the plate spacing \( b_f \). Hence, the fin profile length \( L_{f1} \) is defined by

\[ L_{f1} = \frac{b_f}{2} - \delta \] (5.253)

The \( m \) value is obtained using Equation (5.96) as

\[ m_i = \sqrt{\frac{2h}{k_f \delta \left( 1 + \frac{2h}{k_f \delta} \right)}} \] (5.254)

The single fin efficiency \( \eta_f \) is obtained using Equation (5.95) by

\[ \eta_f = \frac{\tanh(m_i L_{f1})}{m_i L_{f1}} \] (5.255)

The overall surface (fin) efficiency \( \eta_o \) is then obtained using Equations (5.99), (5.248) and (5.249) as

\[ \eta_o = 1 - \frac{A_{f1}}{A_{1}} (1 - \eta_f) \] (5.256)

### 5.8.2 Correlations for Offset Strip Fin (OSF) Geometry

This geometry has one of the highest heat transfer performances relative to the friction factors. Extensive analytical, numerical and experimental investigations have been conducted over the last 50 years. The most comprehensive correlations for \( j \) and \( f \) factors for the laminar, transition, and turbulent regions are provided by Manglik and Bergles [27] in 1995 as follows.
where the hydraulic diameter was defined by them as

\[
D_{h_{MB}} = \frac{4(p_f - \delta)(b - \delta)\lambda}{2((p_f - \delta)\lambda + (b - \delta)\lambda + (b - \delta)\delta + (p_f - \delta)\delta)}
\]

(5.258)

This is an approximation of Equation (5.252). However, Equation (5.258) is in excellent agreement with Equation (5.252), which indicates the general definition of hydraulic diameter, Equation (5.252), can be used in place of Equation (5.258). The correlations of Manglik and Bergles [27] were compared with the experiments (surface 1/8A19.86) reported by Kays and London [8] in Figure 5.42. The comparison for both \(j\) and \(f\) shows good agreement. Note that the comparison covers from laminar through turbulent regions.
Figure 5.42 Colburn factor $j$ and friction factor $f$ for offset-strip-fin (OSF) type plate-fin heat exchangers. The correlation of Manglik and Bergles [27] were compared with the experiments (1/8-19.86 surface) of Kays and London [8].
Example 5.8.1 Plate-Fin Heat Exchanger

A gas-to-air single-pass crossflow heat exchanger is designed for heat recovery from the exhaust gas to preheat incoming air in a solid oxide fuel cell (SOFC) co-generation system. Offset strip fins of the same geometry are employed on the gas and air sides; the geometrical properties and surface characteristics are provided in Figures E5.8.1 and E5.8.2. Both fins and plates (parting sheets) are made from Inconel 625 with $k=18 \text{ W/mK}$. The anode gas flows in the heat exchanger at $3.494 \text{ m}^3/\text{s}$ and $900^\circ\text{C}$. The cathode air on the other fluid side flows at $1.358 \text{ m}^3/\text{s}$ and $200^\circ\text{C}$. The inlet pressure of the gas is at $160 \text{ kPa}$ absolute whereas that of air is at $200 \text{ kPa}$ absolute. Both the gas and air pressure drops are limited to $10 \text{ kPa}$. It is desirable to have an equal length for $L_1$ and $L_2$. Design a gas-to-air single-pass crossflow heat exchanger operating at $\varepsilon=0.824$. Determine the core dimensions of this exchanger. Then, determine the heat transfer rate, outlet fluid temperatures and pressure drops on each fluid. Use the properties of air for the gas. Use also the following geometric information given.

<table>
<thead>
<tr>
<th>Description</th>
<th>value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fin thickness $\delta$</td>
<td>0.102 mm</td>
</tr>
<tr>
<td>Plate thickness $\delta_w$</td>
<td>0.5 mm</td>
</tr>
<tr>
<td>Fin density $N_f$</td>
<td>782 $m^{-1}$</td>
</tr>
<tr>
<td>Spacing between plates $b_1$ and $b_2$</td>
<td>2.49 mm</td>
</tr>
<tr>
<td>Offset strip length $\lambda_1$ and $\lambda_2$</td>
<td>3.175 mm</td>
</tr>
</tbody>
</table>

Figure E5.8.1 Plate-fin heat exchanger, employing offset strip fin.
MathCAD format solution:

Design concept is to develop a MathCAD model as a function of the dimensions of the sizing and then solve the sizing with the design requirements.

Properties:
We will use the arithmetic average temperatures as the appropriate mean temperature on each side by assuming the outlet temperatures at this time.

\[
T_{\text{gas}} := \frac{900^\circ C + 400^\circ C}{2} = 923.15K \quad T_{\text{air}} := \frac{200^\circ C + 600^\circ C}{2} = 673.15K
\]  
\text{(E5.8.1)}

Gas (subscript 1) \quad \text{Air (subscript 2)}
\[
\rho_1 = \text{defined}\_\text{later} \quad \rho_2 = \text{defined}\_\text{later}
\]
\text{(E5.8.2)}

\[
c_{p1} := 1126 \frac{J}{\text{kg.K}} \quad c_{p2} := 1073 \frac{J}{\text{kg.K}}
\]

\[
k_1 := 0.063 \frac{W}{\text{m.K}} \quad k_2 := 0.0524 \frac{W}{\text{m.K}}
\]

\[
\mu_1 := 40.1 \times 10^{-6} \frac{\text{N.s}}{\text{m}^2} \quad \mu_2 := 33.6 \times 10^{-6} \frac{\text{N.s}}{\text{m}^2}
\]

\[
Pr_1 := 0.731 \quad Pr_2 := 0.694
\]

The thermal conductivities of the fins and walls are given as

\[
k_f := 18 \frac{W}{\text{m.K}} \quad k_w := 18 \frac{W}{\text{m.K}}
\]  
\text{(E5.8.3)}
Given information:

\[ T_{1i} := 900^\circ C \quad \text{gas inlet temperature} \]  
\[ T_{2i} := 200^\circ C \quad \text{air inlet temperature} \]  
\[ P_{1i} := 160kPa \quad \text{gas inlet pressure} \]  
\[ P_{2i} := 200kPa \quad \text{air inlet pressure} \]  
\[ Q_1 := 3.494 \frac{m^3}{s} \quad \text{volume flow rate at gas side} \]  
\[ Q_2 := 1.358 \frac{m^3}{s} \quad \text{volume flow rate at air side} \]  

Air and gas densities (inlet):

The gas constants for air is known to be

\[ R_2 := 287.04 \frac{J}{kg \cdot K} \]  

(E5.8.10)

We calculate the air and gas inlet densities using the ideal gas law with an assumption that the gas constants for air and gas are equal.

assuming \( R_1 := R_2 \)  

(E5.8.11)

\[ \rho_{1i} := \frac{P_{1i}}{R_1 T_{1i}} \quad \rho_{1i} = 0.475 \frac{kg}{m^3} \]  

(E5.8.12)

\[ \rho_{2i} := \frac{P_{2i}}{R_2 T_{2i}} \quad \rho_{2i} = 1.473 \frac{kg}{m^3} \]  

(E5.8.13)

Hence, the mass flow rate for each fluid is calculated as

\[ \text{mdot}_1 := \rho_{1i} Q_1 \quad \text{mdot}_1 = 1.66 \frac{kg}{s} \]  

(E5.8.14)

\[ \text{mdot}_2 := \rho_{2i} Q_2 \quad \text{mdot}_2 = 2 \frac{kg}{s} \]  

(E5.8.15)
Design requirements:

In this design problem, the effectiveness is major concern for heat recovery. The designers want to derive the relationship between the effectiveness and the sizing. The frontal widths for both fluids are required to be the same \((L_1=L_2)\).

\[
\varepsilon = 0.824 \quad \text{effectiveness} \\
\Delta P_1 \leq 10\text{kPa} \quad \text{pressure drop at gas side} \\
\Delta P_2 \leq 10\text{kPa} \quad \text{pressure drop at air side} \\
L_1 = L_2 \quad \text{desirable}
\]  

Geometric information:

\[
\delta := 0.102\text{mm} \quad \text{fin thickness} \\
\delta_w := 0.5\text{mm} \quad \text{plate thickness} \\
N_f := 782\text{m}^{-1} \quad \text{fin density} \\
p_{f1} := \frac{1}{N_f} \quad \text{fin pitch} \\
p_{f2} := p_{f1} \quad p_{f1} = 1.2788\text{mm} \\
b_1 := 2.49\text{mm} \quad \text{plate distance} \\
b_2 := 2.49\text{mm} \\
\lambda_1 := 3.175\text{mm} \quad \text{offset strip length} \\
\lambda_2 := 3.175\text{mm}
\] 

