SOLUTIONS MANUAL FOR

Heat Exchangers: Selection, Rating, and Thermal Design

Third Edition

by

Sadik Kakaç
Anchasa Pramuanjaroenkij
Hongtan Liu

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CRC Press
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Boca Raton London New York

CRC Press is an imprint of the
Taylor & Francis Group, an **informa** business

CRC Press
Taylor & Francis Group
6000 Broken Sound Parkway NW, Suite 300
Boca Raton, FL 33487-2742

Printed in the United States of America on acid-free paper
Version Date: 20120627

International Standard Book Number: 978-1-4398-5131-9 (Paperback)

Visit the Taylor & Francis Web site at
http://www.taylorandfrancis.com

and the CRC Press Web site at
http://www.crcpress.com

Problem 2.1

Starting from Eq. (2.22), show that for a parallelflow heat exchanger, Eq. (2.26a) becomes

$$\frac{T_{h_2} - T_{c_2}}{T_{h_1} - T_{c_1}} = \exp\left[-\left(\frac{1}{C_h} + \frac{1}{C_c}\right)UA\right]$$

SOLUTION:

The heat transferred across the area dA is:

$$\delta Q = U(T_h - T_c)dA \qquad (1)$$

The heat transfer rate can also be written as the change in enthalpy of each fluid (with the correct sign) between the area A and A+dA:

* for the hot fluid ($dT_h<0$)

$$\delta Q = -\dot{m}_h c_{p,h} dT_h \qquad (2)$$

* for the cold fluid ($dT_c>0$)

$$\delta Q = \dot{m}_c c_{p,c} dT_c \qquad (3)$$

The notion of heat capacity can be introduced as:

$$C = \dot{m} c_p \qquad (4)$$

This parameter represents the rate of heat transferred by a fluid when its temperature varies with one degree.

The equation (2) and (3) give:

$$\delta Q = -C_h dT_h = C_c dT_c \qquad (5)$$

Equations (1) and (5) give:

$$\frac{dT_h}{T_h - T_c} = -\frac{U}{C_h} dA \qquad (6)$$

$$\frac{dT_c}{T_h - T_c} = -\frac{U}{C_c} dA \qquad (7)$$

Subtracting equation (7) from (6):

$$\frac{d(T_h - T_c)}{T_h - T_c} = \left(\frac{1}{C_c} - \frac{1}{C_h}\right) UdA \qquad (8)$$

Considering the overall heat transfer coefficient U=constant, equation (8) can be integrated:

$$\ln(T_h - T_c) = \left(\frac{1}{C_c} - \frac{1}{C_h}\right) UA + \ln B \qquad (9)$$

$$T_h - T_c = B\exp\left[\left(\frac{1}{C_c} - \frac{1}{C_h}\right) UA\right] \qquad (10)$$

The constant of integration, K is obtained from the boundary condition at the inlet:

at A=0, $T_h - T_c = T_{h1} - T_{c2}$ (11)

$$K = T_{h1} - T_{c2} \tag{12}$$

Introducing equation (12) in (10) we have:

$$\frac{T_h - T_c}{T_{h1} - T_{c2}} = \exp\left[\left(\frac{1}{C_c} - \frac{1}{C_h}\right)UA\right] \tag{13}$$

At the outlet the heat transfer area is A_t=A and T_h-T_c=T_{h2}-T_{c2} and:

$$\frac{T_{h2} - T_{c2}}{T_{h1} - T_{c1}} = e^{-\left(\frac{1}{C_h}+\frac{1}{C_c}\right)UA} \tag{14}$$

Problem 2.2

Show that for a parallel flow heat exchanger the variation of the hot fluid temperature along the heat exchanger is given by

$$\frac{T_h - T_{h1}}{T_{h1} - T_{c1}} = \frac{-C_c}{C_h + C_c}\left\{1 - e^{-\left(\frac{1}{C_h} + \frac{1}{C_c}\right)UA}\right\}$$

Obtain a similar expression for the variation of the cold fluid temperature along the heat exchanger. Also show that for $A \to \infty$, the temperature will be equal to mixing-cup temperature of the fluids which is given by

$$T_\infty = \frac{C_h T_{h1} + C_c T_{c1}}{C_h + C_c}$$

SOLUTION:

The heat transferred across the area dA is:

$$\delta Q = U\left(T_h - T_c\right)dA \qquad (1)$$

The heat transfer rate can also be written as the change in enthalpy of each fluid (with the correct sign) between the area A and A+dA:

* for the hot fluid ($dT_h<0$)

$$\delta Q = -\dot{m}_h c_{p,h} dT_h \qquad (2)$$

* for the cold fluid ($dT_c>0$)

$$\delta Q = \dot{m}_c c_{p,c} dT_c \qquad (3)$$

The notion of heat capacity can be introduced as:

$$C = \dot{m} c_p \qquad (4)$$

Equation (2) and (3) give:

$$\delta Q = -C_h dT_h = C_c dT_c \qquad (5)$$

Equations (1) and (5) give:

$$\frac{dT_h}{T_h - T_c} = -\frac{U}{C_h} dA \qquad (6)$$

$$\frac{dT_c}{T_h - T_c} = -\frac{U}{C_c} dA \qquad (7)$$

Subtracting equation (7) from (6):

$$\frac{d(T_h - T_c)}{T_h - T_c} = \left(\frac{1}{C_c} - \frac{1}{C_h}\right) UdA \qquad (8)$$

Considering the overall heat transfer coefficient U=constant, equation (8) can be integrated:

$$\ln\left(T_h - T_c\right) = \left(\frac{1}{C_c} - \frac{1}{C_h}\right)UA + \ln B \qquad (9)$$

$$T_h - T_c = B\exp\left[\left(\frac{1}{C_c} - \frac{1}{C_h}\right)UA\right] \qquad (10)$$

