**PROBLEM 11.2**

**KNOWN:** Type-302 stainless tube with prescribed inner and outer diameters used in a cross-flow heat exchanger. Prescribed fouling factors and internal water flow conditions.

**FIND:** (a) Overall coefficient based upon the outer surface, $U_o$, with air at $T_o = 15^\circ C$ and velocity $V_o = 20$ m/s in cross-flow; compare thermal resistances due to convection, tube wall conduction and fouling; (b) Overall coefficient, $U_o$, with water (rather than air) at $T_o = 15^\circ C$ and velocity $V_o = 1$ m/s in cross-flow; compare thermal resistances due to convection, tube wall conduction and fouling; (c) For the water-air conditions of part (a), compute and plot $U_o$ as a function of the air cross-flow velocity for $5 \leq V_o \leq 30$ m/s for water mean velocities of $u_{m,i} = 0.2, 0.5$ and $1.0$ m/s; and (d) For the water-water conditions of part (b), compute and plot $U_o$ as a function of the water mean velocity for $0.5 \leq u_{m,i} \leq 2.5$ m/s for air cross-flow velocities of $V_o = 1, 3$ and $8$ m/s.

**SCHEMATIC:**

**ASSUMPTIONS:** (1) Steady-state conditions, (2) Fully developed internal flow.

**PROPERTIES:** Table A.1, Stainless steel, AISI 302 (300 K): $k_w = 15.1$ W/m-K; Table A.6, Water ($\bar{T}_{m,i} = 348$ K): $\rho_i = 974.8$ kg/m$^3$, $\mu_i = 3.746 \times 10^{-4}$ N-s/m$^2$, $k_i = 0.668$ W/m-K, $Pr_i = 2.354$; Table A.4, Air (assume $\bar{T}_{f,o} = 315$K, 1 atm): $k_o = 0.02737$ W/m-K, $\nu_o = 17.35 \times 10^{-6}$ m$^2$/s, $Pr_o = 0.705$.

**ANALYSIS:** (a) For the water-air condition, the overall coefficient, Eq. 11.1, based upon the outer area can be expressed as the sum of the thermal resistances due to convection (cv), tube wall conduction (w) and fouling (f):

$$\frac{1}{U_o A_o} = R_{tot} = R_{cv,i} + R_{f,i} + R_w + R_{f,o} + R_{cv,o}$$

$$R_{cv,i} = \frac{1}{\bar{h}_i A_i} \quad R_{cv,o} = \frac{1}{\bar{h}_o A_o}$$

$$R_{f,i} = \frac{R^*_{f,i}}{A_i} \quad R_{f,o} = \frac{R^*_{f,o}}{A_o}$$

and from Eq. 3.28,

$$R_w = \ln \left( \frac{D_o}{D_i} \right) \left( \frac{2\pi L k_w}{D_i} \right)$$

The convection coefficients can be estimated from appropriate correlations. Continued...
PROBLEM 11.2 (Cont.)

Estimating $\bar{h}_i$: For internal flow, characterize the flow evaluating thermophysical properties at $T_{m,i}$ with

$$Re_D,i = \frac{u_{m,i}D_i}{v_i} = \frac{0.5 \text{ m/s} \times 0.022 \text{ m}}{3.746 \times 10^{-4} \text{ N/s/m}^2 / 974.8 \text{ kg/m}^3} = 28,625$$

For the turbulent flow, use the Dittus-Boelter correlation, Eq. 8.60,

$$Nu_D,i = 0.023 Re_D,i^{0.8} Pr_i^{0.4}$$

$$Nu_D,i = 0.023 (28,625)^{0.8} (2.354)^{0.4} = 119.1$$

$$\bar{h}_i = Nu_D,i k_i/D_i = 119.1 \times 0.668 \text{ W/m}^2 \cdot \text{K} / 0.022 \text{ m} = 3616 \text{ W/m}^2 \cdot \text{K}$$

Estimating $\bar{h}_o$: For external flow, characterize the flow with

$$Re_D,o = \frac{V_o D_o}{v_o} = \frac{20 \text{ m/s} \times 0.027 \text{ m}}{17.35 \times 10^{-6} \text{ m}^2 / \text{s}} = 31,124$$

evaluating thermophysical properties at $T_{f,o} = (T_{s,o} + T_o)/2$ when the surface temperature is determined from the thermal circuit analysis result,

$$\left( T_{m,i} - T_o \right) / R_{tot} = \left( T_{s,o} - T_o \right) / R_{cv,o}$$