Assume the sizing of the heat exchanger

Initially, assume the sizing to have the instant values and update later the dimensions when the final sizing is determined.

\[
L_1 := 0.303\text{m} \quad L_2 := 0.303\text{m} \quad L_3 := 0.948\text{m}
\] 

It is assumed that the number of passages for the gas side and the air side are \(N_p\) and \(N_p+1\), respectively, to minimize the heat loss to the ambient. Using Equation (5.243), \(L_3\) is calculated by
Thus, the total number of passages $N_p$ for the gas side is obtained using Equation (5.244)

$$N_p(L_3) := \frac{L_3 - b_2 - 2 \delta_w}{b_1 + b_2 + 2 \delta_w}$$

(E5.8.30)

Note that the total number of passages is a function of $L_3$. The total number of fins for each fluid is obtained

$$n_{f1}(L_1, L_3) := \frac{L_1}{p_{f1}} N_p(L_3)$$

$$n_{f1}(L_1, L_3) = 3.7424 \times 10^4$$

(E5.8.31)

$$n_{f2}(L_2, L_3) := \frac{L_2}{p_{f2}} (N_p(L_3) + 1)$$

$$n_{f2}(L_2, L_3) = 3.7661 \times 10^4$$

(E5.8.32)

The total primary area $A_p$ for each fluid is calculated using Equation (5.246).

$$A_p(L_1, L_2, L_3) := 2 L_1 L_2 N_p(L_3) - 2 \delta_1 L_2 n_{f1}(L_1, L_3) + 2 b_1 L_2 N_p(L_3) + 2 (b_2 + 2 \delta_w) L_1 (N_p(L_3) + 1)$$

(E5.8.33)

$$A_p(L_1, L_2, L_3) := 2 L_1 L_2 (N_p(L_3) + 1) - 2 \delta_1 n_{f2}(L_2, L_3) + 2 b_2 L_1 (N_p(L_3) + 1) + 2 (b_1 + 2 \delta_w) L_2 N_p(L_3)$$

(E5.8.34)

$$A_p(L_1, L_2, L_3) = 27.263 m^2$$

$$A_p(L_1, L_2, L_3) = 27.431 m^2$$

(E5.8.35)

The number of offset strip fins for each fluid (per the number of fins) is obtained

$$n_{off1}(L_2) := \frac{L_2}{\lambda_1}$$

$$n_{off2}(L_1) := \frac{L_1}{\lambda_2}$$

(E5.8.36)

The total fin area $A_f$ for each fluid is obtained using Equation (5.248) as

$$A_f(L_1, L_2, L_3) := 2 (b_1 - \delta) L_2 n_{f1}(L_1, L_3) + (p_{f1} - \delta) \delta (n_{off1}(L_2) - 1) n_{f1}(L_1, L_3) + 2 p_{f1} \delta n_{f1}(L_1, L_3)$$

(E5.8.37)
\[
A_{f2}(L_1, L_2, L_3) = 2 \left( b_2 - \delta \right) L_1 n_{f2}(L_2, L_3) + (p_{f2} - \delta) \delta \left( n_{off2}(L_1) - 1 \right) n_{f2}(L_2, L_3) + 2 p_{f2} \delta n_{f2}(L_2, L_3)
\]

(E5.8.38)

\[
A_{f1}(L_1, L_2, L_3) = 54.592m^2 \quad A_{f2}(L_1, L_2, L_3) = 54.937m^2
\]

(E5.8.39)

The total surface area \( A_t \) for each fluid is the sum of the fin area \( A_f \) and the primary area \( A_p \).

\[
A_{t1}(L_1, L_2, L_3) := A_{f1}(L_1, L_2, L_3) + A_{p1}(L_1, L_2, L_3)
\]

(E5.8.40)

\[
A_{t2}(L_1, L_2, L_3) := A_{f2}(L_1, L_2, L_3) + A_{p2}(L_1, L_2, L_3)
\]

(E5.8.41)

\[
A_{t1}(L_1, L_2, L_3) = 81.855m^2 \quad A_{t2}(L_1, L_2, L_3) = 82.369m^2
\]

(E5.8.42)

The free-flow area \( A_c \) for each side is obtained using Equation (5.250) as

\[
A_{c1}(L_1, L_3) := (b_1 - \delta)(p_{f1} - \delta)n_{f1}(L_1, L_3) \quad A_{c1}(L_1, L_3) = 0.1052m^2
\]

(E5.8.43)

\[
A_{c2}(L_2, L_3) := (b_2 - \delta)(p_{f2} - \delta)n_{f2}(L_2, L_3) \quad A_{c2}(L_2, L_3) = 0.1058m^2
\]

(E5.8.44)

The frontal area for each fluid is defined using Equation (5.251) by

\[
A_{fr1}(L_1, L_3) := L_1L_3 \quad A_{fr1}(L_1, L_3) = 0.287m^2
\]

(E5.8.45)

\[
A_{fr2}(L_2, L_3) := L_2L_3 \quad A_{fr2}(L_2, L_3) = 0.287m^2
\]

(E5.8.46)

The hydraulic diameter is defined using Equation (5.252) by

\[
D_{h1}(L_1, L_2, L_3) := \frac{4 A_{c1}(L_1, L_3)L_2}{A_{t1}(L_1, L_2, L_3)} \quad D_{h1}(L_1, L_2, L_3) = 1.557mm
\]

(E5.8.47)

\[
D_{h2}(L_1, L_2, L_3) := \frac{4 A_{c2}(L_2, L_3)L_1}{A_{t2}(L_1, L_2, L_3)} \quad D_{h2}(L_1, L_2, L_3) = 1.557mm
\]

(E5.8.48)

The hydraulic diameter of Equation (5.258) defined by Manglik and Bergles [27] is calculated in Equation (E5.8.49) to compare with the general definition of Equation (E5.8.47). We find that they are in good agreement. Therefore, we use Equations (E5.8.47) and (E5.8.48) in the rest of the calculations.
The porosity $\sigma$ for each fluid is calculated as

$$
\sigma_1(L_1, L_3) := \frac{A_{c1}(L_1, L_3)}{A_{fr1}(L_1, L_3)} \quad \sigma_1(L_1, L_3) = 0.366
$$

(E5.8.50)

$$
\sigma_2(L_2, L_3) := \frac{A_{c2}(L_2, L_3)}{A_{fr2}(L_2, L_3)} \quad \sigma_2(L_2, L_3) = 0.368
$$

(E5.8.51)

The volume of the exchanger for each fluid is calculated as

$$
V_{p1}(L_1, L_2, L_3) := b_1 L_1 L_2 N_p(L_3) \quad V_{p1}(L_1, L_2, L_3) = 0.0361 \text{m}^3
$$

(E5.8.52)

$$
V_{p2}(L_1, L_2, L_3) := b_2 L_1 L_2 (N_p(L_3) + 1) \quad V_{p2}(L_1, L_2, L_3) = 0.0363 \text{m}^3
$$

(E5.8.53)

The surface area density $\beta$ for each side is calculated as

$$
\beta_1(L_1, L_2, L_3) := \frac{A_{t1}(L_1, L_2, L_3)}{V_{p1}(L_1, L_2, L_3)} \quad \beta_1(L_1, L_2, L_3) = 2267 \text{m}^2 \text{m}^3
$$

(E5.8.54)

$$
\beta_2(L_1, L_2, L_3) := \frac{A_{t2}(L_1, L_2, L_3)}{V_{p2}(L_1, L_2, L_3)} \quad \beta_2(L_1, L_2, L_3) = 2267 \text{m}^2 \text{m}^3
$$

(E5.8.55)

The mass velocities, velocities, Reynolds Numbers are calculated as

$$
G_1(L_1, L_3) := \frac{mdot_1}{A_{c1}(L_1, L_3)} \quad G_1(L_1, L_3) = 15.786 \frac{\text{kg}}{\text{m}^2 \text{s}}
$$

(E5.8.56)

$$
G_2(L_2, L_3) := \frac{mdot_2}{A_{c2}(L_2, L_3)} \quad G_2(L_2, L_3) = 18.896 \frac{\text{kg}}{\text{m}^2 \text{s}}
$$

(E5.8.57)

$$
Re_1(L_1, L_2, L_3) := \frac{G_1(L_1, L_3) D_{h1}(L_1, L_2, L_3)}{\mu_1} \quad Re_1(L_1, L_2, L_3) = 613.002
$$

(E5.8.58)
\[
\text{Re}_2(L_1, L_2, L_3) = \frac{G_2(L_2, L_3) \cdot D_{h_2}(L_1, L_2, L_3)}{\mu_2} \quad \text{Re}_2(L_1, L_2, L_3) = 875.77
\]