The constant of integration, K is obtained from the boundary condition at the inlet:

at A=0, $\quad T_h - T_c = T_{h1} - T_{c2} \qquad (11)$

$$K = T_{h1} - T_{c2} \qquad (12)$$

Introducing equation (12) in (10) we have:

$$\frac{T_h - T_c}{T_{h1} - T_{c2}} = \exp\left[\left(\frac{1}{C_c} - \frac{1}{C_h}\right)UA\right] \qquad (13)$$

From equation (10) it can be observed that the temperature difference T_h-T_c is an exponential function of surface area A, and T_h-$T_c \to 0$ when $A \to 0$. The variation of the hot fluid temperature and that of the cold fluid temperature can be obtained separately. By multiplying equations (6) and (13):

$$\frac{dT_h}{T_{h1} - T_{c2}} = -\frac{U}{C_h}\exp\left[\left(\frac{1}{C_c} - \frac{1}{C_h}\right)UA\right]dA \qquad (14)$$

Integrating:

$$\frac{T_h}{T_{h1} - T_{c1}} = -\frac{U}{C_h}\frac{\exp\left[-\left(\frac{1}{C_h} + \frac{1}{C_c}\right)UA\right]}{-\frac{C_h + C_c}{C_h C_c}U} + B \qquad (15)$$

$$\frac{T_h}{T_{h1} - T_{c2}} = \frac{C_c}{C_c - C_h}\exp\left[\left(\frac{1}{C_c} - \frac{1}{C_h}\right)UA\right] + B \qquad (16)$$

The constant of integration, B is obtained from the boundary condition:

at A=0, $\quad T_h = T_{h1}$, and

$$B = \frac{T_{h1}}{T_{h1} - T_{c2}} - \frac{C_c}{C_c - C_h} \qquad (17)$$

From (16) and (17) we have:

$$\frac{T_h - T_{h1}}{T_{h1} - T_{c1}} = \frac{-C_c}{C_h + C_c}\left\{1 - \exp\left[-\left(\frac{1}{C_h} + \frac{1}{C_c}\right)UA\right]\right\} \qquad (18)$$

From equations (7) and (13) following the same procedure we obtain:

$$\frac{T_c - T_{c1}}{T_{h1} - T_{c1}} = \frac{C_h}{C_h + C_c}\left\{1 - e^{-\left(\frac{1}{C_h} + \frac{1}{C_c}\right)UA}\right\} \qquad (19)$$

Equation (10) shows that for $A \to \infty$, $T_h = T_c = T_\infty$.

The value of T_∞ can be calculated, for example, from equation (19):

$$T_{\infty} = T_{c1} + \frac{C_c}{C_h + C_c}\left(T_{h1} - T_{c1}\right) \tag{20}$$

$$T_{\infty} = \frac{C_h T_{h1} + C_c T_{c1}}{C_h + C_c} \tag{21}$$

Problem 2.3

Show that the variation of the hot and cold fluid temperature along a counterflow heat exchanger is given by

$$\frac{T_h - T_{h1}}{T_{h1} - T_{c2}} = \frac{C_c}{C_c - C_h}\left\{\exp\left[\left(\frac{1}{Cc} - \frac{1}{C_h}\right)UA\right] - 1\right\}$$

and

$$\frac{T_c - T_{c2}}{T_{h1} - T_{c2}} = \frac{C_h}{C_c - C_h}\left\{\exp\left[\left(\frac{1}{Cc} - \frac{1}{C_h}\right)UA\right] - 1\right\}$$

SOLUTION:

$$\frac{dT_h}{T_h - T_c} = -\frac{U}{C_h} dA \qquad (1)$$

$$\frac{dT_c}{T_h - T_c} = -\frac{U}{C_c} dA \qquad (2)$$

Subtracting equation (2) from (1):

$$\frac{d(T_h - T_c)}{T_h - T_c} = \left(\frac{1}{C_c} - \frac{1}{C_h}\right) UdA \qquad (3)$$

Integrating for constant values of U, C_c and C_h we have

$$\ln\left(T_h - T_c\right) = \left(\frac{1}{C_c} - \frac{1}{C_h}\right) UA + \ln B$$

$$T_h - T_c = B\exp\left[\left(\frac{1}{C_c} - \frac{1}{C_h}\right) UA\right] \qquad (4)$$

where B the constant of integration results from the boundary condition:

at A=0, $T_h - T_c = T_{h1} - T_{c2}$

$$B = T_{h1} - T_{c2} \qquad (5)$$

Introducing equation (5) in (4):

$$\frac{T_h - T_c}{T_{h1} - T_{c2}} = \exp\left[\left(\frac{1}{C_c} - \frac{1}{C_h}\right) UA\right] \qquad (6)$$

Examining the evolution of T_h and T_c separately by multiplying equations (1) and (6), (2) and (6) respectively, we have:

$$\frac{dT_h}{T_{h1}-T_{c2}} = -\frac{U}{C_h}\exp\left[\left(\frac{1}{C_c}-\frac{1}{C_h}\right)UA\right]dA \tag{7.1}$$

$$\frac{dT_c}{T_{h1}-T_{c2}} = -\frac{U}{C_c}\exp\left[\left(\frac{1}{C_c}-\frac{1}{C_h}\right)UA\right]dA \tag{7.2}$$