Assume $T_{f,o} = 315 \text{ K}$, and check later. Using the Churchill-Bernstein correlation, Eq. 7.57, find

$$\overline{Nu}_{D,o} = 0.3 + \frac{0.62 Re_{D,o}^{1/2} Pr_{o}^{1/3}}{\left[ 1 + \left( 0.4 / Pr_o \right)^{2/3} \right]^{1/4}} \left[ 1 + \left( \frac{Re_{D,o}}{282,000} \right)^{5/8} \right]^{4/5}$$

$$\overline{Nu}_{D,o} = 0.3 + \frac{0.62 (31,124)^{1/2} (0.705)^{1/3}}{\left[ 1 + \left( 0.4 / 0.705 \right)^{2/3} \right]^{1/4}} \left[ 1 + \left( \frac{31,124}{282,000} \right)^{5/8} \right]^{4/5}$$

$$\overline{Nu}_{D,o} = 102.6$$

$$\bar{h}_o = Nu_{D,o} k_o / D_o = 102.6 \times 0.02737 \text{ W/m} \cdot \text{K} / 0.027 \text{ m} = 104.0 \text{ W/m} \cdot \text{K}$$

Using the above values for $\bar{h}_i$, and $\bar{h}_o$, and other prescribed values, the thermal resistances and overall coefficient can be evaluated and are tabulated below.

<table>
<thead>
<tr>
<th>$R_{cv,i}$</th>
<th>$R_{i,i}$</th>
<th>$R_{w}$</th>
<th>$R_{f,i}$</th>
<th>$R_{cv,o}$</th>
<th>$R_{f,o}$</th>
<th>$U_o$</th>
<th>$R_{tot}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>(K/W)</td>
<td>(K/W)</td>
<td>(K/W)</td>
<td>(K/W)</td>
<td>(K/W)</td>
<td>(K/W)</td>
<td>(W/m$^2$·K)</td>
<td>(K/W)</td>
</tr>
<tr>
<td>0.00436</td>
<td>0.00578</td>
<td>0.00216</td>
<td>0.00236</td>
<td>0.1134</td>
<td>92.1</td>
<td>0.128</td>
<td></td>
</tr>
</tbody>
</table>

The major thermal resistance is due to outside (air) convection, accounting for 89% of the total resistance. The other thermal resistances are of similar magnitude, nearly 50 times smaller than $R_{cv,o}$.

(b) For the water-water condition, the method of analysis follows that of part (a). For the internal flow, the estimated convection coefficient is the same as part (a). For an assumed outer film coefficient, $T_{f,o} = 292 \text{ K}$, the convection correlation for the outer water flow condition $V_o = 1 \text{ m/s}$ and $T_o = 15^\circ \text{C}$, find
PROBLEM 11.2 (Cont.)

\[ \text{Re}_{D,o} = 26,260 \quad \text{Nu}_{D,o} = 220.6 \quad \overline{h}_o = 4914 \text{ W/m}^2 \cdot \text{K} \]

The thermal resistances and overall coefficient are tabulated below.

<table>
<thead>
<tr>
<th>( R_{cv,i} ) (K/W)</th>
<th>( R_{i,i} ) (K/W)</th>
<th>( R_{w} ) (K/W)</th>
<th>( R_{z,0} ) (K/W)</th>
<th>( R_{cv,o} ) (K/W)</th>
<th>( R_{tot} ) (K/W)</th>
<th>( U_o ) (W/m(^2)\cdot K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.00436</td>
<td>0.00579</td>
<td>0.00216</td>
<td>0.00236</td>
<td>0.00240</td>
<td>0.0171</td>
<td>691</td>
</tr>
</tbody>
</table>

Note that the thermal resistances are of similar magnitude. In contrast with the results for the water-air condition of part (a), the thermal resistance of the outside convection process, \( R_{cv,o} \), is nearly 50 times smaller. The overall coefficient for the water-water condition is 7.5 times greater than that for the water-air condition.