(E5.8.59)

**Colburn factors and friction factors**

The Reynolds numbers indicate laminar flows at both the gas and air sides. \(J\) and \(f\) factors are calculated using Equations (5.257) and (5.258) as

\[
\begin{align*}
J_{1a}(L_1, L_2, L_3) &= 0.6522 \text{Re}_1(L_1, L_2, L_3)^{-0.5403} \left( \frac{P_f - \delta}{b_1 - \delta} \right)^{-0.1541} \left( \frac{\delta}{\lambda_1} \right)^{0.1499} \left( \frac{\delta}{P_f - \delta} \right)^{-0.0678} \\
J_{1b}(L_1, L_2, L_3) &= \left[ 1 + 5.269 \times 10^{-5} \cdot \text{Re}_1(L_1, L_2, L_3)^{1.34} \left( \frac{P_f - \delta}{b_1 - \delta} \right)^{0.504} \left( \frac{\delta}{\lambda_1} \right)^{0.456} \left( \frac{\delta}{P_f - \delta} \right)^{-1.055} \right]^{0.1} \\
J_1(L_1, L_2, L_3) &= J_{1a}(L_1, L_2, L_3) \cdot J_{1b}(L_1, L_2, L_3) \quad J_1(L_1, L_2, L_3) = 0.017
\end{align*}
\]

(E5.8.60)

(E5.8.61)

The Colburn factor \(J_1\) at the gas side is calculated by

\[
J_1(L_1, L_2, L_3) = J_{1a}(L_1, L_2, L_3) \cdot J_{1b}(L_1, L_2, L_3)
\]

(E5.8.62)

\[
\begin{align*}
f_{1a}(L_1, L_2, L_3) &= 9.6243 \text{Re}_1(L_1, L_2, L_3)^{-0.7422} \left( \frac{P_f - \delta}{b_1 - \delta} \right)^{-0.1856} \left( \frac{\delta}{\lambda_1} \right)^{0.3053} \left( \frac{\delta}{P_f - \delta} \right)^{-0.2659} \\
f_{1b}(L_1, L_2, L_3) &= \left[ 1 + 7.669 \times 10^{-8} \cdot \text{Re}_1(L_1, L_2, L_3)^{4.429} \left( \frac{P_f - \delta}{b_1 - \delta} \right)^{0.92} \left( \frac{\delta}{\lambda_1} \right)^{3.767} \left( \frac{\delta}{P_f - \delta} \right)^{0.236} \right]^{0.1} \\
f_1(L_1, L_2, L_3) &= f_{1a}(L_1, L_2, L_3) \cdot f_{1b}(L_1, L_2, L_3)
\end{align*}
\]

(E5.8.63)

(E5.8.64)

The friction factor \(f_1\) at the gas side is

\[
f_1(L_1, L_2, L_3) = f_{1a}(L_1, L_2, L_3) \cdot f_{1b}(L_1, L_2, L_3) \quad f_1(L_1, L_2, L_3) = 0.064
\]

(E5.8.65)

\[
\begin{align*}
j_{2a}(L_1, L_2, L_3) &= 0.6522 \text{Re}_2(L_1, L_2, L_3)^{-0.5403} \left( \frac{P_f - \delta}{b_2 - \delta} \right)^{-0.1541} \left( \frac{\delta}{\lambda_2} \right)^{0.1499} \left( \frac{\delta}{P_f - \delta} \right)^{-0.0678} \\
j_2(L_1, L_2, L_3) &= j_{2a}(L_1, L_2, L_3)
\end{align*}
\]

(E5.8.66)
\[ j_{2b}(L_1, L_2, L_3) := \left[ 1 + 5.269 \times 10^{-5} \cdot \text{Re}^2 L_1 L_2 L_3^{1.34} \left( \frac{p_{f2} - \delta}{b_2 - \delta} \right)^{0.504} \left( \frac{\delta}{\lambda_2} \right)^{0.456} \left( \frac{\delta}{p_{f2} - \delta} \right)^{-1.055} \right]^{0.1} \]

(E5.8.67)

The Colburn factor \( j_2 \) at the air side is

\[ j_2(L_1, L_2, L_3) := j_{2a}(L_1, L_2, L_3) \cdot j_{2b}(L_1, L_2, L_3) \quad j_2(L_1, L_2, L_3) = 0.014 \]

(E5.8.68)

\[ f_{2a}(L_1, L_2, L_3) := 9.6243 \cdot \text{Re}^2 L_1 L_2 L_3^{9.742} \left( \frac{p_{f2} - \delta}{b_2 - \delta} \right)^{-0.1856} \left( \frac{\delta}{\lambda_2} \right)^{0.3053} \left( \frac{\delta}{p_{f2} - \delta} \right)^{-0.2659} \]

(E5.8.69)

\[ f_{2b}(L_1, L_2, L_3) := \left[ 1 + 7.669 \times 10^{-8} \cdot \text{Re}^2 L_1 L_2 L_3^{4.429} \left( \frac{p_{f2} - \delta}{b_2 - \delta} \right)^{0.92} \left( \frac{\delta}{\lambda_2} \right)^{3.767} \left( \frac{\delta}{p_{f2} - \delta} \right)^{0.236} \right]^{0.1} \]

(E5.8.70)

The friction factor \( f_2 \) at the air side is

\[ f_2(L_1, L_2, L_3) := f_{2a}(L_1, L_2, L_3) \cdot f_{2b}(L_1, L_2, L_3) \quad f_2(L_1, L_2, L_3) = 0.05 \]

(E5.8.71)

**Heat Transfer Coefficients**

The heat transfer coefficients in terms of the \( j \) and \( f \) factors are calculated using Equation (5.230) as

\[ h_1(L_1, L_2, L_3) := \frac{j_1(L_1, L_2, L_3) \cdot G_1(L_1, L_3) \cdot c_{p1}}{\frac{2}{\text{Pr}_1^3}} \quad h_1(L_1, L_2, L_3) = 365.99 \text{ W/m}^2\text{K} \]

(E5.8.72)

\[ h_2(L_1, L_2, L_3) := \frac{j_2(L_1, L_2, L_3) \cdot G_2(L_2, L_3) \cdot c_{p2}}{\frac{2}{\text{Pr}_2^3}} \quad h_2(L_1, L_2, L_3) = 363.515 \text{ W/m}^2\text{K} \]

(E5.8.73)

**Overall surface (fin) efficiency**

Since the offset strip fins are used on both the gas and air sides, we will use the multiple fin analysis. \( m \) values are obtained using Equation (5.254) as
Using Equation (5.253), the fin length $L_f$ for each fluid will be

$$L_{f1} := \frac{b_1}{2} - \delta$$

$$L_{f2} := \frac{b_2}{2} - \delta$$

(E5.8.76)

The single fin efficiency $\eta_f$ for each fluid is calculated using Equation (5.255) as

$$\eta_{f1}(L_1, L_2, L_3) := \frac{\tanh(m_1(L_1, L_2, L_3)L_{f1})}{m_1(L_1, L_2, L_3)L_{f1}}$$

$$\eta_{f2}(L_1, L_2, L_3) := \frac{\tanh(m_2(L_1, L_2, L_3)L_{f2})}{m_2(L_1, L_2, L_3)L_{f2}}$$

(E5.8.77)

The overall surface (fin) efficiencies $\eta_o$ are obtained using Equation (E5.256)

$$\eta_{o1}(L_1, L_2, L_3) := 1 - (1 - \eta_{f1}(L_1, L_2, L_3))\frac{A_{f1}(L_1, L_2, L_3)}{A_{t1}(L_1, L_2, L_3)}$$

$$\eta_{o2}(L_1, L_2, L_3) := 1 - (1 - \eta_{f2}(L_1, L_2, L_3))\frac{A_{f2}(L_1, L_2, L_3)}{A_{t2}(L_1, L_2, L_3)}$$

(E5.8.78)

(E5.8.79)

The conduction area for the wall thermal resistance is given by

$$A_w(L_1, L_2, L_3) := 2L_1L_2(N_p(L_3) + 1)$$

$$A_w(L_1, L_2, L_3) = 29.185m^2$$

(E5.8.80)

The thermal resistance of the wall is given by

$$R_w(L_1, L_2, L_3) := \frac{\delta_w}{k_wA_w(L_1, L_2, L_3)}$$

(E5.8.81)

$$UA_t(L_1, L_2, L_3) := \frac{1}{\eta_{o1}(L_1, L_2, L_3)h(t_1, L_3)L_{f1}(L_1, L_2, L_3)}$$