Integrating:

$$\frac{T_h}{T_{h1}-T_{c2}} = -\frac{U}{C_h}\frac{\exp\left[\left(\frac{1}{C_c}-\frac{1}{C_h}\right)UA\right]}{\left(\frac{1}{C_c}-\frac{1}{C_h}\right)U}+B$$

$$\frac{T_h}{T_{h1}-T_{c2}} = \frac{C_c}{C_c-C_h}\exp\left[\left(\frac{1}{C_c}-\frac{1}{C_h}\right)UA\right]+B \tag{8.1}$$

$$\frac{T_c}{T_{h1}-T_{c2}} = -\frac{U}{C_c}\frac{\exp\left[\left(\frac{1}{C_c}-\frac{1}{C_h}\right)UA\right]}{\left(\frac{1}{C_c}-\frac{1}{C_h}\right)U}+B'$$

$$\frac{T_c}{T_{h1}-T_{c2}} = \frac{C_h}{C_c-C_h}\exp\left[\left(\frac{1}{C_c}-\frac{1}{C_h}\right)UA\right]+B' \tag{8.2}$$

For A=0, $T_h=T_{h1}$, $T_c = T_{c2}$ and:

$$\frac{T_{h1}}{T_{h1}-T_{c2}} = \frac{C_c}{C_c-C_h}+B$$

$$B = \frac{T_{h1}}{T_{h1}-T_{c2}} - \frac{C_c}{C_c-C_h} \tag{9.1}$$

$$\frac{T_{c2}}{T_{h1}-T_{c2}} = \frac{C_h}{C_c-C_h}+B'$$

$$B = \frac{T_{c2}}{T_{h1}-T_{c2}} - \frac{C_h}{C_c-C_h} \tag{9.2}$$

Substituting (9.1) in (8.1), (9.2) in (8.2), respectively:

$$\frac{T_h-T_{h1}}{T_{h1}-T_{c2}} = \frac{C_c}{C_c-C_h}\left\{\exp\left[\left(\frac{1}{C_c}-\frac{1}{C_h}\right)UA\right]-1\right\} \tag{10.1}$$

$$\frac{T_c-T_{c2}}{T_{h1}-T_{c2}} = \frac{C_h}{C_c-C_h}\left\{\exp\left[\left(\frac{1}{C_c}-\frac{1}{C_h}\right)UA\right]-1\right\} \tag{10.2}$$

Problem 2.4

From problem 2.3, show that for the case $C_h<C_c$, $\frac{d^2T_h}{dA^2}>0$ and $\frac{d^2T_c}{dA^2}>0$, and therefore temperature curves are convex and for the case $C_h>C_c$, $\frac{d^2T_h}{dA^2}<0$, and $\frac{d^2T_c}{dA^2}<0$, therefore, the temperature curves are concave (see Figure 2.6).

SOLUTION:

The hot fluid has a smaller heat capacity than the cold fluid, that is why it is the one who "commands the transfer"

Differentiating equation (10.1) in problem 2.3:

$$dT_h = d\left(T_h - T_c\right)$$

$$\frac{dT_h}{dA} = (T_{h1} - T_{c2})\left(-\frac{1}{C_h}\right)U\exp\left[\left(\frac{1}{C_c}-\frac{1}{C_h}\right)UA\right]$$

$$\frac{d^2T_h}{dA^2} = (T_{h1} - T_{c2})\left(-\frac{1}{C_h}\right)\left(\frac{1}{C_c}-\frac{1}{C_h}\right)U^2\exp\left[\left(\frac{1}{C_c}-\frac{1}{C_h}\right)UA\right]$$

$$\frac{d^2T_h}{dA^2} = \frac{(T_{h1} - T_{c2})(C_c - C_h)}{C_cC_h^2}U^2\exp\left[\left(\frac{1}{C_c}-\frac{1}{C_h}\right)UA\right] > 0 \qquad (1)$$

Similarly, from equation (10.2):

$$\frac{dT_c}{dA} = (T_{h1} - T_{c2})\left(-\frac{1}{C_c}\right)U\exp\left[\left(\frac{1}{C_c}-\frac{1}{C_h}\right)UA\right]$$

$$\frac{d^2T_c}{dA^2} = \frac{(T_{h1} - T_{c2})(C_c - C_h)}{C_c^2C_h}U^2\exp\left[\left(\frac{1}{C_c}-\frac{1}{C_h}\right)UA\right] > 0 \qquad (2)$$

Since, the second derivatives with respect to area of both T_h and T_c are positive as seen in equations (1) and (2), both the temperature curves are convex.

Problem 2.5

Show that when the heat capacities of hot and cold fluids are equal (C_c=C_h=C), the variation of the hot and cold fluid temperature along a counter flow heat exchanger are linear with the surface area as:

$$\frac{T_c - T_{c_2}}{T_{h_1} - T_{c_2}} = \frac{T_h - T_{h_1}}{T_{h_1} - T_{c_2}} = -\frac{UA}{C}$$

SOLUTION:

When the two fluids have the same heat capacity, from equation (6) in problem 2.3:

$$\delta Q = U(T_h - T_c)dA = -C_h dT_h \qquad (1)$$

In equation (10.2) in problem 2.3 when $C_c \rightarrow C_h$ we have:

$$\frac{T_c - T_{c2}}{T_{h1} - T_{c2}} = \lim \frac{C_h}{C_c - C_h}\left(e^{\frac{C_h - C_c}{C_c C_h}UA} - 1\right) = \lim\left(-C_h \frac{UA}{C_c C_h}\right) = -\frac{UA}{C_c} \qquad (2)$$

Similarly, from equation (10.1) in problem 2.3:

$$\frac{d(T_h - T_c)}{T_h - T_c} = -\frac{U}{C_c}dA, \qquad \text{When } C_c \rightarrow C_h \qquad (3)$$

But C_c=C_h=C and from (2) and (3):

$$\ln(T_h - T_c) = -\frac{UA}{C_h} + \ln D \qquad (4)$$

Problem 2.6

Assume that in a condenser, there will be no-subcooling and condensate leaves the condenser at saturation temperature, T_h. Show that variation of the coolant temperature along the condenser is given by

$$\frac{T_c - T_{c1}}{T_h - T_{c1}} = 1 - \exp\left[-\frac{UA}{C_c}\right]$$

SOLUTION:

The heat transferred along a surface element dA is:

$$\delta Q = U(T_h - T_c)dA = -C_h dT_h \qquad (1)$$

Because T_h = constant in a condenser, we can write:

$$dT_h = d(T_h - T_c) \qquad (2)$$

Using equations (1) and (2):

$$\frac{d(T_h - T_c)}{T_h - T_c} = -\frac{U}{C_c} dA\ , \qquad (3)$$

Integrating:

$$\ln(T_h - T_c) = -\frac{UA}{C_h} + \ln D$$

$$T_h - T_c = B\exp\left(-\frac{U}{C_c} A\right) \qquad (4)$$

The constant of integration, B can be calculated with the boundary condition:

$T_c = T_{c1}$, for A=0.

$$T_h - T_{c1} = B \qquad (5)$$

The temperature distribution for the cold fluid can be obtained by introducing (5) in (4) as:

$$T_h - T_c = (T_h - T_{c1})\exp\left[-\frac{UA}{C_c}\right]$$

$$\frac{T_c - T_{c1}}{T_h - T_{c1}} = 1 - \exp\left(-\frac{UA}{C_c}\right)$$

Problem 2.7

In a boiler (evaporator), the temperature of hot gases decreases from T_{h1} to T_{h2}, while boiling occurs at a constant temperature T_c. Obtain an expression, as in Problem 2.6, for the variation of hot fluid temperature with the surface area.

SOLUTION:

The rate of heat transfer δQ across the heat transfer area dA can be expressed as:

$$\delta Q = U(T_h - T_c)dA = -C_h dT_h \tag{1}$$

In an evaporator T_c = constant and

$$dT_h = d(T_h - T_c) \tag{2}$$

From equations (1) and (2):

$$\frac{d(T_h - T_c)}{T_h - T_c} = -\frac{U}{C_h}dA \tag{3}$$

$$\ln(T_h - T_c) = -\frac{UA}{C_h} + \ln D$$

$$T_h - T_c = D\exp\left(-\frac{UA}{C_h}\right) \tag{4}$$

The boundary condition at A=0 gives the value of the constant D:

at A=0 $\quad T_h = T_{h1}$

$$T_{h1} - T_c = D \tag{5}$$

Introducing (5) in (4):

$$T_h - T_c = (T_{h1} - T_c)\exp\left(-\frac{U}{C_h}A\right) \tag{6}$$

Rearranging:

$$1 - \frac{T_h - T_c}{T_{h1} - T_c} = 1 - \exp\left(-\frac{U}{C_h}A\right)$$

$$\frac{T_h - T_{h1}}{T_{h1} - T_c} = -\left[1 - \exp\left(-\frac{U}{C_h}A\right)\right] \tag{7}$$

Problem 2.8

Show that Eq. (2.46) is also applicable for $C_h > C_c$, that is $C^* = C_c/C_h$.

SOLUTION:

From Eq. (2.26b)

$$T_{h2} - T_{c1} = (T_{h1} - T_{c2})\exp\left[UA\left(\frac{1}{C_c} - \frac{1}{C_h}\right)\right] \qquad (1)$$

For the case $C_h > C_c$, $C_c = C_{\min}$, $C_h = C_{\max}$,

$$T_{h2} - T_{c1} = (T_{h1} - T_{c2})\exp\left[\frac{UA}{C_{\min}}\left(1 - \frac{C_c}{C_h}\right)\right] \qquad (2)$$

$$= (T_{h1} - T_{c2})\exp[NTU(1 - C^*)]$$

From heat balance equation

$$C_c(T_{c2} - T_{c1}) = C_h(T_{h1} - T_{h2}) \qquad (3)$$

or

$$C^*(T_{c2} - T_{c1}) = (T_{h1} - T_{h2}) \qquad (4)$$

The heat exchanger efficiency

$$\varepsilon = \frac{Q}{Q_{\max}} = \frac{C_{\min}(T_{c2} - T_{c1})}{C_{\min}(T_{h1} - T_{c1})} = \frac{T_{c2} - T_{c1}}{T_{h1} - T_{c1}}$$

$$= \frac{(T_{c2} - T_{c1})(1 - C^*)}{(T_{h1} - T_{c1})(1 - C^*)}$$

$$= \frac{T_{c2} - T_{c1} - C^*(T_{c2} - T_{c1})}{T_{h1} - T_{c1} - C^*(T_{h1} - T_{c1})} \qquad (5)$$

$$= \frac{T_{c2} - T_{c1} - T_{h1} + T_{h2}}{T_{h2} - T_{c1} - C^*(T_{h1} - T_{c2}) - T_{h2} + T_{h1} - C^*T_{c2} + C^*T_{c1}}$$

$$= \frac{T_{h2} - T_{c1} - (T_{h1} - T_{c2})}{T_{h2} - T_{c1} - C^*(T_{h1} - T_{c2})}$$

or

$$\varepsilon = \frac{1-\dfrac{T_{h1}-T_{c2}}{T_{h2}-T_{c1}}}{1-C^*\dfrac{T_{h1}-T_{c2}}{T_{h2}-T_{c1}}} \tag{6}$$

$$= \frac{1-\exp[-NTU(1-C^*)]}{1-C^*\exp[-NTU(1-C^*)]}$$

This proves that for $C_h > C_c$, Eq. (2.46) can also be derived from Eq. (2.16b).

Problem 2.9

Obtain the expression for exchanger heat transfer effectiveness, ε, for parallel flow given by Eq. (2.47).

