The Effectiveness – NTU Method

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General Considerations

- Computational Features/Limitations of the LMTD Method:
  - The LMTD method may be applied to design problems for which the fluid flow rates and inlet temperatures, as well as a desired outlet temperature, are prescribed.
  - For a specified HX type, the required size (surface area), as well as the other outlet temperature, are readily determined.
  - If the LMTD method is used in performance calculations for which both outlet temperatures must be determined from knowledge of the inlet temperatures, the solution procedure is iterative.
  - For both design and performance calculations, the effectiveness-NTU method may be used without iteration.

LMTD Method

$$Q = U\,A_s\,\Delta T_{lm}$$

The procedure to be followed by the selection process is:

1. Select the type of heat exchanger suitable for the application.
2. Determine any unknown inlet or outlet temperature and the heat transfer rate using an energy balance.
3. Calculate the log mean temperature difference T_{lm} and the correction factor F, if necessary.
4. Obtain (select or calculate) the value of the overall heat transfer coefficient U.
5. Calculate the heat transfer surface area A_s .

The Effectiveness – NTU Method

- In an attempt to eliminate the iterations from the solution of such problems, Kays and London came up with a method in 1955 called the **effectiveness–NTU method, which greatly simplified heat exchanger analysis.**

- This method is based on a dimensionless parameter called the **heat transfer effectiveness, defined as**

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{\text{Actual heat transfer rate}}{\text{Maximum possible heat transfer rate}}$$

The actual heat transfer rate in a heat exchanger can be determined from an energy balance on the hot or cold fluids and can be expressed as

$$\dot{Q} = C_c(T_{c,\,\text{out}} - T_{c,\,\text{in}}) = C_h(T_{h,\,\text{in}} - T_{h,\,\text{out}})$$

where $C_c = \dot{m}_c C_{pc}$ and $C_h = \dot{m}_c C_{ph}$ are the heat capacity rates of the cold and the hot fluids, respectively.

$$\Delta T_{\max} = T_{h,\,\text{in}} - T_{c,\,\text{in}}$$

$$\dot{Q}_{\max} = C_{\min}(T_{h,\,\text{in}} - T_{c,\,\text{in}})$$

$$\dot{Q} = \varepsilon \dot{Q}_{\max} = \varepsilon C_{\min}(T_{h,\,\text{in}} - T_{c,\,\text{in}})$$

Definitions

- Heat exchanger effectiveness : ε

$$\varepsilon = \frac{q}{q_{\max}}$$

$$0 \le \varepsilon \le 1$$

- Maximum possible heat rate :

$$q_{\max} = C_{\min}\left(T_{h,i} - T_{c,i}\right)$$

$$C_{\min} = \begin{cases} C_h \ \text{ if } C_h < C_c \\ \text{or} \\ C_c \ \text{ if } C_c < C_h \end{cases}$$

➢ Will the fluid characterized by C_{min} or C_{max} experience the largest possible temperature change in transit through the HX?

➢ Why is C_{min} and not C_{max} used in the definition of q_{max}?

$$\ln \frac{T_{h,\,\mathrm{out}} - T_{c,\,\mathrm{out}}}{T_{h,\,\mathrm{in}} - T_{c,\,\mathrm{in}}} = -UA_s\left(\frac{1}{\dot{m}_h C_{ph}} + \frac{1}{\dot{m}_c C_{pc}}\right)$$

For a parallel-flow heat exchanger can be rearranged as

$$\ln \frac{T_{h,\,\mathrm{out}} - T_{c,\,\mathrm{out}}}{T_{h,\,\mathrm{in}} - T_{c,\,\mathrm{in}}} = -\frac{UA_s}{C_c}\left(1 + \frac{C_c}{C_h}\right)$$

$$T_{h,\,\mathrm{out}} = T_{h,\,\mathrm{in}} - \frac{C_c}{C_h}\left(T_{c,\,\mathrm{out}} - T_{c,\,\mathrm{in}}\right)$$

$$\ln \frac{T_{h,\,\text{in}} - T_{c,\,\text{in}} + T_{c,\,\text{in}} - T_{c,\,\text{out}} - \dfrac{C_c}{C_h}(T_{c,\,\text{out}} - T_{c,\,\text{in}})}{T_{h,\,\text{in}} - T_{c,\,\text{in}}} = -\frac{UA_s}{C_c}\left(1 + \frac{C_c}{C_h}\right)$$

$$\ln \left[1 - \left(1 + \frac{C_c}{C_h}\right)\frac{T_{c,\,\text{out}} - T_{c,\,\text{in}}}{T_{h,\,\text{in}} - T_{c,\,\text{in}}}\right] = -\frac{UA_s}{C_c}\left(1 + \frac{C_c}{C_h}\right)$$

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\text{max}}} = \frac{C_c(T_{c,\,\text{out}} - T_{c,\,\text{in}})}{C_{\text{min}}(T_{h,\,\text{in}} - T_{c,\,\text{in}})} \quad \longrightarrow \quad \frac{T_{c,\,\text{out}} - T_{c,\,\text{in}}}{T_{h,\,\text{in}} - T_{c,\,\text{in}}} = \varepsilon \frac{C_{\text{min}}}{C_c}$$

$$\varepsilon_{\text{parallel flow}} = \frac{1 - \exp\left[-\dfrac{UA_s}{C_c}\left(1 + \dfrac{C_c}{C_h}\right)\right]}{\left(1 + \dfrac{C_c}{C_h}\right)\dfrac{C_{\text{min}}}{C_c}}$$

Effectiveness relations of the heat exchangers typically involve the dimensionless group UA_s / C_{min}.
This quantity is called *the* ***number of transfer units*** **NTU** and is expressed as

$$\mathrm{NTU} = \frac{UA_s}{C_{\min}} = \frac{UA_s}{(\dot{m}C_p)_{\min}}$$

In heat exchanger analysis, it is also convenient to define another dimensionless quantity called the **capacity ratio *c as***

$$c = \frac{C_{\min}}{C_{\max}}$$

$$\varepsilon = \text{function}\,(UA_s/C_{\min}, C_{\min}/C_{\max}) = \text{function}\,(\mathrm{NTU}, c)$$