(E5.8.82)
\[
UA_2(L_1, L_2, L_3) := \frac{1}{\eta o_2(L_1, L_2, L_3) h_2(L_1, L_2, L_3) A_t_2(L_1, L_2, L_3)}
\]

\[
UA(L_1, L_2, L_3) := \frac{1}{UA_1(L_1, L_2, L_3) + R_w(L_1, L_2, L_3) + UA_2(L_1, L_2, L_3)}
\]

(E5.8.83)

(E5.8.84)

The overall heat transfer coefficient times the heat transfer area is given by

\[
UA(L_1, L_2, L_3) = 13334m^2 \cdot \frac{W}{m^2 \cdot K}
\]

(E5.8.84a)

\textbf{ε-NTU Method}

The heat capacity rates are given as

\[
C_1 := \text{mdot}_1 \cdot c_{p1} \quad C_1 = 1.869 \times 10^3 \frac{W}{K}
\]

(E5.8.85)

\[
C_2 := \text{mdot}_2 \cdot c_{p2} \quad C_2 = 2.146 \times 10^3 \frac{W}{K}
\]

(E5.8.86)

\[
C_{\text{min}} := \min(C_1, C_2) \quad C_{\text{min}} = 1.869 \times 10^3 \frac{W}{K}
\]

(E5.8.87)

\[
C_{\text{max}} := \max(C_1, C_2) \quad C_{\text{max}} = 2.146 \times 10^3 \frac{W}{K}
\]

(E5.8.88)

\[
C_r := \frac{C_{\text{min}}}{C_{\text{max}}} \quad C_r = 0.871
\]

(E5.8.89)

The number of transfer unit (NTU) is calculated using Equation (5.77) by

\[
\text{NTU}(L_1, L_2, L_3) := \frac{UA(L_1, L_2, L_3)}{C_{\text{min}}} \quad \text{NTU}(L_1, L_2, L_3) = 7.133
\]

(E5.8.90)

The effectiveness of the plate-fin heat exchanger is calculated using Equation (5.38) by

\[
\varepsilon_{\text{hx}}(L_1, L_2, L_3) := 1 - \exp\left[\frac{1}{C_r \cdot \text{NTU}(L_1, L_2, L_3)^{0.22}} \left(\exp\left(-C_r \cdot \text{NTU}(L_1, L_2, L_3)^{0.78}\right) - 1\right)\right]
\]

\[
\varepsilon_{\text{hx}}(L_1, L_2, L_3) = 0.824
\]

(E5.8.91)
The heat transfer rate is calculated by

\[ q(L_1, L_2, L_3) := \varepsilon_{hx}(L_1, L_2, L_3) C_{\text{min}} \left( T_{1i} - T_{2i} \right) \]

\[ q(L_1, L_2, L_3) = 1.078 \times 10^6 \text{ W} \]

(E5.8.92)

The air outlet temperatures are calculated by

\[ T_{1o}(L_1, L_2, L_3) := T_{1i} - \varepsilon_{hx}(L_1, L_2, L_3) C_{\text{min}} \frac{1}{C_1} \left( T_{1i} - T_{2i} \right) \]

\[ T_{1o}(L_1, L_2, L_3) = 323.203^\circ C \]

(E5.8.93)

\[ T_{2o}(L_1, L_2, L_3) := T_{2i} + \varepsilon_{hx}(L_1, L_2, L_3) C_{\text{min}} \frac{1}{C_2} \left( T_{1i} - T_{2i} \right) \]

\[ T_{2o}(L_1, L_2, L_3) = 702.481^\circ C \]

(E5.8.94)

**Pressure Drops**

The pressure drops for the plate-fin heat exchanger is expressed using Equation (5.202) as

\[ \Delta P = \frac{G^2}{2 \rho_i} \left[ 1 - \sigma^2 + K_c \right] + 2 \left( \frac{\rho_i}{\rho_o} - 1 \right) + \frac{4 \varepsilon L}{D_h} \left( \frac{\rho_i}{\rho_m} \right) \left( 1 - \sigma^2 - K_e \right) \left( \frac{\rho_i}{\rho_o} \right) \]

(E5.8.95)

**Contraction and Expansion Coefficients, K_c and K_e**

We develop equations for Figure 5.31. The jet contraction ratio for the square-tube geometry is given using Equation (5.210) by

\[ C_{c\_tube}(\sigma) := 4.374 \times 10^{-4} e^{6.737 \sqrt{\sigma}} + 0.621 \]

(E5.8.96)

The friction factor used for this calculation is given in Equation (5.205).

\[ f_d(Re) := \begin{cases} 0.049 Re^{-0.2} & \text{if } Re \geq 2300 \\ \frac{16}{Re} & \text{otherwise} \end{cases} \]

(E5.8.97)

The velocity-distribution coefficient for circular tubes is given in Equation (5.207).

\[ K_{d\_tube}(Re) := \begin{cases} 1.09068 \left( 4 f_d(Re) \right) + 0.05884 \sqrt{4 f_d(Re)} + 1 & \text{if } Re \geq 2300 \\ 1.33 & \text{otherwise} \end{cases} \]

(E5.8.98)
which is converted to the square tubes as shown in Equation (5.209)

\[ K_{d\_square}(\text{Re}) := \begin{cases} 1 + 1.17(K_{d\_tube}(\text{Re}) - 1) & \text{if } \text{Re} \geq 2300 \\ 1.39 & \text{otherwise} \end{cases} \]  \hfill (E5.8.99)

The contraction coefficient is given in Equation (5.203).

\[ K_{c\_square}(\sigma,\text{Re}) := \frac{1 - 2C_{c\_tube}(\sigma) + C_{c\_tube}(\sigma)^2(2K_{d\_square}(\text{Re}) - 1)}{C_{c\_tube}(\sigma)^2} \]  \hfill (E5.8.100)

The expansion coefficient is given in Equation (5.204).

\[ K_{e\_square}(\sigma,\text{Re}) := 1 - 2K_{d\_square}(\text{Re})\sigma + \sigma^2 \]  \hfill (E5.8.101)

\( K_c \) and \( K_e \) for each fluid are determined with the porosity and the Reynolds number where the Reynolds number is assumed to be fully turbulent (Re=10^7) because of the frequent boundary layer interruptions due to the offset strip fins. The values may also be obtained graphically from Figure 5.31.

\[ K_{e1}(L_1,L_3) := K_{c\_square}(\sigma_1(L_1,L_3),10^7) \quad K_{c1}(L_1,L_3) = 0.33 \]  \hfill (E5.8.102)

\[ K_{e2}(L_2,L_3) := K_{c\_square}(\sigma_2(L_2,L_3),10^7) \quad K_{c2}(L_2,L_3) = 0.329 \]  \hfill (E5.8.103)

\[ K_{e1}(L_1,L_3) := K_{e\_square}(\sigma_1(L_1,L_3),10^7) \quad K_{e1}(L_1,L_3) = 0.39 \]  \hfill (E5.8.104)

\[ K_{e2}(L_2,L_3) := K_{e\_square}(\sigma_2(L_2,L_3),10^7) \quad K_{e2}(L_2,L_3) = 0.387 \]  \hfill (E5.8.105)

**Gas and air densities**

The gas and air outlet densities are a function of both the outlet temperatures and the outlet pressures that are unknown. The outlet densities are expressed using the ideal gas law as

\[ \rho_{1o}(P_{1o},L_1,L_2,L_3) := \frac{P_{1o}}{R_1T_{1o}(L_1,L_2,L_3)} \]  \hfill (E5.8.106)

\[ \rho_{2o}(P_{2o},L_1,L_2,L_3) := \frac{P_{2o}}{R_2T_{2o}(L_1,L_2,L_3)} \]  \hfill (E5.8.107)
Since $\rho_m$ is the mean density and defined as

$$\frac{1}{\rho_m} = \frac{1}{2} \left( \frac{1}{\rho_i} + \frac{1}{\rho_o} \right)$$  \hspace{1cm} \text{(E5.8.108)}$$

Thus, for each fluid,

$$\rho_{1m}(P_{10}, L_1, L_2, L_3) := \left[ \frac{1}{2} \left( \frac{1}{\rho_{1i}} + \frac{1}{\rho_{1o}(P_{10}, L_1, L_2, L_3)} \right) \right]^{-1}$$  \hspace{1cm} \text{(E5.8.109)}$$

$$\rho_{2m}(P_{20}, L_1, L_2, L_3) := \left[ \frac{1}{2} \left( \frac{1}{\rho_{2i}} + \frac{1}{\rho_{2o}(P_{20}, L_1, L_2, L_3)} \right) \right]^{-1}$$  \hspace{1cm} \text{(E5.8.110)}$$

**Pressure drops**

We divide Equation (E5.8.95) into several short equations to fit the present page.