(c) For the water-air condition, using the IHT workspace with the analysis of part (a), \( U_o \) was calculated as a function of the air cross-flow velocity for selected mean water velocities.

![Graph](image)

Water (i) - air (o) condition

The effect of increasing the cross-flow air velocity is to increase \( U_o \) since the \( R_{cv,o} \) is the dominant thermal resistance for the system. While increasing the water mean velocity will increase \( \overline{h}_i \), because \( R_{cv,i} \ll R_{cv,o} \), this increase has only a small effect on \( U_o \).

(d) For the water-water condition, using the IHT workplace with the analysis of part (b), \( U_o \) was calculated as a function of the mean water velocity for selected air cross-flow velocities.

![Graph](image)

Water (i) - water (o) condition

Because the thermal resistances for the convection processes, \( R_{cv,i} \) and \( R_{cv,o} \), are of similar magnitude according to the results of part (b), we expect to see \( U_o \) significantly increase with increasing water mean velocity and air cross-flow velocity.
PROBLEM 11.10

KNOWN: Heat exchanger with two shell passes and eight tube passes having an area \(925\, \text{m}^2\); 45,500 kg/h water is heated from 80°C to 150°C; hot exhaust gases enter at 350°C and exit at 175°C.

FIND: Overall heat transfer coefficient.

SCHEMATIC:

ASSUMPTIONS: (1) Negligible losses to surroundings, (2) Negligible kinetic and potential energy changes, (3) Constant properties, (4) Exhaust gas properties are approximated as those of atmospheric air.

PROPERTIES: Table A-6, Water \(T_c = (80+150)\, ^\circ\text{C}/2 = 388\, \text{K}\): \(c = c_{p,f} = 4236\, \text{J/kg}\cdot\text{K}\).

ANALYSIS: The overall heat transfer coefficient follows from Eqs. 11.9 and 11.18 written in the form

\[
U = q / AF\Delta T_{m,CF}
\]

where \(F\) is the correction factor for the HXer configuration, Fig. 11.11, and \(\Delta T_{m,CF}\) is the log mean temperature difference (CF), Eqs. 11.15 and 11.16. From Fig. 11.11, find

\[
R = \frac{T_{h,i} - T_{h,o}}{T_{c,o} - T_{c,i}} = \frac{(350-175)\, ^\circ\text{C}}{(150-80)\, ^\circ\text{C}} = 2.5 \quad P = \frac{T_{c,o} - T_{c,i}}{T_{h,i} - T_{c,i}} = \frac{(150-80)\, ^\circ\text{C}}{(350-80)\, ^\circ\text{C}} = 0.26
\]

find \(F \approx 0.97\). The log-mean temperature difference, Eqs. 11.15 and 11.17, is

\[
\Delta T_{m,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{(350-150)\, ^\circ\text{C} - (175-80)\, ^\circ\text{C}}{\ln[(350-150)/(175-80)]} = 141.1\, ^\circ\text{C}.
\]

From an overall energy balance on the cold fluid (water), the heat rate is

\[
q = \dot{m}_c \, c \left( T_{c,o} - T_{c,i} \right)
\]

\[
q = 45,500\, \text{kg/h} \times 1\, \text{h/3600s} \times 4236\, \text{J/kg}\cdot\text{K} \times (150-80)\, ^\circ\text{C} = 3.748 \times 10^6\, \text{W}.
\]

Substituting values with \(A = 925\, \text{m}^2\), find

\[
U = 3.748 \times 10^6\, \text{W} / 925\, \text{m}^2 \times 0.97 \times 141.1\, \text{K} = 29.6\, \text{W} / \text{m}^2\cdot\text{K}.
\]

COMMENTS: Compare the above result with representative values for air-water exchangers, as given in Table 11.2. Note that in this exchanger, two shells with eight tube passes, the correction factor effect is very small, since \(F \approx 0.97\).
**PROBLEM 11.35**

**KNOWN:** Steam at 0.14 bar condensing in a shell and tube HXer (one shell, two tube passes consisting of 130 brass tubes off length 2 m, \(D_i = 13.4 \text{ mm, } D_o = 15.9 \text{ mm}\)). Cooling water enters at 20°C with a mean velocity 1.25 m/s. Heat transfer convection coefficient for condensation on outer tube surface is \(h_o = 13,500 \text{ W/m}^2\cdot\text{K} \).