TABLE 13–4

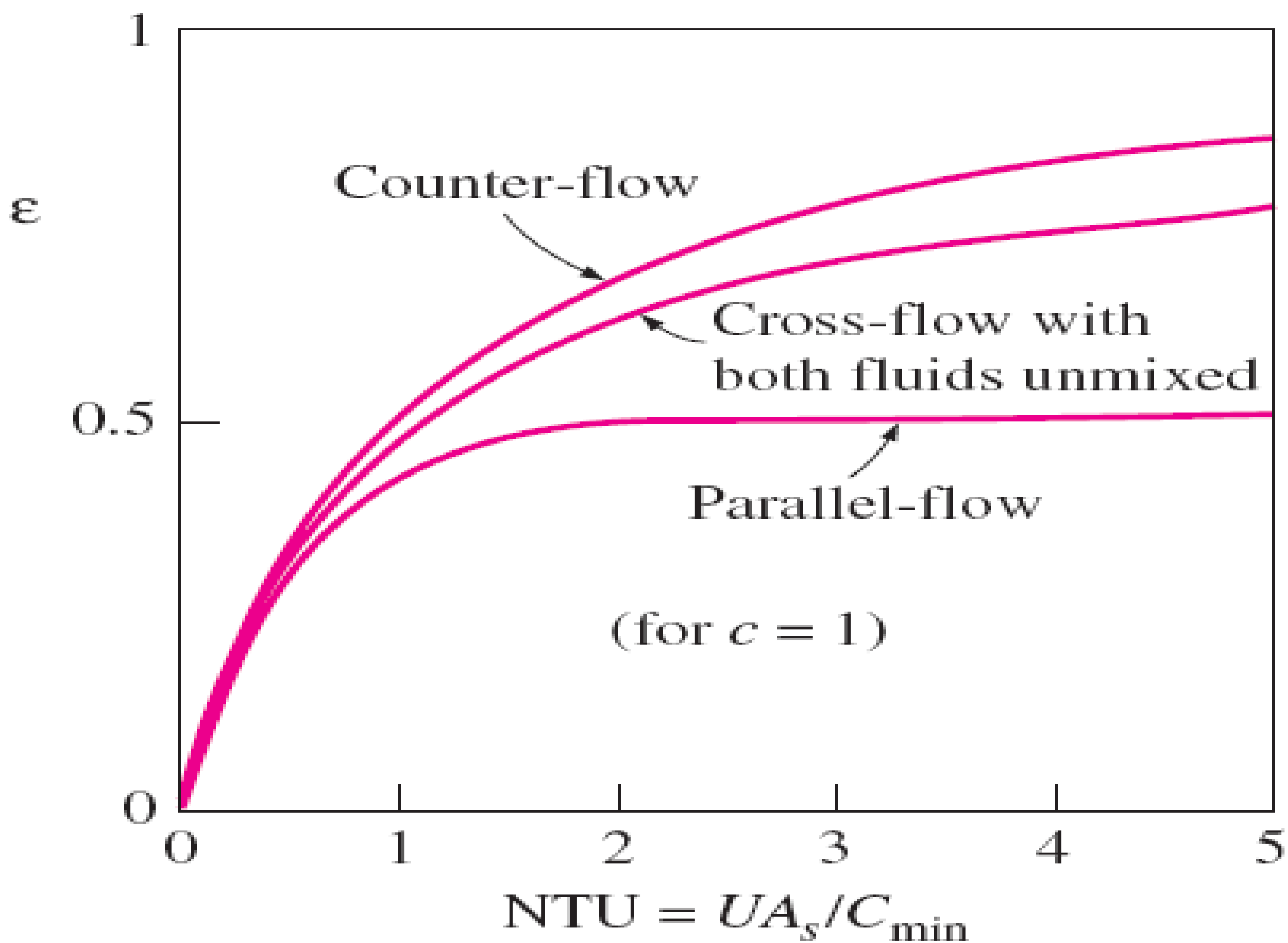
Effectiveness relations for heat exchangers: $NTU = UA_s/C_{min}$ and $c = C_{min}/C_{max} = (\dot{m}C_p)_{min}/(\dot{m}C_p)_{max}$ (Kays and London, Ref. 5.)

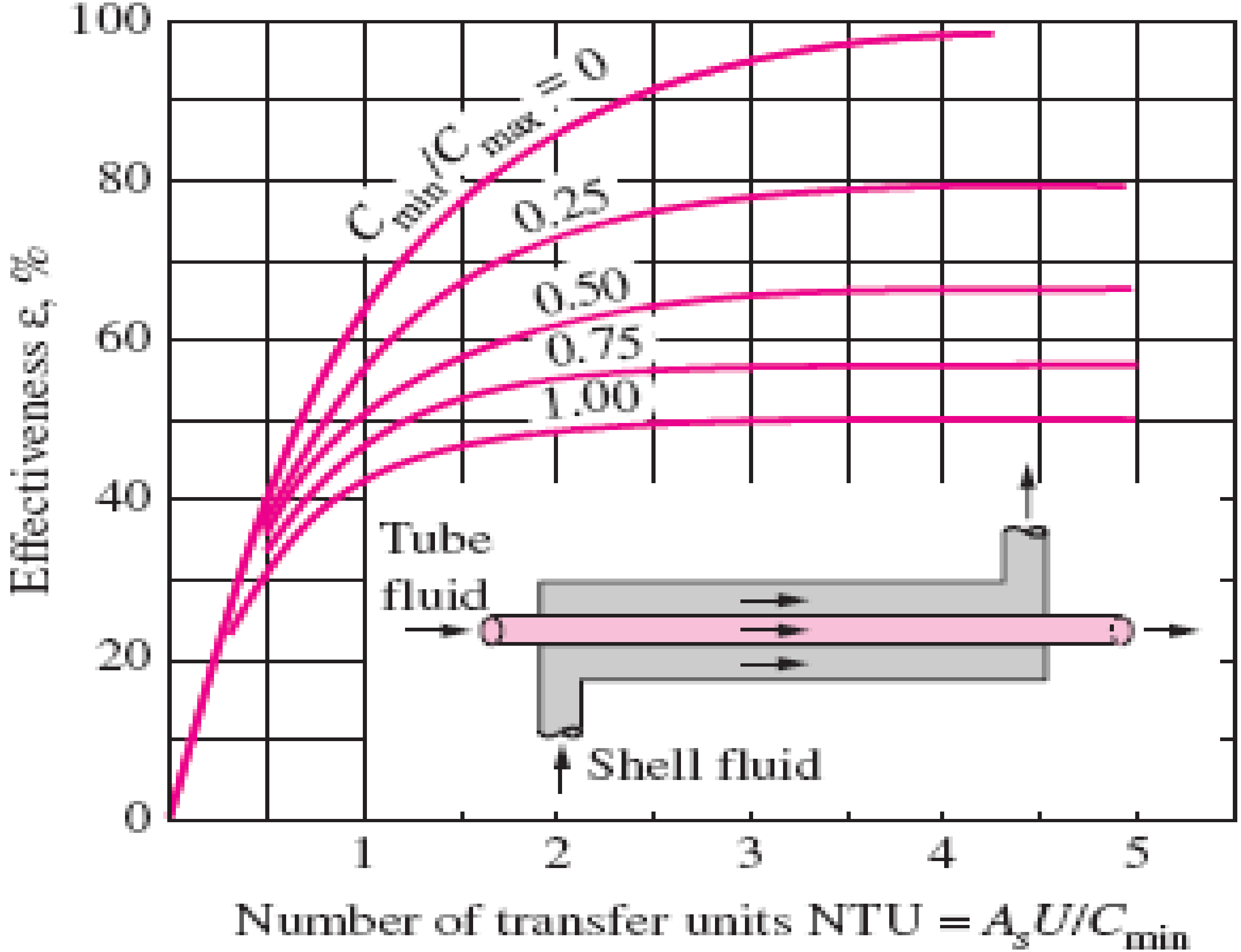
Heat exchanger type	Effectiveness relation
1 *Double pipe:*	
Parallel-flow	$\varepsilon = \dfrac{1 - \exp[-NTU(1 + c)]}{1 + c}$
Counter-flow	$\varepsilon = \dfrac{1 - \exp[-NTU(1 - c)]}{1 - c\exp[-NTU(1 - c)]}$
2 *Shell and tube:*	
One-shell pass 2, 4, . . . tube passes	$\varepsilon = 2\left\{1 + c + \sqrt{1 + c^2}\dfrac{1 + \exp[-NTU\sqrt{1 + c^2}]}{1 - \exp[-NTU\sqrt{1 + c^2}]}\right\}^{-1}$