$$\Delta P_{12}(L_1, L_3) := \frac{G_i(L_1, L_3)^2}{2\rho_{1i}} \left( 1 - \sigma_1(L_1, L_3)^2 + K_{c1}(L_1, L_3) \right)$$  \hspace{1cm} \text{(E5.8.111)}$$

$$K_{123}(P_{10}, L_1, L_2, L_3) := 2 \left( \frac{\rho_{1i}}{\rho_{1o}(P_{10}, L_1, L_2, L_3)} - 1 \right) + \frac{4f_i(L_1, L_2, L_3)L_2}{D_{h1}(L_1, L_2, L_3)} \left( \frac{\rho_{1i}}{\rho_{1m}(P_{10}, L_1, L_2, L_3)} \right)$$  \hspace{1cm} \text{(E5.8.112)}$$

$$\Delta P_{13}^i(P_{10}, L_1, L_2, L_3) := \frac{G_i(L_1, L_3)^2}{2\rho_{1i}} \cdot K_{123}(P_{10}, L_1, L_2, L_3)$$  \hspace{1cm} \text{(E5.8.113)}$$

$$\Delta P_{13}(P_{10}, L_1, L_2, L_3) := \frac{G_i(L_1, L_3)^2}{2\rho_{1i}} \left[ \left( 1 - \sigma_1(L_1, L_3)^2 - K_{e1}(L_1, L_3) \right) \frac{\rho_{1i}}{\rho_{1o}(P_{10}, L_1, L_2, L_3)} \right]$$  \hspace{1cm} \text{(E5.8.114)}$$

The pressure drop at the gas side, which is Equation (E5.8.95), is the sum of the above three equations, Equations (E5.8.111), (E5.8.113) and (E5.8.114) as

$$\Delta P_1(P_{10}, L_1, L_2, L_3) := \Delta P_{12}(L_1, L_3) + \Delta P_{123}(P_{10}, L_1, L_2, L_3) - \Delta P_{13}(P_{10}, L_1, L_2, L_3)$$  \hspace{1cm} \text{(E5.8.115)}$$

Now the gas outlet pressure can be obtained using the MathCAD’s 'root' function.
A guess value that would be a closest value to the solution is needed to be provided for the root function as

\[ P_{10} := 160 \text{kPa} \]  
(E5.8.116)

Solving

\[ P_{10}(L_1, L_2, L_3) := \text{root}[\left( P_{11} - P_{10} \right) - \Delta P_1 \left( P_{10}, L_1, L_2, L_3 \right)], P_{10} \]  
(E5.8.117)

The gas outlet pressure is finally obtained by

\[ P_{10}(L_1, L_2, L_3) = 149.996 \text{kPa} \]  
(E5.8.118)

The pressure drop at the gas side is calculated

\[ \Delta P_1(L_1, L_2, L_3) := P_{11} - P_{10}(L_1, L_2, L_3) \quad \Delta P_1(L_1, L_2, L_3) = 10.004 \text{kPa} \]  
(E5.8.119)

The air outlet pressure can be obtained in a similar way.

\[ \Delta P_{212}(L_2, L_3) := \frac{G_2(L_1, L_3)^2}{2 \rho_{2i}} \left( 1 - \sigma_2(L_2, L_3)^2 + K_{c2}(L_2, L_3) \right) \]  
(E5.8.120)

\[ K_{223}(P_{20}, L_1, L_2, L_3) := 2 \left( \rho_{2i} \right) \left( \frac{\rho_{2i}}{\rho_{20}(P_{20}, L_1, L_2, L_3)} - 1 \right) + \frac{4 f_2(L_1, L_2, L_3) L_1}{D_{h2}(L_1, L_2, L_3)} \left( \frac{\rho_{2i}}{\rho_{2m}(P_{20}, L_1, L_2, L_3)} \right) \]  
(E5.8.121)

\[ \Delta P_{223}(P_{20}, L_1, L_2, L_3) := \frac{G_2(L_1, L_3)^2}{2 \rho_{2i}} \cdot K_{223}(P_{20}, L_1, L_2, L_3) \]  
(E5.8.122)

\[ \Delta P_{234}(P_{20}, L_1, L_2, L_3) := \frac{G_2(L_1, L_3)^2}{2 \rho_{2i}} \left( 1 - \sigma_2(L_2, L_3)^2 - K_{c2}(L_2, L_3) \right) \frac{\rho_{2i}}{\rho_{20}(P_{20}, L_1, L_2, L_3)} \]  
(E5.8.123)

\[ \Delta P_{2}(P_{20}, L_1, L_2, L_3) := \Delta P_{212}(L_2, L_3) + \Delta P_{223}(P_{20}, L_1, L_2, L_3) - \Delta P_{234}(P_{20}, L_1, L_2, L_3) \]  
(E5.8.124)

A guess value is needed for MathCAD as
\[ P_{20} := 20 \text{kPa} \]  
(E5.8.125)

\[ P_{20}(L_1, L_2, L_3) := \text{root}[(P_{2i} - P_{20}) - \Delta P_2\left(P_{20}, L_1, L_2, L_3\right)], P_{20}] \]  
(E5.8.126)

\[ P_{20}(L_1, L_2, L_3) = 192.216 \text{kPa} \]  
(E5.8.127)

The pressure drop at the air side is calculated by

\[ \Delta P_2(L_1, L_2, L_3) := P_{2i} - P_{20}(L_1, L_2, L_3) \]

\[ \Delta P_2(L_1, L_2, L_3) = 7.784 \text{kPa} \]  
(E5.8.128)

**Sizing of the Plate-Fin Heat Exchanger**

The calculations for the sizing usually require a number of iterations, which is a time-consuming work. This can be easily performed by MathCAD. The three unknowns of \( L_1, L_2, \) and \( L_3 \) are found using a combination of three requirements. The higher pressure drop for fluid 1 (gas) is expected due to the higher volume flow rate. The following combination of the three requirements will be sufficient for the solution. Guess values can be updated and iterated with the new values to have a convergence.

**Guess values**

\[ L_1 := 1 \text{m}, \quad L_2 := 1 \text{m}, \quad L_3 := 1 \text{m} \]  
(E5.8.129)

**Given**

\[ \varepsilon_{\text{hx}}(L_1, L_2, L_3) = 0.824 \]  
(E5.8.130)

\[ \Delta P_1(L_1, L_2, L_3) = 10 \text{kPa} \]  
(E5.8.131)

\[ L_1 = L_2 \]  
(E5.8.132)

\[ \begin{bmatrix} \frac{L_1}{L_2} \\ \frac{L_2}{L_3} \end{bmatrix} := \text{Find}(L_1, L_2, L_3) \]  
(E5.8.133)

The dimensions of the sizing with the requirements are finally found as

\[ L_1 = 0.303 \text{m}, \quad L_2 = 0.303 \text{m}, \quad L_3 = 0.948 \text{m} \]  
(E5.8.134)

Now return to Equation (E5.8.28) and update the values for all the correct numerical values.

**Summary of the results and information**

Given information:
\[ T_{1i} = 900 \, ^\circ C \] gas inlet temperature (E5.8.136)

\[ T_{2i} = 200 \, ^\circ C \] air inlet temperature (E5.8.137)

\[ P_{1i} = 160 \, kPa \] gas inlet pressure (E5.8.138)

\[ P_{2i} = 200 \, kPa \] air inlet pressure (E5.8.139)

\[ Q_1 = \frac{3.494 \, m^3}{s} \] volume flow rate at gas side (E5.8.140)

\[ Q_2 = \frac{1.358 \, m^3}{s} \] volume flow rate at air side (E5.8.141)

Requirements:

\[ e_{hx} = 0.824 \] effectiveness (E5.8.142)

\[ \Delta P_1 \leq 10 \, kPa \] pressure drop at gas side (E5.8.143)

\[ \Delta P_2 \leq 10 \, kPa \] pressure drop at air side (E5.8.144)

\[ L_1 = L_2 \] desirable (E5.8.145)

Dimensions of the sizing:

\[ L_1 = 0.303 \, m \quad L_2 = 0.303 \, m \quad L_3 = 0.948 \, m \] (E5.8.146)

Outlet temperatures and pressure drops:

\[ T_{1o}(L_1, L_2, L_3) = 323.2 \, ^\circ C \] gas outlet temperature (E5.8.147)

\[ T_{2o}(L_1, L_2, L_3) = 702.484 \, ^\circ C \] air outlet temperature (E5.8.148)

\[ \Delta P_1(L_1, L_2, L_3) = 10 \, kPa \] pressure drop at gas side (E5.8.149)

\[ \Delta P_2(L_1, L_2, L_3) = 7.78 \, kPa \] pressure drop at air side (E5.8.150)