SOLUTION:

From Eq. (2.26c)

$$T_{h2} - T_{c2} = (T_{h1} - T_{c1}) \exp\left[-UA\left(\frac{1}{C_c} + \frac{1}{C_h}\right)\right] \quad (1)$$

Assume $C_h > C_c$, $C_c = C_{\min}$, $C_h = C_{\max}$,

$$T_{h2} - T_{c2} = (T_{h1} - T_{c1}) \exp\left[-\frac{UA}{C_{\min}}\left(1 + \frac{C_c}{C_h}\right)\right] \quad (2)$$

$$= (T_{h1} - T_{c1}) \exp[-NTU(1 + C^*)]$$

From heat balance equation

$$C_c(T_{c2} - T_{c1}) = C_h(T_{h1} - T_{h2}) \quad (3)$$

or

$$C^*(T_{c2} - T_{c1}) = (T_{h1} - T_{h2}) \quad (4)$$

The heat exchanger efficiency

$$\varepsilon = \frac{Q}{Q_{\max}} = \frac{C_{\min}(T_{c2} - T_{c1})}{C_{\min}(T_{h1} - T_{c1})} = \frac{T_{c2} - T_{c1}}{T_{h1} - T_{c1}}$$

$$= \frac{(T_{c2} - T_{c1})(1 + C^*)}{(T_{h1} - T_{c1})(1 + C^*)}$$

$$= \frac{(T_{c2} - T_{c1})\left(1 + \dfrac{T_{h1} - T_{h2}}{T_{c2} - T_{c1}}\right)}{(T_{h1} - T_{c1})(1 + C^*)} \quad (5)$$

$$= \frac{T_{c2} - T_{c1} + T_{h1} - T_{h2}}{(T_{h1} - T_{c1})(1 + C^*)}$$

$$= \frac{(T_{h1} - T_{c1}) - (T_{h2} - T_{c2})}{(T_{h1} - T_{c1})(1 + C^*)}$$

$$= \frac{1 - \exp[-NTU(1 + C^*)]}{1 + C^*}$$

This proves that for $C_h > C_c$, Eq. (2.47) can be derived from Eq. (2.16c). For case $C_h < C_c$, similar result can also be obtained.

Problem 2.10

5,000 kg/hr of water will be heated from 20°C to 35°C by hot water at 140°C. A 15°C hot water temperature drop is allowed. A number of double-pipe heat exchangers with annuli and pipes each connected in series will be used. Hot water flows through the inner tube. The thermal conductivity of the material is 50 W/m.K.

Fouling factors: R_{fi} 0.000176 m^2.K/W

R_{fo} = 0.000352 m^2.K/W.

Inner tube diameters: ID = 0.0525m, OD = 0.0603m

Annulus diameters: ID = 0.0779m, OD = 0.0889m.

The heat transfer coefficients in the inner tube and in the annulus are 4620 W / m^2 · K and 1600 W / m^2 · K, respectively. Calculate the overall heat transfer coefficient and the surface area of the heat exchanger for both parallel and counter flow arrangements.

GIVEN:

-A double pipe heat exchanger, with hot water flows through the inner tube.

-Cold water inlet temperature (T_{c1}) = 20°C

-Cold water outlet temperature (T_{c2}) = 35°C

-Cold water mass flow rate ($\dot{m}_h$) = 5000 kg/hr = 1.3889 kg/s

-Hot Water inlet temperature (T_{h1}) = 140°C

-Hot water temperature drop (ΔT_h) = 15°C

-Thermal conductivity of tube material (k_w) = 50 W/m.K

-Heat transfer coefficient in the inner tube (h_i) = 4620 W/m^2.K

-Heat transfer coefficient in the annulus (h_o) = 1600 W/m^2.K

-Fouling factors: (R_{fi}) = 0.000176 m2.K/W

(R_{fo}) = 0.000352 m2.K/W

-Inner tube diameters : (ID) = 0.0525 m, (OD) = 0.0603 m

-Annulus diameters: (ID) = 0.0779 m, (OD) = 0.0889 m

FIND:

a. Overall heat transfer coefficient (U_o)

b. Surface area (A).

SOLUTION:

a.

The total thermal resistance R_t can be expressed as: [eq. (2.11)]

$$R_t = \frac{1}{UA} = \frac{1}{U_i A_i} = \frac{1}{U_o A_o}$$

$$= \frac{1}{h_i A_i} + \frac{\ln\left(r_o/r_i\right)}{2\pi kL} + \frac{R_{fi}}{A_i} + \frac{R_{fo}}{A_o} + \frac{1}{A_o h_o}$$

$$\frac{1}{U_o} = \frac{A_o}{h_i A_i} + \frac{A_o \cdot \ln\left(r_o/r_i\right)}{2\pi kL} + \frac{A_o R_{fi}}{A_i} + R_{fo} + \frac{1}{h_o}$$

$$= \frac{d_o}{h_i d_i} + \frac{d_o \cdot \ln\left(r_o/r_i\right)}{2k} + \frac{d_o R_{fi}}{d_i} + R_{fo} + \frac{1}{h_o}$$

$$= \frac{0.0603}{4620 \times 0.0525} + \frac{0.0603 \times \ln\left(\frac{0.0603}{0.0525}\right)}{2 \times 50} + \frac{0.000176 \times 0.0603}{0.0525} + 0.000352 + \frac{1}{1600}$$

$$= 0.001511 \quad W/m^2.K$$

$$Q = \dot{m}_c c_{p,c}\left(T_{c2} - T_{c1}\right) = 17.222 \times 4.179 \times (50 - 20) = 2159.12 \quad kW$$

U_o is the overall heat transfer coefficient based on outer surface area, i.e.

$$T_{h2} = T_{h1} - \frac{\dot{m}_c c_{p,c}}{\dot{m}_h c_{p,h}}(T_{c2} - T_{c1})$$

$$= 100 - \frac{62000 \times 4.179}{80000 \times 4.22} \times 30 = 76.98 \quad ^oC$$

b.