TABLE 13–4

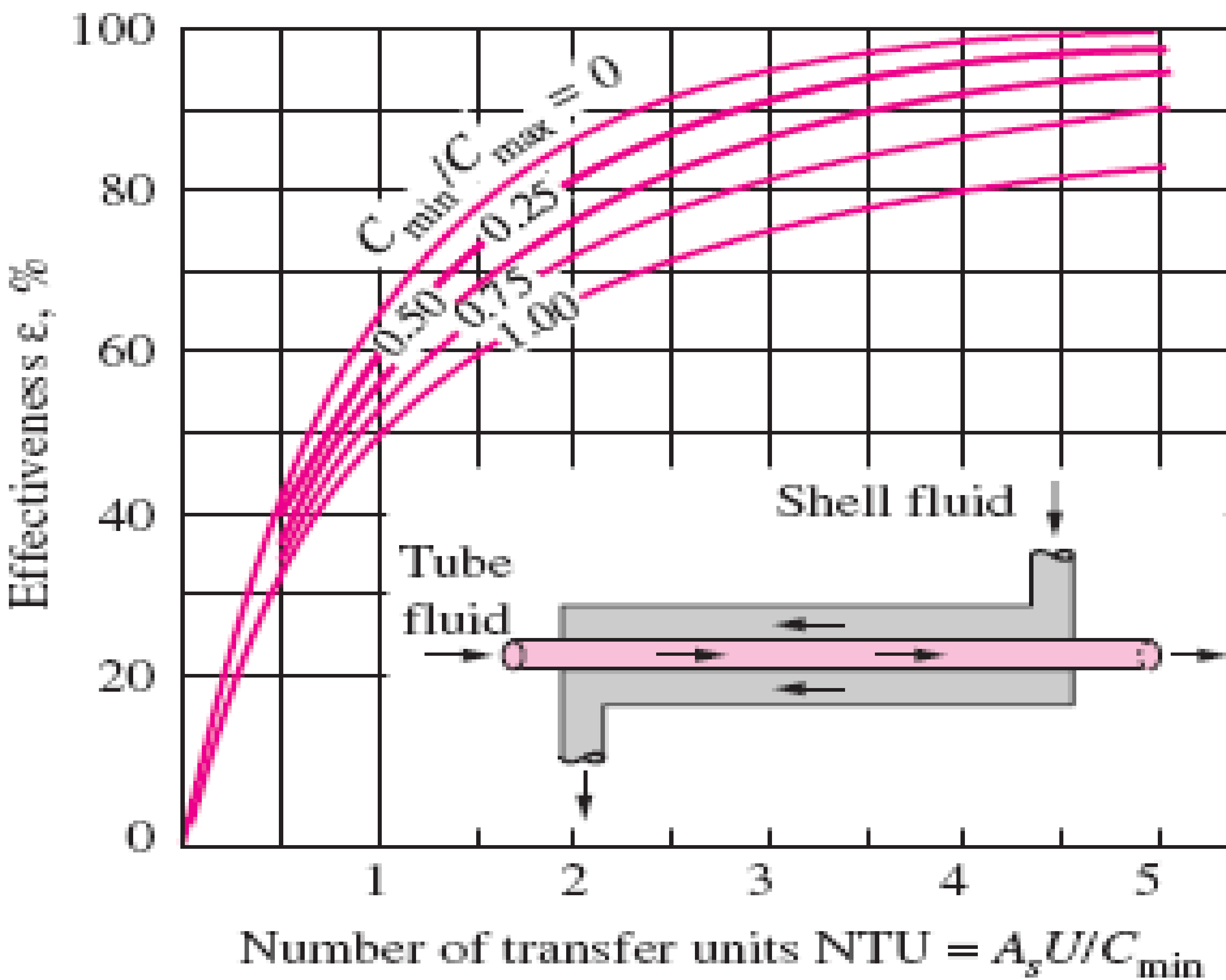
Effectiveness relations for heat exchangers: $\text{NTU} = UA_s/C_{\min}$ and $c = C_{\min}/C_{\max} = (\dot{m}C_p)_{\min}/(\dot{m}C_p)_{\max}$ (Kays and London, Ref. 5.)

Heat exchanger type	Effectiveness relation
3 *Cross-flow (single-pass)*	
Both fluids unmixed	$\varepsilon = 1 - \exp\left\{\frac{\text{NTU}^{0.22}}{c}\left[\exp\left(-c\ \text{NTU}^{0.78}\right) - 1\right]\right\}$
$C_{\max}$ mixed, $C_{\min}$ unmixed	$\varepsilon = \frac{1}{c}(1 - \exp\{1 - c[1 - \exp(-\text{NTU})]\})$
$C_{\min}$ mixed, $C_{\max}$ unmixed	$\varepsilon = 1 - \exp\left\{-\frac{1}{c}[1 - \exp(-c\ \text{NTU})]\right\}$
4 *All heat exchangers with $c = 0$*	$\varepsilon = 1 - \exp(-\text{NTU})$