Densities of gas and air at inlet and outlet

\[ \rho_{1i} = \frac{0.475 \, kg}{m^3} \quad \rho_{1o}(P_{1o}(L_1, L_2, L_3), L_1, L_2, L_3) = \frac{0.876 \, kg}{m^3} \] (E5.8.151)
\[ \rho_{2i} = 1.473 \frac{kg}{m^3} \]
\[ \rho_{2o} = 0.686 \frac{kg}{m^3} \]

Thermal quantities:

\[ \text{NTU}(L_1, L_2, L_3) = 7.134 \]
\[ \varepsilon_{hx}(L_1, L_2, L_3) = 0.824 \]
\[ q(L_1, L_2, L_3) = 1.078 \times 10^6 \text{W} \]
\[ h_1(L_1, L_2, L_3) = 365.944 \frac{W}{m^2 \cdot K} \]
\[ h_2(L_1, L_2, L_3) = 363.469 \frac{W}{m^2 \cdot K} \]

Hydraulic quantities:

\[ D_{h1}(L_1, L_2, L_3) = 1.557 \text{mm} \]
\[ D_{h2}(L_1, L_2, L_3) = 1.557 \text{mm} \]
\[ G_1(L_1, L_3) = 15.782 \frac{kg}{m^2 \cdot s} \]
\[ G_2(L_2, L_3) = 18.891 \frac{kg}{m^2 \cdot s} \]
\[ \text{Re}_1(L_1, L_2, L_3) = 612.85 \]
\[ \text{Re}_2(L_1, L_2, L_3) = 875.555 \]

Geometric quantities:

\[ b_1 = 2.49 \text{mm} \]
\[ b_2 = 2.49 \text{mm} \]
\[ p_{f1} = 1.279 \text{mm} \]
\[ p_{f2} = 1.279 \text{mm} \]
\[ \lambda_1 = 3.175 \text{mm} \]
\[ \lambda_2 = 3.175 \text{mm} \]
\[ \delta = 0.102 \text{mm} \]
\[ \delta_w = 0.5 \text{mm} \]
\begin{align*}
N_p(L_3) &= 157.996 & \text{number of passages at gas side} & \text{(E5.8.169)} \\
N_p(L_3) + 1 &= 158.996 & \text{number of passages at air side} & \text{(E5.8.170)} \\
\sigma_1(L_1, L_3) &= 0.366 & \text{porosity} & \text{(E5.8.171)} \\
\sigma_2(L_2, L_3) &= 0.368 & \text{(E5.8.172)} \\
\beta_1(L_1, L_2, L_3) &= 2267 \frac{m^2}{m^3} & \text{surface area density} & \text{(E5.8.173)} \\
\beta_2(L_1, L_2, L_3) &= 2267 \frac{m^2}{m^3} & \text{(E5.8.174)} \\
A_{t1}(L_1, L_2, L_3) &= 81.869 m^2 & \text{total heat transfer area} & \text{(E5.8.175)} \\
A_{t2}(L_1, L_2, L_3) &= 82.383 m^2 & \text{(E5.8.176)} \\
\end{align*}

The results satisfies the design requirements.
References


Problems

Double Pipe Heat Exchanger

5.1 A counterflow double pipe heat exchanger is used to cool ethylene glycol for a chemical process with city water. Ethylene glycol at a flow rate of 0.63 kg/s is required to be cooled from 80°C to 65°C using water at a flow rate of 1.7 kg/s and 23°C, which shown in Figure P5.1. The double-pipe heat exchanger is composed of 2-m long carbon-steel hairpins. The inner and outer pipes are 3/4 and 1 1/2 nominal schedule 40, respectively. The ethylene glycol flows through the inner tube. When the heat exchanger is initially in service (no fouling), calculate the outlet temperatures, the heat transfer rate and the pressure drops for the exchanger. How many hairpins will be required?

![Figure P5.1 and P5.2 Double-pipe heat exchanger](image)

5.2 A counterflow double pipe heat exchanger is used to cool ethylene glycol for a chemical process with city water. Ethylene glycol at a flow rate of 0.63 kg/s is required to be cooled from 80°C to 65°C using water at a flow rate of 1.7 kg/s and 23°C, which is shown in Figure P5.2. The double-pipe heat exchanger is composed of 2-m long carbon-steel hairpins. The inner and outer pipes are 3/4 and 1 1/2 nominal schedule 40, respectively. The ethylene glycol flows through the inner tube. Fouling factors of $0.176 \times 10^{-3}$ m²K/W for water and $0.325 \times 10^{-3}$ m²K/W for ethylene glycol are specified. Calculate the outlet temperatures, the heat transfer rate and the pressure drops for the exchanger. How many hairpins will be required?
Shell-and-Tube Heat Exchanger

5.3 A miniature shell-and-tube heat exchanger is designed to cool glycerin with cold water. The glycerin at a flow rate of 0.25 kg/s enters the exchanger at 60°C and leaves at 36°C. The water at a rate of 0.54 kg/s enters at 18°C, which is shown in Figure P5.3. The tube material is carbon steel. Fouling factors of $0.253 \times 10^{-3} \text{ m}^2\text{K/W}$ for water and $0.335 \times 10^{-3} \text{ m}^2\text{K/W}$ for glycerin are specified. Route the glycerin through the tubes. The permissible maximum pressure drop on each side is 30 kPa. The volume of the exchanger is required to be minimized. Since the exchanger is custom designed, the tube size may be smaller than NPS 1/8 (DN 6 mm) that is the smallest size in Table C.6 in Appendix C, wherein the tube pitch ratio of 1.25 and the diameter ratio of 1.3 can be applied. Design the shell-and-tube heat exchanger.

![Figure P5.3 Shell-and-tube heat exchanger](image)

Plate Heat Exchanger

5.4 Hot water will be cooled by cold water using a miniature plate heat exchanger as shown in Figure P5.4.1. The hot water with a flow rate of 1.2 kg/s enters the plate heat exchanger at 75°C and it will be cooled to 55°C. The cold water enters at a flow rate of 1.3 kg/s and 24°C. The maximum permissible pressure drop for each stream is 30 kPa (4.5 psi). Using a single pass chevron plates (Stainless steel AISI 304) with $\beta = 45^\circ$, determine the rating ($T, q, \varepsilon$ and $\Delta P$) and sizing ($W_p, L_p$, and $H_p$) of the plate heat exchanger.

<table>
<thead>
<tr>
<th>Description</th>
<th>value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of passes $N_p$</td>
<td>1</td>
</tr>
<tr>
<td>Chevron angle $\beta$</td>
<td>40 degree</td>
</tr>
<tr>
<td>Total number of plates $N_t$</td>
<td>40</td>
</tr>
<tr>
<td>Plate thickness $\delta$</td>
<td>0.2 mm</td>
</tr>
<tr>
<td>Corrugation pitch $\lambda$</td>
<td>4 mm</td>
</tr>
<tr>
<td>Port diameter $D_p$</td>
<td>25 mm</td>
</tr>
<tr>
<td>Thermal conductivity $k_w$</td>
<td>14.9 W/mK</td>
</tr>
</tbody>
</table>
5.5 (Time-consuming work) Cold water will be heated by wastewater using a plate heat exchanger as shown in Figure P5.4.1. The cold water with a flow rate of 140 kg/s enters the plate heat exchanger at 22°C and it will be heated to 42°C. The hot wastewater enters at a flow rate of 130 kg/s and 65°C. The maximum permissible pressure drop for each fluid is 70 kPa (10 psi). Using a single pass chevron plates (Stainless steel AISI 304) with \( \beta = 30° \), determine firstly the number of plates \( N_t \) and the corrugation pitch \( \lambda \) by developing a MathCAD model as a function of not only the sizing (\( W_p, L_p, \) and \( H_p \)) but also \( N_t \) and \( \lambda \), and then determine the rating (\( T \), \( q \), \( \varepsilon \) and \( \Delta P \)) and sizing of the plate heat exchanger. Show your detailed work for the
determination of the number of plates and corrugation pitch. The following data list is used for the calculations.