Heat balance equation:

$$Q = \dot{m}_c c_{p,c}\left(T_{c2} - T_{c1}\right)$$

$c_{p,c}$ = 4.179 kJ/kg.K (from table B.2 in appendix B)

$$\therefore \; Q = 1.3889 \times 4179 \times (35 - 20) = 87063.2 \; W$$

for parallel flow:

$$\Delta T_m = \frac{120 - 90}{\ln\frac{120}{90}} = 104.28 \quad ^oC$$

for counter flow:

$$\Delta T_m = \Delta T_1 = \Delta T_2 = 105 \ {}^oC$$

So,

$$A_o = \frac{Q}{U_o \Delta T_m} = \frac{87063.2}{661.7 \times 104.28} = 1.262 \ m^2 \qquad \text{for parallel flow.}$$

$$A_o = \frac{Q}{U_o \Delta T_m} = \frac{87063.2}{661.7 \times 105} = 1.253 \ m^2 \qquad \text{for counter flow.}$$

Problem 2.11

Water at a rate of 45,500 kg/hr is heated from 80°C to 150°C in a shell-and-tube heat-exchanger having two shell passes and eight tube passes with a total surface area of 925m². Hot exhaust gases having approximately the same thermal physical properties as air enter at 350°C and exit at 175°C. Determine the overall heat transfer coefficient based on the outside surface area.

GIVEN:

-A shell-and-tube heat exchanger having two shell passes and eight tube passes.

-Cold water inlet temperature (T_{c1}) = 80°C

-Cold water outlet temperature (T_{c2}) = 150°C

-Cold water mass flow rate ($\dot{m}_h$) = 45,500 kg/hr = 12.6389 kg/s

-Hot gas inlet temperature (T_{h1}) = 350°C

-Hot gas outlet temperature (T_{h2}) = 175°C

-Total surface area (A) = 925 m²

FIND:

Overall heat transfer coefficient (U)

SOLUTION:

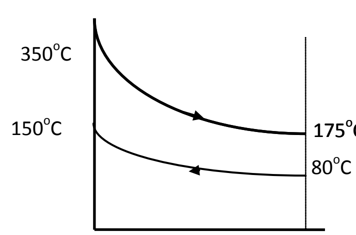
The heat balance equation:

$$Q = \dot{m}_c c_{p,c}\left(T_{c2} - T_{c1}\right) = \dot{m}_h c_{p,h}\left(T_{h2} - T_{h1}\right)$$

$c_{p,c}$ = 4.227 kJ/(kg.K) (at average temperature of $\frac{80+150}{2} = 115^o\,C$)

$$Q = \dot{m}_c c_{p,c}\left(T_{c2} - T_{c1}\right) = 17.222 \times 4.179 \times (50-20) = 2159.12 \quad kW$$

$$\Delta T_{lm,cf} = \frac{\Delta T_2 - \Delta T_1}{\ln\frac{\Delta T_2}{\Delta T_1}} = \frac{200-95}{\ln\frac{200}{95}} = 141.05 \quad {}^oC$$



$$P = \frac{\Delta T_c}{T_{h1} - T_{c1}} = \frac{70}{350-80} = 0.26$$

$$R = \frac{T_{h1} - T_{h2}}{T_{c2} - T_{c1}} = \frac{175}{70} = 2.5$$

From Figure 2.8, F=0.96

$$Q = UAF\Delta T_{lm,cf}$$

$$\therefore \ \mathrm{U} = \frac{\mathrm{Q}}{\mathrm{AF} \cdot \Delta \mathrm{T_m}} = \frac{3739.5 \times 10^3}{925 \times 0.96 \times 141.05} = 29.86 \ \ \mathrm{W/(m^2 \cdot K)}$$

Problem 2.12

A shell-and-tube heat exchanger given in Problem 2.11 is used to heat 62,000 kg / hr of water from 20°C to about 50°C. Hot water at 100°C is available. Determine how the heat transfer rate and the water outlet temperature vary with the hot water mass flow rate. Calculate the heat transfer rates and the outlet temperatures for hot water flow rates:

a. **80,000 kg / hr**

b. **40,000 kg / hr**

GIVEN:

-A shell-and-tube heat exchanger having two shell passes and eight tube passes.

-Cold water inlet temperature (T_{c1}) = 20°C

-Cold water outlet temperature (T_{c2}) = 50°C

-Cold water mass flow rate ($\dot{m}_h$) = 62,000 kg/hr = 17.222 kg/s

-Hot Water inlet temperature (T_{h1}) = 100°C

-Total surface area (A) = 925 m^2

FIND:

Heat transfer rates (Q) and outlet temperature of hot water (T_{h2}) for mass flow rate of

a. 80,000 kg/hr

b. 40,000 kg/hr

SOLUTION:

The heat balance equation:

$$Q = \dot{m}_c c_{p,c}\left(T_{c2} - T_{c1}\right) = \dot{m}_h c_{p,h}\left(T_{h2} - T_{h1}\right)$$

$$c_{p,c} = 4.179 \text{ kJ/(kg.K)} \quad (T = 35^oC)$$
$$c_{p,h} = 4.22 \text{ kJ/(kg.K)} \quad (T = 100^oC)$$

a. $\dot{m}_h = 80{,}000 \quad kg / hr$

$$Q = \dot{m}_c c_{p,c}\left(T_{c2} - T_{c1}\right) = 17.222 \times 4.179 \times (50 - 20) = 2159.12 \quad kW$$

$$T_{h2} = T_{h1} - \frac{\dot{m}_c c_{p,c}}{\dot{m}_h c_{p,h}}(T_{c2} - T_{c1})$$

$$= 100 - \frac{62000 \times 4.179}{80000 \times 4.22} \times 30 = 76.98 \quad {}^oC$$

b. $\dot{m}_h = 40{,}000 \quad kg/hr$

$$Q = \dot{m}_c c_{p,c}\left(T_{c2} - T_{c1}\right) = 17.222 \times 4.179 \times (50 - 20) = 2159.12 \quad kW$$

$$T_{h2} = T_{h1} - \frac{\dot{m}_c c_{p,c}}{\dot{m}_h c_{p,h}}(T_{c2} - T_{c1})$$

$$= 100 - \frac{62000 \times 4.179}{40000 \times 4.22} \times 30 = 53.95 \quad {}^oC$$