**FIND:** (a) Overall heat transfer coefficient, \(U\), for the HXer, outlet temperature of cooling water, \(T_{c,o}\), and condensation rate of the steam \(\dot{m}_h\); and (b) Compute and plot \(T_{c,o}\) and \(\dot{m}_h\) as a function of the water flow rate \(10 \leq \dot{m}_c \leq 30 \text{ kg/s}\) with all other conditions remaining the same, but accounting for changes in \(U\).

**SCHEMATIC:**

**ASSUMPTIONS:** (1) Negligible heat loss to surroundings, (2) Negligible kinetic and potential energy changes, (3) Constant properties, (4) Fully developed water flow in tubes.

**PROPERTIES:** *Table A-6,* Steam (0.14 bar): \(T_{sat} = T_h = 327 \text{ K, } h_{fg} = 2373 \text{ kJ/kg, } c_p = 1898 \text{ J/kg} \cdot \text{K};
*Table A-6,* Water (Assume \(T_{c,o} \approx 44\degree \text{C or } \bar{T}_c = 305 \text{ K})): \(v_f = 1.005 \times 10^{-3} \text{ m}^3/\text{kg} ,
\(c_p = 4178 \text{ J/kg} \cdot \text{K, } \mu_f = 769 \times 10^{-6} \text{ N} \cdot \text{s/m}^2 ,
\(k_f = 0.620 \text{ W/m} \cdot \text{K, } Pr_f = 5.2;
*Table A-1,* Brass - 70/30 (Evaluate at \(T = (T_h + \bar{T}_c)/2 = 316 \text{ K})): \(k = 114 \text{ W/m} \cdot \text{K}.

**ANALYSIS:** (a) The overall heat transfer coefficient based upon the outside tube area follows from Eq. 11.5,
\[
U_o = \left[\frac{1}{h_o} + \frac{\rho_o D_i}{k} \left(\frac{T_h - T_c}{T_h}ight) \left(\frac{1}{h_i}\right)\right]^{-1}.
\] (1)

The value for \(h_i\) can be estimated from an appropriate internal flow correlation. First determine the nature of the flow within the tubes. From Eq. 8.1,

\[
Re_{D_i} = \frac{\rho u_m D_i}{\mu} = \frac{\left(1.005 \times 10^{-3} \text{ m}^3/\text{kg}\right)^{-1} \times 1.25 \text{ m/s} \times 13.4 \times 10^{-3} \text{ m}}{769 \times 10^{-6} \text{ N} \cdot \text{s/m}^2} = 21,673.
\]

The water flow is turbulent and fully developed (\(L/D_i = 2 \text{ m} / 13.4 \times 10^{-3} \text{ m} = 150 > 10\)). The Dittus-Boelter correlation with \(n = 0.4\) is appropriate,

\[
Nu_D = h_i D_i/k_f = 0.023 Re_{D_i}^{0.8} Pr_f^{0.4} = 0.023 \times (21,673)^{0.8} (5.2)^{0.4} = 130.9
\]

Continued...
PROBLEM 11.35 (Cont.)

\[ h_i = \frac{k_f \cdot \text{Nu}_D}{D_i} = \frac{0.620 \text{W/m} \cdot \text{K}}{13.4 \times 10^{-3} \text{m}} \times 130.9 = 6057 \text{W/m}^2 \cdot \text{K}. \]

Substituting numerical values into Eq. (1), the overall heat transfer coefficient is

\[
U_o = \left[ \frac{1}{13,500 \text{W/m}^2 \cdot \text{K}} + \frac{(15.9 \times 10^{-3} \text{m})^2}{115 \text{W/m} \cdot \text{K}} \frac{15.9}{13.4} + \frac{15.9}{13.4} \times \frac{1}{6057 \text{W/m}^2 \cdot \text{K}} \right]^{-1}
\]

\[
U_o = \left[ 7.407 \times 10^{-5} + 1.183 \times 10^{-5} + 19.590 \times 10^{-5} \right]^{-1} \text{W/m}^2 \cdot \text{K} = 3549 \text{W/m}^2 \cdot \text{K}.
\]