<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of passes $N_p$</td>
<td>1</td>
</tr>
<tr>
<td>Chevron angle $\beta$</td>
<td>30 degree</td>
</tr>
<tr>
<td>Plate thickness $\delta$</td>
<td>0.6 mm</td>
</tr>
<tr>
<td>Port diameter $D_p$</td>
<td>200 mm</td>
</tr>
<tr>
<td>Thermal conductivity $k_w$</td>
<td>14.9 W/mK</td>
</tr>
</tbody>
</table>

**Finned-Tube Heat Exchanger**

5.6 A circular finned-tube heat exchanger is used to cool hot coolant (50% ethylene glycol) with an air stream in an engine. The coolant enters at 96°C with a flow rate of 1.8 kg/s and must leave at 90°C. The air stream enters at a flow rate of 2.5 kg/s and 20°C. The allowable air pressure loss is 250 Pa and the allowable coolant pressure loss is 500 Pa. The tubes of the exchanger have an equilateral triangular pitch. The thermal conductivities of the aluminum fins and the copper tubes are 200 W/mK and 385 W/mK, respectively. The tube pitch is 2.476 cm and the fin pitch is 3.43 fins/cm. The fin thickness is 0.046 cm and the three diameters are $d_i=0.765$ cm for the tube inside diameter, $d_o=0.965$ cm for the tube outside diameter, and $d_e=2.337$ cm for the fin outside diameter. Design the circular finned-tube heat exchanger by providing the dimensions of the exchanger, outlet temperatures, effectiveness, and pressure drops.

![Flow diagram of a circular finned-tube heat exchanger](image)

Figure P5.6.1 Flow diagram of a circular finned-tube heat exchanger
Plate-Fin Heat Exchanger

5.7 A gas-to-air single-pass crossflow heat exchanger is contemplated for heat recovery from the exhaust gas to preheat incoming air in a portable solid oxide fuel cell (SOFC) system. Offset strip fins of the same geometry are employed on the gas and air sides; the geometrical properties and surface characteristics are provided in Figures P5.7.1 and P5.7.2. Both fins and plates (parting sheets) are made from Inconel 625 with $k=18$ W/m.K. The anode gas flows in the heat exchanger at 0.075 m$^3$/s and 750°C. The cathode air on the other fluid side flows at 0.018 m$^3$/s and 90°C. The inlet pressure of the gas is at 130 kPa absolute whereas that of air is at 170 kPa absolute. Both the gas and air pressure drops are limited to 3 kPa. It is desirable to have an equal length for $L_1$ and $L_2$. Design a gas-to-air single-pass crossflow heat exchanger operating at $\epsilon =0.85$. Determine the core dimensions of this exchanger. Then, determine the heat transfer rate, outlet fluid temperatures and pressure drops on each fluid. Use the properties of air for the gas. Use the following geometric information.

<table>
<thead>
<tr>
<th>Description</th>
<th>value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fin thickness $\delta$</td>
<td>0.092 mm</td>
</tr>
<tr>
<td>Plate thickness $\delta_w$</td>
<td>0.25 mm</td>
</tr>
<tr>
<td>Fin density $N_f$</td>
<td>980 m$^{-1}$</td>
</tr>
<tr>
<td>Spacing between plates $b_1$ and $b_2$</td>
<td>1.3 mm</td>
</tr>
<tr>
<td>Offset strip length $\lambda_1$ and $\lambda_2$</td>
<td>1.7 mm</td>
</tr>
</tbody>
</table>

![Figure P5.7.1 Plate-fin heat exchanger, employing offset strip fin.](image-url)
Figure P5.7.2 Schematic of offset strip fin (OSF) geometry
## Appendix C

Table C.5 Thermophysical properties of fluids

<table>
<thead>
<tr>
<th>Engine oil</th>
<th>T (°C)</th>
<th>ρ (kg/m³)</th>
<th>cₚ (J/KgK)</th>
<th>k (W/mK)</th>
<th>µ x 10² (N.s/m²)</th>
<th>Pr</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0</td>
<td>899</td>
<td>1796</td>
<td>0.147</td>
<td>384.8</td>
<td>47100</td>
</tr>
<tr>
<td></td>
<td>20</td>
<td>888</td>
<td>1880</td>
<td>0.145</td>
<td>79.92</td>
<td>10400</td>
</tr>
<tr>
<td></td>
<td>40</td>
<td>876</td>
<td>1964</td>
<td>0.144</td>
<td>21.02</td>
<td>2870</td>
</tr>
<tr>
<td></td>
<td>60</td>
<td>864</td>
<td>2047</td>
<td>0.14</td>
<td>7.249</td>
<td>1050</td>
</tr>
<tr>
<td></td>
<td>80</td>
<td>852</td>
<td>2131</td>
<td>0.138</td>
<td>3.195</td>
<td>490</td>
</tr>
<tr>
<td></td>
<td>100</td>
<td>840</td>
<td>2219</td>
<td>0.137</td>
<td>1.705</td>
<td>276</td>
</tr>
<tr>
<td></td>
<td>120</td>
<td>828</td>
<td>2307</td>
<td>0.135</td>
<td>1.027</td>
<td>175</td>
</tr>
<tr>
<td></td>
<td>140</td>
<td>816</td>
<td>2395</td>
<td>0.133</td>
<td>0.653</td>
<td>116</td>
</tr>
<tr>
<td></td>
<td>160</td>
<td>805</td>
<td>2483</td>
<td>0.132</td>
<td>0.451</td>
<td>84</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>50% Ethylene glycol</th>
<th>T (°C)</th>
<th>ρ (kg/m³)</th>
<th>cₚ (J/KgK)</th>
<th>k (W/mK)</th>
<th>µ x 10² (N.s/m²)</th>
<th>Pr</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0</td>
<td>1083</td>
<td>3180</td>
<td>0.379</td>
<td>1.029</td>
<td>86.3</td>
</tr>
<tr>
<td></td>
<td>20</td>
<td>1072</td>
<td>3310</td>
<td>0.319</td>
<td>0.459</td>
<td>47.6</td>
</tr>
<tr>
<td></td>
<td>40</td>
<td>1061</td>
<td>3420</td>
<td>0.404</td>
<td>0.238</td>
<td>20.1</td>
</tr>
<tr>
<td></td>
<td>60</td>
<td>1048</td>
<td>3520</td>
<td>0.417</td>
<td>0.139</td>
<td>11.8</td>
</tr>
<tr>
<td></td>
<td>80</td>
<td>1034</td>
<td>3590</td>
<td>0.429</td>
<td>0.099</td>
<td>8.3</td>
</tr>
<tr>
<td></td>
<td>100</td>
<td>1020</td>
<td>3650</td>
<td>0.442</td>
<td>0.080</td>
<td>6.6</td>
</tr>
<tr>
<td></td>
<td>120</td>
<td>1003</td>
<td>3680</td>
<td>0.454</td>
<td>0.066</td>
<td>5.4</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Ethylene glycol</th>
<th>T (°C)</th>
<th>ρ (kg/m³)</th>
<th>cₚ (J/KgK)</th>
<th>k (W/mK)</th>
<th>µ x 10² (N.s/m²)</th>
<th>Pr</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0</td>
<td>1130</td>
<td>2294</td>
<td>0.242</td>
<td>6.501</td>
<td>615</td>
</tr>
<tr>
<td></td>
<td>20</td>
<td>1116</td>
<td>2382</td>
<td>0.249</td>
<td>2.140</td>
<td>204</td>
</tr>
<tr>
<td></td>
<td>40</td>
<td>1101</td>
<td>2474</td>
<td>0.256</td>
<td>0.957</td>
<td>93</td>
</tr>
<tr>
<td></td>
<td>60</td>
<td>1087</td>
<td>2562</td>
<td>0.26</td>
<td>0.516</td>
<td>51</td>
</tr>
<tr>
<td></td>
<td>80</td>
<td>1077</td>
<td>2650</td>
<td>0.261</td>
<td>0.321</td>
<td>32.4</td>
</tr>
<tr>
<td></td>
<td>100</td>
<td>1058</td>
<td>2742</td>
<td>0.263</td>
<td>0.215</td>
<td>22.4</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Glycerin</th>
<th>T (°C)</th>
<th>ρ (kg/m³)</th>
<th>cₚ (J/KgK)</th>
<th>k (W/mK)</th>
<th>µ x 10² (N.s/m²)</th>
<th>Pr</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0</td>
<td>1276</td>
<td>2261</td>
<td>0.282</td>
<td>1060.4</td>
<td>84700</td>
</tr>
<tr>
<td></td>
<td>10</td>
<td>1270</td>
<td>2319</td>
<td>0.284</td>
<td>381.0</td>
<td>31000</td>
</tr>
<tr>
<td></td>
<td>20</td>
<td>1264</td>
<td>2386</td>
<td>0.286</td>
<td>149.2</td>
<td>12500</td>
</tr>
<tr>
<td></td>
<td>30</td>
<td>1258</td>
<td>2445</td>
<td>0.286</td>
<td>62.9</td>
<td>5380</td>
</tr>
<tr>
<td></td>
<td>40</td>
<td>1252</td>
<td>2512</td>
<td>0.286</td>
<td>27.5</td>
<td>2450</td>
</tr>
<tr>
<td>T (°C)</td>
<td>ρ (kg/m³)</td>
<td>cₚ (J/KgK)</td>
<td>k (W/mK)</td>
<td>μ x 10⁶ (N.s/m²)</td>
<td>Pr</td>
<td></td>
</tr>
<tr>
<td>-------</td>
<td>-----------</td>
<td>-------------</td>
<td>----------</td>
<td>------------------</td>
<td>----</td>
<td></td>
</tr>
<tr>
<td>0</td>
<td>1002</td>
<td>4217</td>
<td>0.552</td>
<td>1792</td>
<td>13.6</td>
<td></td>
</tr>
<tr>
<td>20</td>
<td>1000</td>
<td>4181</td>
<td>0.597</td>
<td>1006</td>
<td>7.02</td>
<td></td>
</tr>
<tr>
<td>40</td>
<td>994</td>
<td>4178</td>
<td>0.628</td>
<td>654</td>
<td>4.34</td>
<td></td>
</tr>
<tr>
<td>60</td>
<td>985</td>
<td>4184</td>
<td>0.651</td>
<td>471</td>
<td>3.02</td>
<td></td>
</tr>
<tr>
<td>80</td>
<td>974</td>
<td>4196</td>
<td>0.668</td>
<td>355</td>
<td>2.22</td>
<td></td>
</tr>
<tr>
<td>100</td>
<td>960</td>
<td>4216</td>
<td>0.68</td>
<td>282</td>
<td>1.74</td>
<td></td>
</tr>
<tr>
<td>120</td>
<td>945</td>
<td>4250</td>
<td>0.685</td>
<td>233</td>
<td>1.45</td>
<td></td>
</tr>
<tr>
<td>140</td>
<td>928</td>
<td>4283</td>
<td>0.684</td>
<td>199</td>
<td>1.24</td>
<td></td>
</tr>
<tr>
<td>160</td>
<td>909</td>
<td>4342</td>
<td>0.67</td>
<td>173</td>
<td>1.10</td>
<td></td>
</tr>
<tr>
<td>180</td>
<td>889</td>
<td>4417</td>
<td>0.675</td>
<td>154</td>
<td>1.00</td>
<td></td>
</tr>
<tr>
<td>200</td>
<td>866</td>
<td>4505</td>
<td>0.665</td>
<td>139</td>
<td>0.94</td>
<td></td>
</tr>
<tr>
<td>220</td>
<td>842</td>
<td>4610</td>
<td>0.572</td>
<td>126</td>
<td>0.89</td>
<td></td>
</tr>
<tr>
<td>240</td>
<td>815</td>
<td>4756</td>
<td>0.635</td>
<td>117</td>
<td>0.87</td>
<td></td>
</tr>
<tr>
<td>260</td>
<td>785</td>
<td>4949</td>
<td>0.611</td>
<td>108</td>
<td>0.87</td>
<td></td>
</tr>
<tr>
<td>280</td>
<td>752</td>
<td>5208</td>
<td>0.58</td>
<td>102</td>
<td>0.91</td>
<td></td>
</tr>
<tr>
<td>300</td>
<td>714</td>
<td>5728</td>
<td>0.54</td>
<td>96</td>
<td>1.11</td>
<td></td>
</tr>
</tbody>
</table>