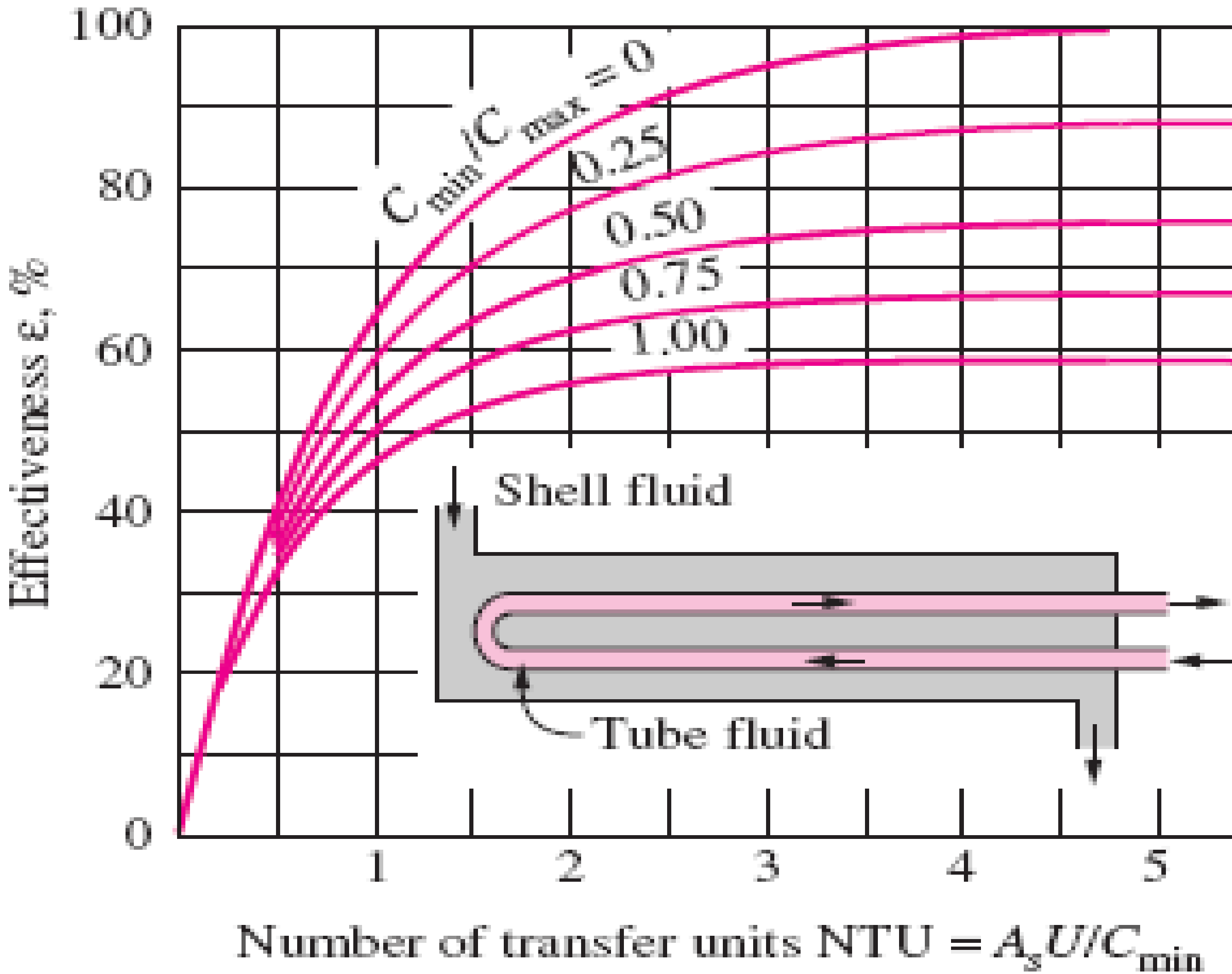

1
ε
0.5
0
Counter-flow
Cross-flow with
both fluids unmixed
Parallel-flow
(for $c = 1$)
0
1
2
3
4
5
$\mathrm{NTU} = UA_s/C_{\min}$



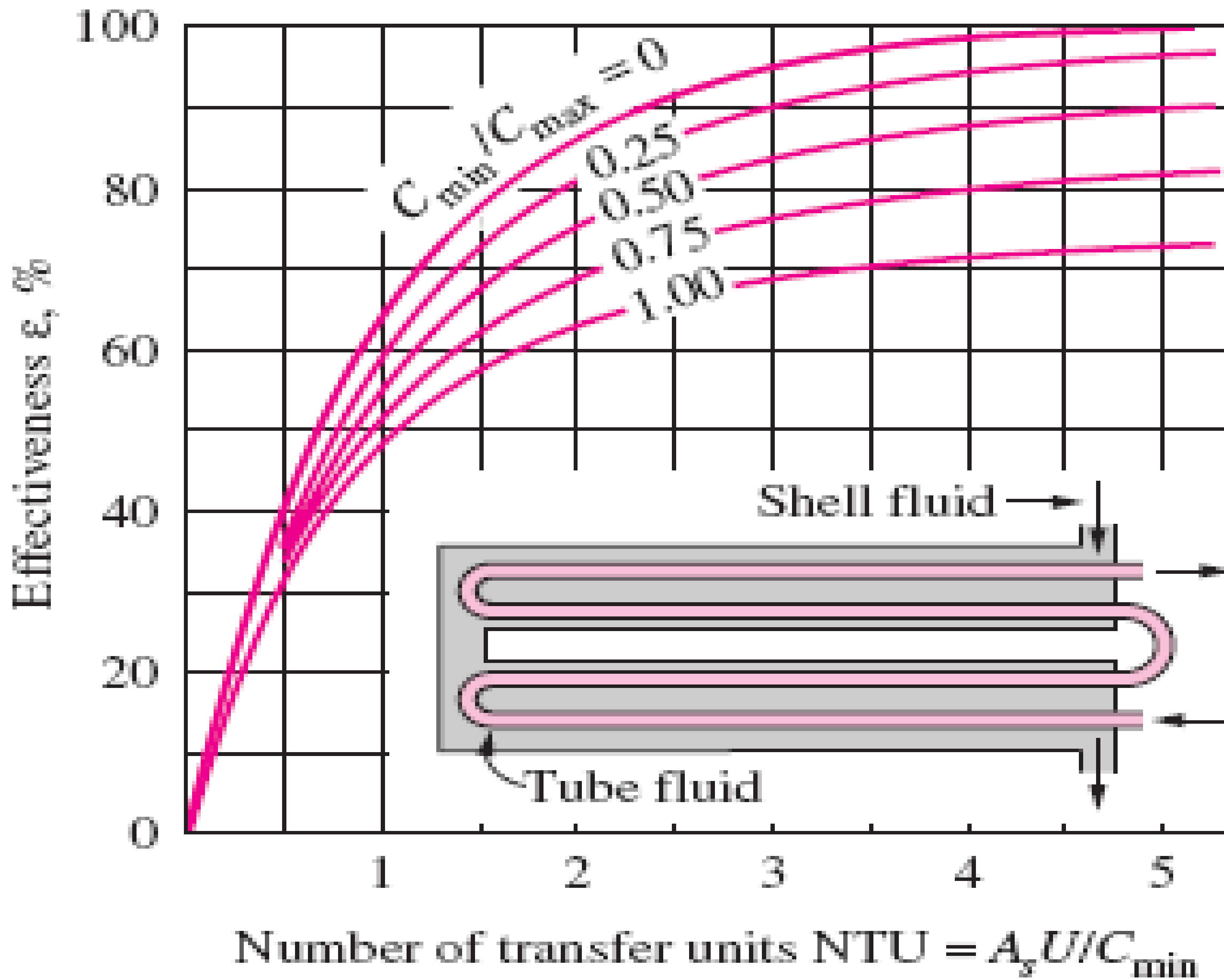
(*a*) Parallel-flow



(*b*) Counter-flow



(c) One-shell pass and 2, 4, 6, … tube passes



(*d*) Two-shell passes and 4, 8, 12, … tube passes

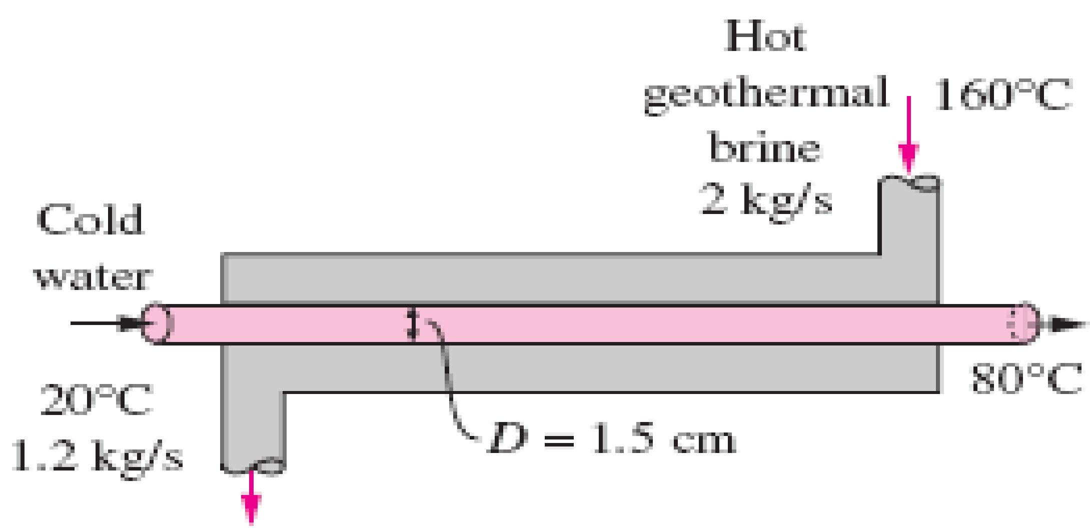
TABLE 13–5

NTU relations for heat exchangers $NTU = UA_s/C_{min}$ and $c = C_{min}/C_{max} = (\dot{m}C_p)_{min}/(\dot{m}C_p)_{max}$ (Kays and London, Ref. 5.)