To find the outlet temperature of the water, we’ll employ the \( \varepsilon - \text{NTU} \) method. From an energy balance on the cold fluid,

\[
T_{c,o} = T_{c,i} + \frac{q}{C_c}
\]

where the heat rate can be expressed as

\[
q = \varepsilon q_{\text{max}} \quad q_{\text{max}} = C_{\text{min}} \left( T_{h,i} - T_{h,o} \right).
\]

The minimum capacity rate is that of the cold water since \( C_h \to \infty \). Evaluating, find

\[
C_{\text{min}} = C_c = \left( \dot{m}_c \cdot c_p \right)_c = 22.8 \text{kg/s} \times 4178 \text{J/kg} \cdot \text{K} = 95,270 \text{W/K}.
\]

where

\[
\dot{m}_c = (\rho \cdot A_m) N = 995.0 \text{kg/m}^3 \times \pi/4 (0.0134 \text{m})^2 \times 1.25 \text{m/s} \times 130 = 22.8 \text{kg/s}
\]

To determine \( \varepsilon \), use Fig. 11.16 (one shell and any multiple of tube passes) with

\[
\text{NTU} = \frac{U_o \cdot A_o}{C_{\text{min}}} = \frac{3549 \text{W/m}^2 \cdot \text{K} \left( \pi 0.0159 \text{m} \times 2 \text{m} \times 130 \times 2 \right)}{95,270 \text{W/K}} = 0.968
\]

where 130 and 2 represent the number of tubes and passes, respectively, to find \( \varepsilon \approx 0.62 \). Combining Eqs. (4) and (5) into Eq. (3), find

\[
T_{c,o} = T_{c,i} + \varepsilon C_{\text{min}} \left( T_{h,i} - T_{c,i} \right)/C_c = 20^\circ \text{C} + 0.62 (327 - 293) \text{K} = 41.1^\circ \text{C}.
\]

The condensation rate of the steam is given by

\[
\dot{m}_h = q/h_{fg}
\]

where the heat rate can be determined from Eq. (3) with \( T_{c,o} \),

\[
\dot{m}_h = C_c \left( T_{c,o} - T_{c,i} \right)/h_{fg} = 95,270 \text{W/K} \left( 41.1 - 20.0 \right) \text{K}/2373 \times 10^3 \text{J/kg} \cdot \text{K} = 0.85 \text{kg/s}.
\]

(b) Using the IHT Heat Exchanger Tool, All Exchangers, \( C_r = 0 \), and the Properties Tool for Water, a model was developed and the cold outlet temperature and condensation rate were computed and plotted.

Continued...
With increasing cold flow rate, the cold outlet temperature decreases as expected. The condensation rate increases with increasing cold flow rate. Note that $T_{c,o}$ and $\dot{m}_h$ are nearly linear with the cold flow rate.

**COMMENTS:** For part (a) analysis, note that the assumption $T_{c,o} = 44^\circ C$ used for evaluation of the cold fluid properties was reasonable. Using the IHT model of part (b), we found $T_{c,o} = 40.2^\circ C$ and $\dot{m}_h = 0.812$ kg/s.
PROBLEM 11.50

KNOWN: Feed water heater (single shell, two tube passes) with inlet temperature 20°C supplies 10,000 kg/h of water at 65°C by condensing steam at 1.3 bar. Overall heat transfer coefficient is 2000 W/m²·K.

FIND: (a) Required area using LMTD and NTU approaches, (b) Steam condensation rate.

SCHEMATIC:

ASSUMPTIONS: (1) Negligible heat loss to surroundings, (2) Negligible kinetic and potential energy changes, (3) Constant properties.

PROPERTIES: Table A-6, Steam (1.3 bar, saturated): \( T_h = 380.3 \) K, \( h_{fg} = 2238 \times 10^3 \) J/kg·K; Table A-6, Water (\( T_c = 316 \) K): \( c_p = 4179 \) J/kg·K.