Problem 2.13

Water at a flow rate of 5,000 kg/hr (c_p=4182 J/kg.K) is heated from 10°C to 35°C in an oil cooler by engine oil having an inlet temperature of 65°C (c_p= 2072 J/kg.K) with a flow rate of 6,000 kg/hr. Take the overall heat transfer coefficient to be 3,500 W/m².K. What are the areas required for:

a. Parallel flow

b. Counterflow

GIVEN:

-Cold water inlet temperature (T_{c1}) = 10°C

-Cold water outlet temperature (T_{c2}) = 35°C

-Cold water mass flow rate ($\dot{m}_h$) = 5,000 kg/hr = 1.389 kg/s

-Hot Water inlet temperature (T_{h1}) = 65°C

-Hot water mass flow rate ($\dot{m}_h$) = 6,000 kg/hr = 1.667 kg/s

-Overall heat coefficient (U) = 3,500 W/m².K

-Specific heat of cold water ($c_{p,c}$) = 4182 J/kg.K

-Specific heat of hot water ($c_{p,h}$) = 2072 J/kg.K

FIND:

Heat transfer area (A) required for:

a. parallel flow;

b. counter flow.

SOLUTION:

The heat balance equation:

$$Q = \dot{m}_c c_{p,c}\left(T_{c2} - T_{c1}\right) = \dot{m}_h c_{p,h}\left(T_{h2} - T_{h1}\right)$$

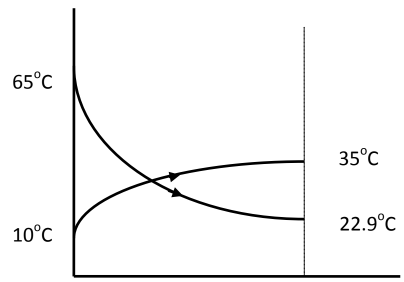
$$Q = \dot{m}_h c_{p,h}\left(T_{h2} - T_{h1}\right) = \frac{9.4}{3600} \times 1060 \times (616 - 232) = 1063 \quad W$$

$$T_{c2} = T_{c1} + \frac{\dot{m}_h c_{p,h}}{\dot{m}_c c_{p,c}}(T_{h2} - T_{h1})$$

$$= 16 + \frac{1063}{\left(\frac{0.3 \times 10^{-3} \times 999}{60}\right) \times 4180} = 67 \quad {}^oC$$

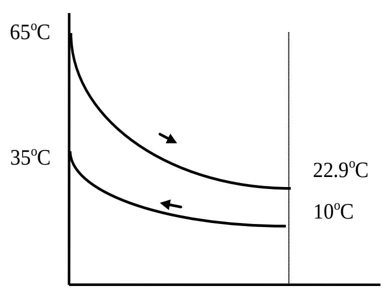
a. Parallel flow:

The parallel flow is not an acceptable solution, because of the temperatures cross.



Impossible case

b. counter flow:



$$\Delta T_{lm,cf} = \frac{\Delta T_2 - \Delta T_1}{\ln \frac{\Delta T_2}{\Delta T_1}} = \frac{(65-35)-(22.9-10)}{\ln \frac{65-35}{22.9-10}} = 20.3 \quad {}^oC$$

$$A = \frac{Q}{U \cdot \Delta T_{lm,cf}} = \frac{145.208 \times 10^3}{3500 \times 20.3} = 2.04 \ m^2$$

Problem 2.14

In order to cool a mass flow rate of 9.4 kg / h of air from 616°C to 232°C, it is passed through the inner tube of double-pipe heat exchanger with counterflow, which is 1.5 m long with an outer diameter of the inner tube of 2 cm.

a. Calculate the heat transfer rate. For air, $c_{p,h}$ = 1060 J/kg.K

b. The cooling water enters the annular side at 16°C with a mass flow rate of 0.3 L/min.

Calculate the exit temperature of the water. For water, $c_{p,c}$ = 4180 J/kg.K

c. Determine the effectiveness of this heat exchanger, NTU. The overall heat transfer coefficient based on the outside heat transfer surface area is 38.5 W/m^2.K. Calculate the surface area of the heat exchanger and number of double-pipe heat exchangers.

<u>GIVEN:</u>

-Hot air inlet temperature (T_{h1}) = 616°C

-Hot air outlet temperature (T_{h2}) = 232°C

-Hot air mass flow rate ($\dot{m}_h$) = 9.4 kg/hr = 0.002611 kg/s

-Specific heat of hot air ($c_{p,h}$) = 1060 J/kg.K

-Cooling water inlet temperature (T_{c1}) = 16°C

-Specific heat of cooling water ($c_{p,c}$) = 4180 J/kg.K

-Hot water mass flow rate ($\dot{m}_h$) = 0.3 l/min

-The overall heat transfer coefficient of the hot fluid is (U) = 38.5 W/m^2.K

<u>FIND:</u>

a. Heat transfer rate Q

b. Exit temperature of the water T

c. Effectiveness of the heat exchanger ε, NTU, and heat transfer surface area A

SOLUTION:

a.