Heat exchanger type	NTU relation
1 *Double-pipe:* Parallel-flow	$NTU = -\frac{\ln[1-\varepsilon(1+c)]}{1+c}$
Counter-flow	$NTU = \frac{1}{c-1}\ln\left(\frac{\varepsilon-1}{\varepsilon c-1}\right)$
2 *Shell and tube:* One-shell pass 2, 4, . . . tube passes	$NTU = -\frac{1}{\sqrt{1+c^2}}\ln\left(\frac{2/\varepsilon-1-c-\sqrt{1+c^2}}{2/\varepsilon-1-c+\sqrt{1+c^2}}\right)$
3 *Cross-flow (single-pass)* C_{max} mixed, C_{min} unmixed	$NTU = -\ln\left[1+\frac{\ln(1-\varepsilon c)}{c}\right]$
C_{min} mixed, C_{max} unmixed	$NTU = -\frac{\ln[c\ln(1-\varepsilon)+1]}{c}$
4 *All heat exchangers with $c = 0$*	$NTU = -\ln(1-\varepsilon)$

Example

A counter-flow double-pipe heat exchanger is to heat water from 20°C to 80°C at a rate of 1.2 kg/s. The heating is to be accomplished by geothermal water available at 160°C at a mass flow rate of 2 kg/s. The inner tube is thin-walled and has a diameter of 1.5 cm. If the overall heat transfer coefficient of the heat exchanger is 640 W/m2 .°C, determine the length of the heat exchanger required to achieve the desired heating.



Assumptions

1. Steady operating conditions exist.
2. The heat exchanger is well insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid.
3. Changes in the kinetic and potential energies of fluid streams are negligible.
4. There is no fouling.
5. Fluid properties are constant.

Analysis In the effectiveness–NTU method, we first determine the heat capacity rates of the hot and cold fluids and identify the smaller one:

$$C_h = \dot{m}_h C_{ph} = (2 \text{ kg/s})(4.31 \text{ kJ/kg} \cdot {}^\circ\text{C}) = 8.62 \text{ kW/}^\circ\text{C}$$
$$C_c = \dot{m}_c C_{pc} = (1.2 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot {}^\circ\text{C}) = 5.02 \text{ kW/}^\circ\text{C}$$

Therefore,

$$C_{\min} = C_c = 5.02 \text{ kW/}^\circ\text{C}$$

and

$$c = C_{\min}/C_{\max} = 5.02/8.62 = 0.583$$

Then the maximum heat transfer rate is determined from Eq. 13-32 to be

$$\begin{aligned}\dot{Q}_{\max} &= C_{\min}(T_{h,\text{ in}} - T_{c,\text{ in}}) \\ &= (5.02 \text{ kW/}^\circ\text{C})(160 - 20)^\circ\text{C} \\ &= 702.8 \text{ kW}\end{aligned}$$

That is, the maximum possible heat transfer rate in this heat exchanger is 702.8 kW. The actual rate of heat transfer in the heat exchanger is

$$\dot{Q} = [\dot{m}C_p(T_{\text{out}} - T_{\text{in}})]_{\text{water}} = (1.2\ \text{kg/s})(4.18\ \text{kJ/kg}\cdot{}^\circ\text{C})(80 - 20)^\circ\text{C} = 301.0\ \text{kW}$$

Thus, the effectiveness of the heat exchanger is

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{301.0\ \text{kW}}{702.8\ \text{kW}} = 0.428$$

Knowing the effectiveness, the NTU of this counter-flow heat exchanger can be determined from Figure 13–26*b* or the appropriate relation from Table 13–5. We choose the latter approach for greater accuracy:

$$\text{NTU} = \frac{1}{c - 1}\ln\left(\frac{\varepsilon - 1}{\varepsilon c - 1}\right) = \frac{1}{0.583 - 1}\ln\left(\frac{0.428 - 1}{0.428 \times 0.583 - 1}\right) = 0.651$$

Then the heat transfer surface area becomes

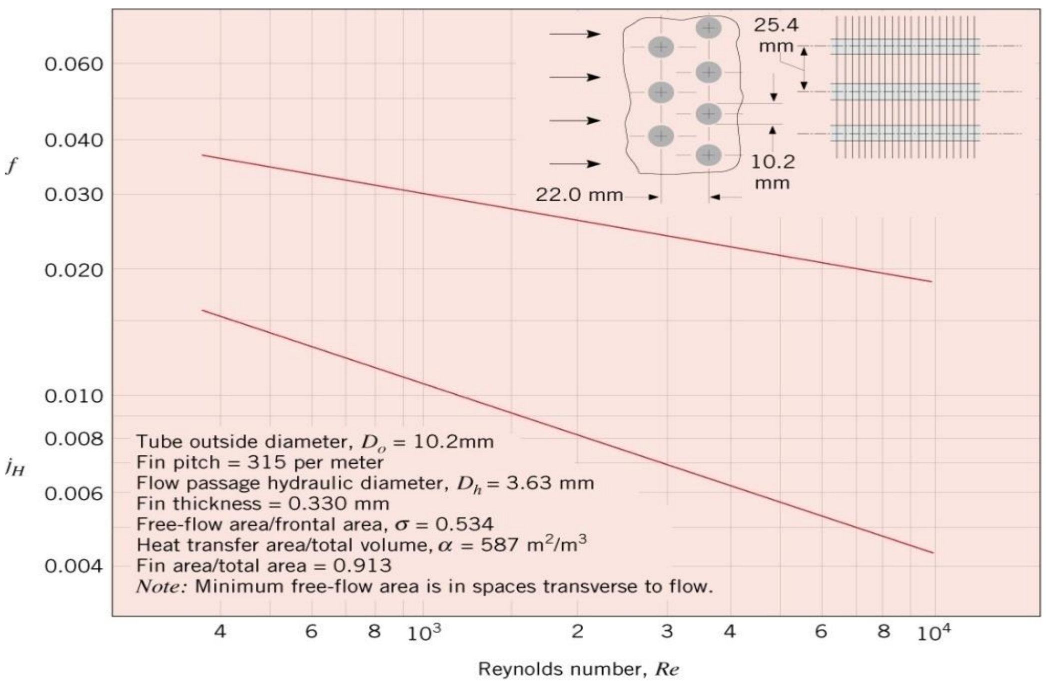
$$\text{NTU} = \frac{UA_s}{C_{\min}} \longrightarrow A_s = \frac{\text{NTU}\ C_{\min}}{U} = \frac{(0.651)(5020\ \text{W/}^\circ\text{C})}{640\ \text{W/m}^2\cdot{}^\circ\text{C}} = 5.11\ \text{m}^2$$