ANALYSIS: (a) Using the LMTD approach, from Eqs. 11.14 and 11.18,

\[
A = \frac{q}{U F \Delta T_{\ell m, CF}} \quad \Delta T_{\ell m, CF} = \frac{[\Delta T_1 - \Delta T_2]}{\ln(\Delta T_1/\Delta T_2)}
\]

Since \( T_h \) is uniform throughout the HXer, \( F = 1 \). From an energy balance on the cold fluid,

\[
q = \dot{m}_c c_p (T_{c,o} - T_{c,i}) = \frac{10,000 \text{ kg}}{3600 \text{ s}} \times \frac{4179 \text{ J}}{\text{kg} \cdot \text{K}} (338 - 293) \text{ K} = 5.224 \times 10^5 \text{ W}.
\]

Substituting numerical values into Eq. (1) find that

\[
A = 5.224 \times 10^5 \text{ W} / 2000 \text{ W/m}^2 \cdot \text{K} = 4.21 \text{ m}^2.
\]

Using the NTU approach, recognize that \( C_{\text{min}} = C_c \) and \( C_{\text{max}} = C_h \to \infty \) so that \( C_{\text{min}}/C_{\text{max}} = 0 \). The effectiveness, defined by Eq. 11.20, is

\[
\varepsilon \equiv \frac{q}{q_{\text{max}}} = \frac{C_c (T_{c,o} - T_{c,i})}{C_{\text{min}} (T_{h,i} - T_{c,i})} = \frac{(338 - 293) \text{ K}}{(380.3 - 293) \text{ K}} = 0.515.
\]

From Fig. 11.16, with \( \varepsilon = 0.52 \) and \( C_{\text{min}}/C_{\text{max}} = 0 \), find NTU = 0.70. Hence,

\[
A = C_{\text{min}} \text{ NTU} / U = \frac{(10,000 \text{ kg/s}) (3600 \text{ J/kg} \cdot \text{K})}{2000 \text{ W/m}^2 \cdot \text{K}} \times 0.70 = 4.1 \text{ m}^2.
\]

(b) The condensation rate of steam is

\[
\dot{m}_h = \frac{q}{h_{fg}} = 5.224 \times 10^5 \text{ W} / (2238 \times 10^3 \text{ J/kg}) = 0.233 \text{ kg/s} = 840 \text{ kg/h}.
\]

COMMENTS: Note both methods of solution given the same result. Eq. 11.31 could have been used to obtain a more precise NTU value.
PROBLEM 11.59

KNOWN: Flow rates and inlet temperatures of exhaust gases and combustion air used in a cross-flow (one fluid mixed) heat exchanger. Overall heat transfer coefficient. Desired air outlet temperature.

FIND: Required heat exchanger surface area.

SCHEMATIC:

ASSUMPTIONS: (1) Steady-state conditions, (2) Negligible heat loss to surroundings, (3) Negligible kinetic and potential energy changes, (4) Constant properties, (5) Gas properties are those of air.

PROPERTIES: Table A-4, Air (\(T_m = 700\) K, 1 atm): \(c_p = 1075\) J/kgK.

ANALYSIS: From Eqs. 11.6 and 11.7,

\[
T_{h,o} = T_{h,i} - \frac{m_c \cdot c_p \cdot c}{m_h \cdot c_p \cdot h} (T_c,o - T_c,i) = 1100K - \frac{10}{15} (850 - 300) K = 733K.
\]

From Eqs. 11.15, 11.17 and 11.18,

\[
\Delta T_{\text{eff}} = F \frac{(T_{h,i} - T_{c,o}) - (T_{h,o} - T_{c,i})}{\ln \left( \frac{T_{h,i} - T_{c,o}}{T_{h,o} - T_{c,i}} \right)} = 5.92 \times 10^6 W
\]

it follows from Eq. 11.14 that

\[
A = \frac{5.92 \times 10^6 W}{100 W/m^2 \cdot K \times 0.73 (333K)} = 243 m^2.
\]

COMMENTS: Using the effectiveness-NTU method, from Eq. 11.22

\[
\varepsilon = \frac{T_{c,o} - T_{c,i}}{T_{h,i} - T_{c,i}} = \frac{(850 - 300) K}{(1100 - 300) K} = 0.688.
\]

Hence, with \(C_{\text{mixed}}/C_{\text{unmixed}} = C_c/C_h = 0.67\), Fig. 11.19 gives NTU \(\approx 2.3\). From Eq. 11.25,

\[
A = NTU \cdot \frac{C_{\text{min}}}{U} = 2.3 \times \frac{10 \text{ kg/s} \times 1075 \text{ J/kg} \cdot \text{K}}{100 \text{ W/m}^2 \cdot \text{K}} = 247 m^2.
\]