Table C.6 Pipe Dimensions

<table>
<thead>
<tr>
<th>Nominal Pipe Size</th>
<th>NPS (in.)</th>
<th>DN (mm)</th>
<th>O.D. (in.)</th>
<th>O.D. (mm)</th>
<th>Schedule</th>
<th>I.D. (in.)</th>
<th>I.D. (mm)</th>
<th>O.D./I.D</th>
</tr>
</thead>
<tbody>
<tr>
<td>1/8</td>
<td>6</td>
<td>0.405</td>
<td>10.29</td>
<td>10</td>
<td>0.307</td>
<td>7.80</td>
<td>1.32</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>40</td>
<td>0.269</td>
<td>6.83</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>80</td>
<td>0.215</td>
<td>5.46</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1/4</td>
<td>8</td>
<td>0.540</td>
<td>13.72</td>
<td>10</td>
<td>0.410</td>
<td>10.41</td>
<td>1.32</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>40</td>
<td>0.364</td>
<td>9.24</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>80</td>
<td>0.302</td>
<td>7.67</td>
<td></td>
<td></td>
</tr>
<tr>
<td>3/8</td>
<td>10</td>
<td>0.675</td>
<td>17.15</td>
<td>40</td>
<td>0.493</td>
<td>12.52</td>
<td>1.37</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>80</td>
<td>0.423</td>
<td>10.74</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1/2</td>
<td>15</td>
<td>0.840</td>
<td>21.34</td>
<td>40</td>
<td>0.622</td>
<td>15.80</td>
<td>1.35</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>80</td>
<td>0.546</td>
<td>13.87</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>160</td>
<td>0.464</td>
<td>11.79</td>
<td></td>
<td></td>
</tr>
<tr>
<td>3/4</td>
<td>20</td>
<td>1.050</td>
<td>26.67</td>
<td>40</td>
<td>0.824</td>
<td>20.93</td>
<td>1.27</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>80</td>
<td>0.742</td>
<td>18.85</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>25</td>
<td>1.315</td>
<td>33.40</td>
<td>40</td>
<td>1.049</td>
<td>26.64</td>
<td>1.25</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>80</td>
<td>0.957</td>
<td>24.31</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1 1/4</td>
<td>32</td>
<td>1.660</td>
<td>42.16</td>
<td>40</td>
<td>1.380</td>
<td>35.05</td>
<td>1.20</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>80</td>
<td>1.278</td>
<td>32.46</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Size</td>
<td>Thickness</td>
<td>Density</td>
<td>Strength</td>
<td>Tensile</td>
<td>Yield</td>
<td>Elongation</td>
<td></td>
<td></td>
</tr>
<tr>
<td>------</td>
<td>-----------</td>
<td>---------</td>
<td>----------</td>
<td>---------</td>
<td>-------</td>
<td>------------</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1 1/2</td>
<td>40</td>
<td>1.900</td>
<td>48.26</td>
<td>40</td>
<td>1.610</td>
<td>40.89</td>
<td>1.18</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>50</td>
<td>2.375</td>
<td>60.33</td>
<td>40</td>
<td>2.067</td>
<td>52.50</td>
<td>1.15</td>
<td></td>
</tr>
<tr>
<td>2 1/2</td>
<td>65</td>
<td>2.875</td>
<td>73.03</td>
<td>40</td>
<td>2.469</td>
<td>62.71</td>
<td>1.16</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>80</td>
<td>3.500</td>
<td>88.90</td>
<td>40</td>
<td>3.068</td>
<td>77.93</td>
<td>1.14</td>
<td></td>
</tr>
<tr>
<td>3 1/2</td>
<td>90</td>
<td>4.000</td>
<td>101.60</td>
<td>40</td>
<td>3.548</td>
<td>90.12</td>
<td>1.13</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>100</td>
<td>4.500</td>
<td>114.30</td>
<td>40</td>
<td>4.026</td>
<td>102.26</td>
<td>1.12</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>125</td>
<td>5.563</td>
<td>141.30</td>
<td>10 S</td>
<td>5.295</td>
<td>134.49</td>
<td>1.05</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>150</td>
<td>6.625</td>
<td>168.28</td>
<td>10 S</td>
<td>6.357</td>
<td>161.47</td>
<td>1.04</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>200</td>
<td>8.625</td>
<td>219.08</td>
<td>10 S</td>
<td>8.329</td>
<td>211.56</td>
<td>1.04</td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>250</td>
<td>10.750</td>
<td>273.05</td>
<td>10 S</td>
<td>10.420</td>
<td>264.67</td>
<td>1.03</td>
<td></td>
</tr>
<tr>
<td>12</td>
<td>300</td>
<td>12.750</td>
<td>323.85</td>
<td>10 S</td>
<td>12.390</td>
<td>314.71</td>
<td>1.03</td>
<td></td>
</tr>
<tr>
<td>14</td>
<td>350</td>
<td>14.000</td>
<td>355.60</td>
<td>10</td>
<td>13.500</td>
<td>342.90</td>
<td>1.04</td>
<td></td>
</tr>
<tr>
<td>16</td>
<td>400</td>
<td>16.000</td>
<td>406.40</td>
<td>10</td>
<td>15.500</td>
<td>393.70</td>
<td>1.03</td>
<td></td>
</tr>
<tr>
<td>18</td>
<td>450</td>
<td>18.000</td>
<td>457.20</td>
<td>10 S</td>
<td>17.624</td>
<td>447.65</td>
<td>1.02</td>
<td></td>
</tr>
</tbody>
</table>