To provide this much heat transfer surface area, the length of the tube must be

$$A_s = \pi DL \longrightarrow L = \frac{A_s}{\pi D} = \frac{5.11\ \text{m}^2}{\pi(0.015\ \text{m})} = \mathbf{108\ m}$$

Compact Heat Exchangers

- Analysis based on $\varepsilon - NTU$ method
- Convection (and friction) coefficients have been determined for selected HX cores by Kays and London. Proprietary data have been obtained by manufacturers of many other core configurations.
- Results for a circular tube-continuous fin HX core:

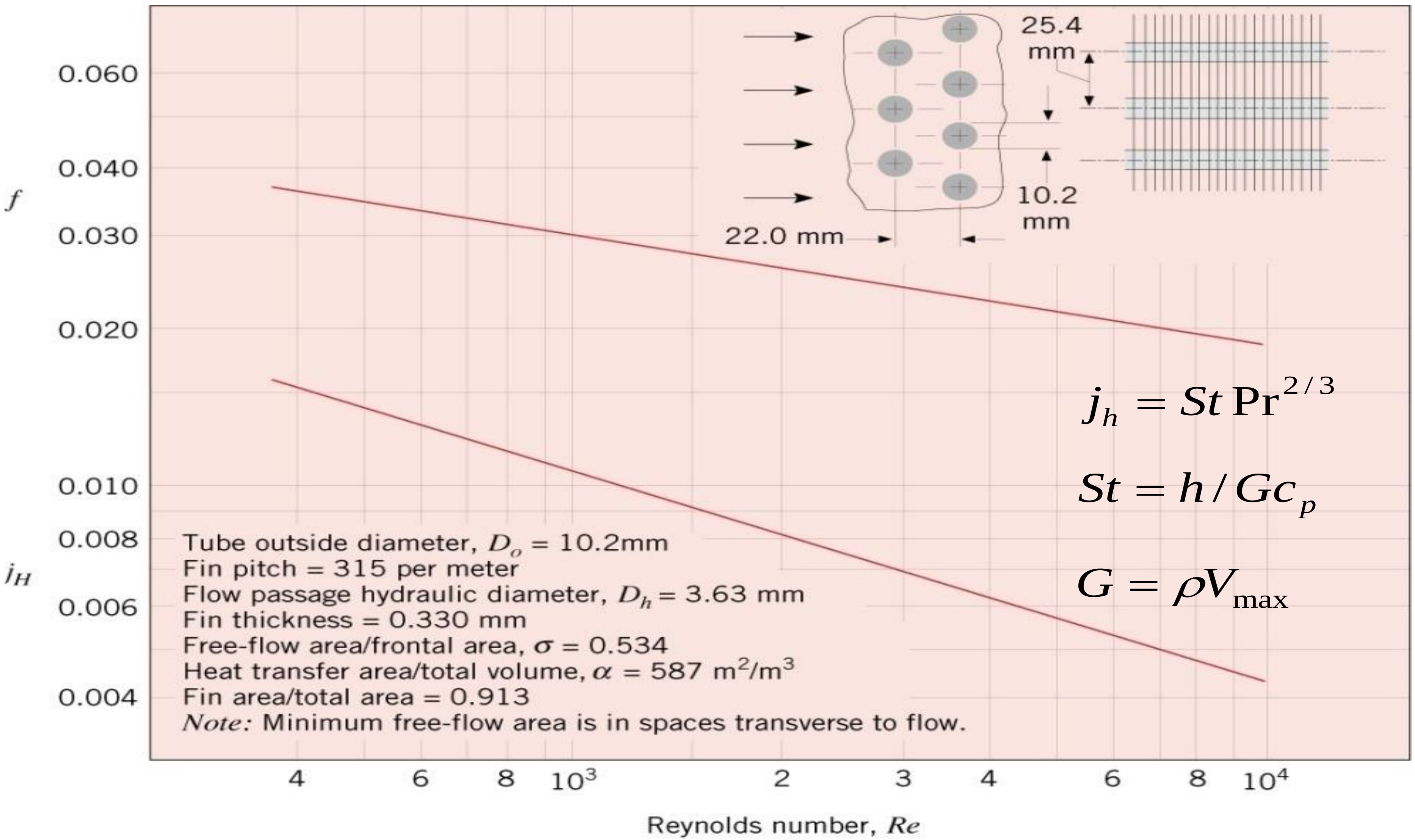


$$j_h = St\,\mathrm{Pr}^{2/3}$$

$$St = h / Gc_p$$

$$G = \rho V_{\max}$$

Results for a circular tube-continuous fin HX core

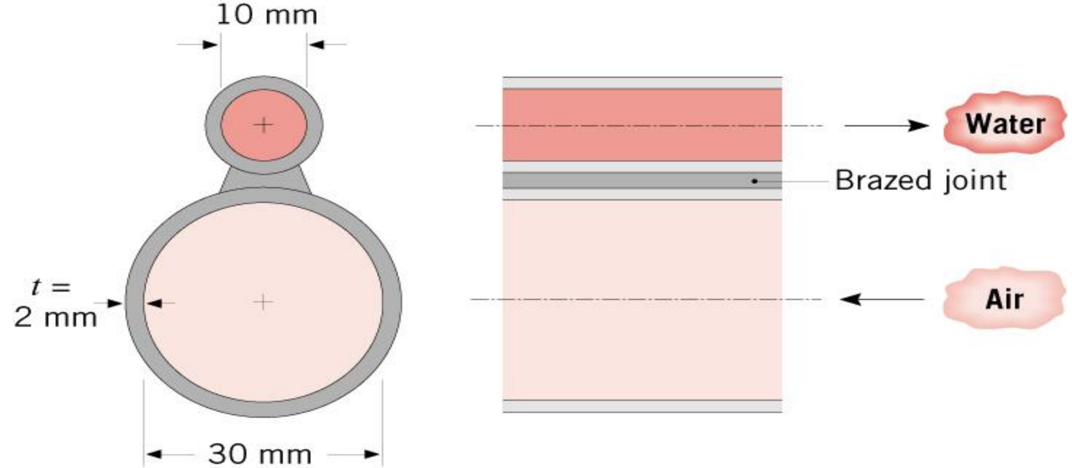


$$j_h = St\,\mathrm{Pr}^{2/3}$$

$$St = h / Gc_p$$

$$G = \rho V_{\max}$$

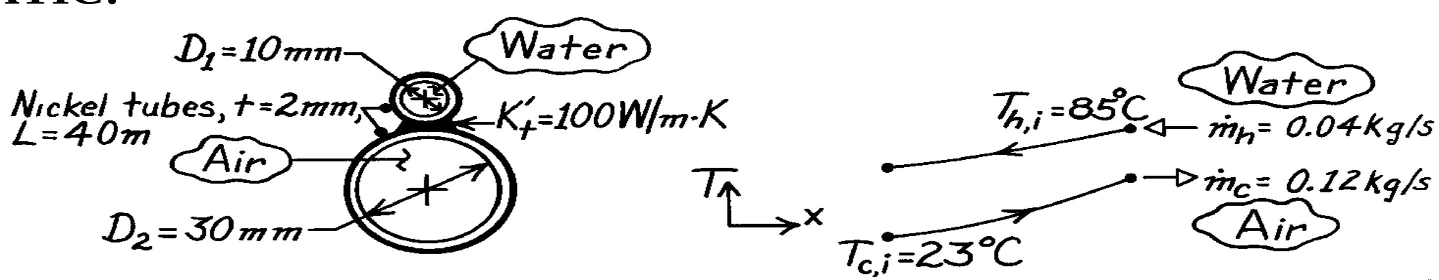
Problem 11.28: Use of twin-tube (brazed) heat exchanger to heat air by extracting energy from a hot water supply.



KNOWN: Counterflow heat exchanger formed by two brazed tubes with prescribed hot and cold fluid inlet temperatures and flow rates.

FIND: Outlet temperature of the air.

SCHEMATIC:



ASSUMPTIONS: (1) Negligible loss/gain from tubes to surroundings, (2) Negligible changes in kinetic and potential energy, (3) Flow in tubes is fully developed since L/D_h = 40 m/0.030m = 1333.

PROPERTIES: *Table A-6*, Water ($\overline{T}_h$ = 335 K): $c_h = c_{p,h}$ = 4186 J/kg·K, $\mu = 453 \times 10^{-6}$ N·s/m^2, k = 0.656 W/m·K, Pr = 2.88; *Table A-4*, Air (300 K): $c_c = c_{p,c}$ = 1007 J/kg·K, μ = 184.6 $\times 10^{-7}$ N·s/m^2, k = 0.0263 W/m·K, Pr = 0.707; *Table A-1*, Nickel ($\overline{T}$ = (23 + 85)°C/2 = 327 K): k = 88 W/m·K.

ANALYSIS: Using the NTU - ε method, from Eq. 11.30a,

$$\varepsilon = \frac{1-\exp\left[-\mathrm{NTU}\left(1-C_r\right)\right]}{1-C_r\exp\left[-\mathrm{NTU}\left(1-C_r\right)\right]} \qquad \mathrm{NTU} = UA/C_{min} \qquad C_r = C_{min}/C_{max}. \tag{1,2,3}$$

and the outlet temperature is determined from the expression

$$\varepsilon = C_c\left(T_{c,o}-T_{c,i}\right)/C_{min}\left(T_{h,i}-T_{c,i}\right). \tag{4}$$

From Eq. 11.1, the overall heat transfer coefficient is

$$\frac{1}{UA} = \frac{1}{\left(\eta_o hA\right)_h} + \frac{1}{K_t' L} + \frac{1}{\left(\eta_o hA\right)_c} \tag{5}$$

Since circumferential conduction may be significant in the tube walls, η_o needs to be evaluated for each of the tubes.

The convection coefficients are obtained as follows:

Water-side: $\quad Re_D = \dfrac{4\dot{m}_h}{\pi D \mu} = \dfrac{4 \times 0.04\ kg/s}{\pi \times 0.010 m \times 453 \times 10^{-6} N \cdot s/m^2} = 11,243.$

The flow is turbulent, and since fully developed, the Dittus-Boelter correlation may be used,

$$\overline{Nu}_h = \bar{h}_h D/k = 0.023 Re_D^{0.8} Pr^{0.3} = 0.023(11,243)^{0.8}(2.88)^{0.3} = 54.99$$

$$\bar{h}_h = 54.99 \times 0.656\ W/m \cdot K/0.01m = 3,607\ W/m^2 \cdot K.$$

Air-side: $\quad Re_D = \dfrac{4\dot{m}_c}{\pi D \mu} = \dfrac{4 \times 0.120\ kg/s}{\pi \times 0.030 m \times 184.6 \times 10^{-7}\ N \cdot s/m^2} = 275,890.$

The flow is turbulent and, since fully developed,

$$\overline{Nu}_c = \bar{h}_c D/K = 0.023 Re_D^{0.8} Pr^{0.4} = 0.023(275,890)^{0.8}(0.707)^{0.4} = 450.9$$

$$\bar{h}_c = 450.9 \times 0.0263\ W/m \cdot K/0.030m = 395.3\ W/m^2 \cdot K.$$

Water-side temperature effectiveness: $\quad A_h = \pi D_h L = \pi(0.010m)40m = 1.257\, m^2$

$$\eta_{o,h} = \eta_{f,h} = \tanh(mL_h)/mL_h \qquad m = \left(\bar{h}_h P/kA\right)^{1/2} = \left(h_h/kt\right)^{1/2}$$

$$m = \left(3607\ W/m^2 \cdot K/88\ W/m \cdot K \times 0.002m\right)^{1/2} = 143.2\, m^{-1}$$

With $L_h = 0.5\ \pi D_h$, $\eta_{o,h} = \tanh(143.2\ m^{-1} \times 0.5\ \pi \times 0.010m)/143.2\ m^{-1} \times 0.5\ \pi \times 0.010\ m =$ 0.435.

Air-side temperature effectiveness: $A_c = \pi D_c L = \pi(0.030m)40m = 3.770\ m^2$

$$\eta_{o,c} = \eta_{f,c} = \tanh\left(mL_c\right)/mL_c \quad m = \left(395.3\ W/m^2 \cdot K/88\ W/m \cdot K \times 0.002m\right)^{1/2} = 47.39\,m^{-1}$$

With $L_c = 0.5\pi D_c$, $\eta_{o,c} = \tanh(47.39\ m^{-1} \times 0.5\ \pi \times 0.030m)/47.39\ m^{-1} \times 0.5\ \pi \times 0.030m =$ 0.438.

Hence, from Eq. (5) the UA product is

$$\frac{1}{UA} = \frac{1}{0.435 \times 3607\ W/m^2 \cdot K \times 1.257\,m^2} + \frac{1}{100\ W/m \cdot K\left(40m\right)} + \frac{1}{0.438 \times 395.3\ W/m^2 \cdot K \times 3.770\,m^2}$$

$$UA = \left[5.070 \times 10^{-4} + 2.50 \times 10^{-4} + 1.533 \times 10^{-3}\right]^{-1} W/K = 437\ W/K.$$

With

$$\left.\begin{aligned} C_h &= \dot{m}_h c_h = 0.040\,kg/s \times 4186\ J/kg \cdot K = 167.4\ W/K \leftarrow C_{max} \\ C_c &= \dot{m}_c c_c = 0.120\,kg/s \times 1007\ J/kg \cdot K = 120.8\ W/K \leftarrow C_{min} \end{aligned}\right\} C_r = C_{min}/C_{max} = 0.722$$

$$NTU = \frac{UA}{C_{min}} = \frac{437\ W/K}{120.8\ W/K} = 3.62$$

and from Eq. (1) the effectiveness is

$$\varepsilon = \frac{1-\exp\left[-3.62(1-0.722)\right]}{1-0.722\ \exp\left[-3.62(1-0.722)\right]} = 0.862$$

Hence, from Eq. (4), with $C_{min} = C_c$,

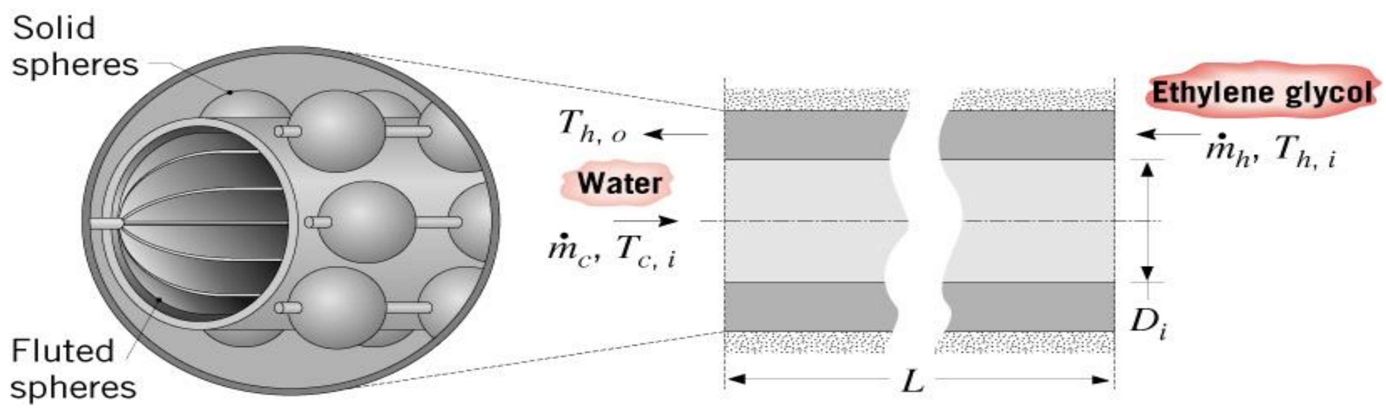
$$0.862 = \frac{C_c\left(T_{c,o} - 23°C\right)}{C_c\,(85-23)°C} \qquad T_{c,o} = 76.4°C \qquad <$$

COMMENTS: (1) Using the overall energy balance, the water outlet temperature is

$$T_{h,o} = T_{h,i} + \left(C_c / C_h\right)\left(T_{c,o} - T_{c,i}\right) = 85°C - 0.722(76.4-23)°C = 46.4°C.$$

(2) To initially evaluate the properties, we assumed that $\overline{T}_h \approx 335$ K and $\overline{T}_c \approx 300$ K. From the calculated values of $T_{h,o}$ and $T_{c,o}$, more appropriate estimates of $\overline{T}_h$ and $\overline{T}_c$ are 338 K and 322 K, respectively. We conclude that proper thermophysical properties were used for water but that the estimates could be improved for air.

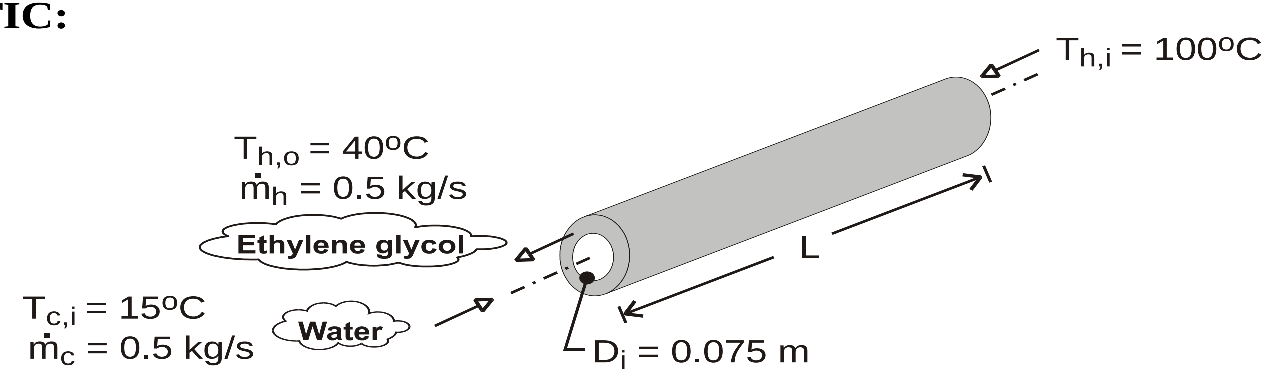
Problem 11.65: Use of *fluted spheres* and *solid spheres* to enhance the performance of a concentric tube, water/glycol heat exchanger.



KNOWN: Flow rates and inlet temperatures of water and glycol in counterflow heat exchanger. Desired glycol outlet temperature. Heat exchanger diameter and overall heat transfer coefficient without and with spherical inserts.

FIND: (a) Required length without spheres, (b) Required length with spheres, (c) Explanation for reduction in fouling and pump power associated with using spheres.

SCHEMATIC:



ASSUMPTIONS: (1) Negligible kinetic energy, potential energy and flow work changes, (2) Negligible heat loss to surroundings, (3) Constant properties, (4) Negligible tube wall thickness.

PROPERTIES: *Table A-5*, Ethylene glycol $\left(\overline{T}_h = 70°C\right)$: $c_{p,h}$ = 2606 J/kg·K; *Table A-6*, Water $\left(\overline{T}_c \approx 35°C\right)$: $c_{p,c}$ = 4178 J/kg·K.

ANALYSIS: (a) With $C_h = C_{min}$ = 1303 W/K and $C_c = C_{max}$ = 2089 W/K, C_r = 0.624. With actual and maximum possible heat rates of

$$q = C_h\left(T_{h,i} - T_{h,o}\right) = 1303 \text{ W/K}(100-40)°C = 78{,}180 \text{ W}$$

$$q_{max} = C_{min}\left(T_{h,i} - T_{c,i}\right) = 1303 \text{ W/K}(100-15)°C = 110{,}755 \text{ W}$$

the effectiveness is $\varepsilon = q/q_{max}$ = 0.706. From Eq. 11.30b,

$$NTU = \frac{1}{C_r - 1}\ln\left(\frac{\varepsilon - 1}{\varepsilon C_r - 1}\right) = -2.66 \ln\left(\frac{0.294}{0.559}\right) = 1.71$$

Hence, with $A = \pi DL$ and $NTU = UA/C_{min}$,

$$L = \frac{C_{min}\ NTU}{\pi D_i U} = \frac{1303 \text{ W/K} \times 1.71}{\pi(0.075\text{m})1000 \text{ W/m}^2\cdot\text{K}} = 9.46\text{m}$$

(b) Since $\dot{m}_c$, $\dot{m}_h$, $T_{h,i}$, $T_{h,o}$ and $T_{c,i}$ are unchanged, C_r, ε and NTU are unchanged. Hence, with U = 2000 W/m^2·K,

$$L = 4.73\text{m}$$ <

(c) Because the spheres induce mixing of the flows, the potential for contaminant build-up on the surfaces, and hence fouling, is reduced. Although the obstruction to flow imposed by the spheres acts to increase the pressure drop, the reduction in the heat exchanger length reduces the pressure drop. The second effect may exceed that of the first, thereby reducing pump power requirements.

COMMENTS: The water outlet temperature is $T_{c,o} = T_{c,i} + q/C_c = 15°C + 78{,}180$ W/2089 W/K = 52.4°C. The mean temperature $\left(\bar{T}_c = 33.7°C\right)$ is close to that used to evaluate the specific heat of water.

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The End

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