The heat balance equation:

$$Q = \dot{m}_c c_{p,c}\left(T_{c2} - T_{c1}\right) = \dot{m}_h c_{p,h}\left(T_{h2} - T_{h1}\right)$$

$$Q = \dot{m}_h c_{p,h}\left(T_{h2} - T_{h1}\right) = \frac{9.4}{3600} \times 1060 \times (616 - 232) = 1063 \quad W$$

b.

Water density at 16°C: ρ_w = 999 kg/m^3

So, the exit temperature can be obtained by:

$$T_{c2} = T_{c1} + \frac{\dot{m}_h c_{p,h}}{\dot{m}_c c_{p,c}}(T_{h2} - T_{h1})$$

$$= 16 + \frac{1063}{\left(\frac{0.3 \times 10^{-3} \times 999}{60}\right) \times 4180} = 67 \quad {}^oC$$

c.

The heat capacity of the hot fluid is

$$C_h = \dot{m}_h c_{p,h} = \frac{9.4}{3600} \times 1060 = 2.77 \quad W/K$$

The heat capacity of the cold fluid is:

$$C_c = \dot{m}_c c_{p,c} = \frac{0.3 \times 10^{-3} \times 999}{60} \times 4180 = 20.9 \quad W/K$$

It can be observed that $C_{min} = C_h$ = 2.77 W/K and the effectiveness will be:

$$\varepsilon = \frac{\left(T_{h1} - T_{h2}\right)}{\left(T_{h1} - T_{c1}\right)} = \frac{616 - 232}{616 - 16} = 0.64$$

From the table 2.2, for a counterflow heat exchanger we have:

$$NTU = \frac{1}{1 - C^*} \ln \frac{1 - C^* \varepsilon}{1 - \varepsilon}$$

$$C^* = \frac{C_{min}}{C_{max}} = \frac{2.77}{20.9} = 0.1325$$

$$NTU = \frac{1}{1 - 0.1325} \ln \frac{1 - 0.1325 \times 0.64}{1 - 0.64} = 1.153 \ln 2.54 = 1.076$$

From the definition of NTU, the overall heat transfer coefficient is

$$U = \frac{C_{min} NTU}{A \cdot N_{hp}}$$

$$A_o = \pi DL = \pi \times 2 \times 10^{-2} \times 1.5 = 0.094 \ m^2$$

$$\therefore N_{hp} = \frac{C_{min} NTU}{A_o \cdot U_o} = \frac{2.77 \times 1.076}{0.094 \times 38.5} = 0.82 \approx 1$$

The number of hairpin is 1.

Problem 2.15

A Shell-and-tube heat exchanger is designed to heat water from 40°C to 60°C having a mass flow rate of 20,000 kg/h. Water at 180°C flows through tubes with a mass flow rate of 10,000 kg/h. The tubes have an inner diameter of d_i = 20 mm, the Renolds number is Re =10,000. The overall heat transfer coefficient is estimated to be U = 450 W/m^2.K.

a. Calculated the heat transfer Q of the heat exchanger and the exit temperature of the hot fluid.

b. If the heat exchanger is counterflow with one tube and one shell pass; determine:

i. The heat transfer area;

ii. The velocity of the fluid through the tubes;

iii. The cross-sectional area of the tubes;

iv. The number of the tubes and the length of the heat exchanger.

GIVEN

-A shell-and-tube heat exchanger (counterflow, one tube and one shell pass)

-Cold water inlet temperature (T_{c1}) = 40°C

-Hot water outlet temperature (T_{c2}) = 60°C

-Mass flow rate of cold water ($\dot{m}_c$) = 20,000 kg/h.

-Hot water inlet temperature (T_{h1}) = 180°C

-Mass flow rate of hot water ($\dot{m}_h$) = 10,000 kg/h.

-Renolds number (Re) = 10,000

-Tube diameter (d_i) = 20 mm

-Overall heat transfer coefficient (U) = 450 W/m^2.K

FIND

a. The heat transfer Q and the exit temperature of the hot fluid T_{h2}

b.

i. Heat transfer area A_o

ii. Fluid velocity in tubes u_m

iii. Cross-sectional area of tubes A_c

iv. Number of tubes N_t

SOLUTION:

a.

Cold water bulk mean temperature:

$$T_{b,c} = \frac{T_{c1}+T_{c2}}{2} = \frac{40+60}{2} = 50^{o}C$$

$\therefore c_{p,c} = 4.181$ kJ/(kg.K) at $T_{b,c}=50^{o}C$

and $c_{p,h} = 4.407$ kJ/(kg.K) at $T_{h1}= 180^{o}C$

So, from heat balance:

$$\dot{Q} = \dot{m}_c c_{p,c}(T_{c2}-T_{c1}) = \dot{m}_h c_{p,h}(T_{h1}-T_{h2})$$

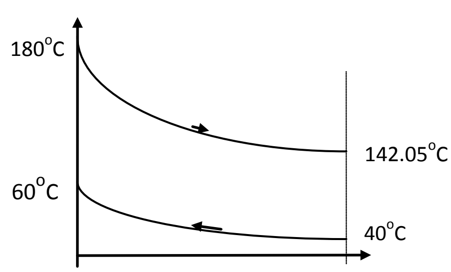
$$\dot{Q} = \dot{m}_c c_{p,c}(T_{c2}-T_{c1}) = \frac{20{,}000}{3600}\times 4181\times(60-40) = 464{,}556 \text{ W}$$

$$T_{h2} = T_{h1} - \frac{\dot{Q}}{\dot{m}_h c_{p,h}} = 180 - \frac{464{,}556}{\frac{10{,}000}{3600}\times 4407} = 142.05^{o}C$$

b.

i.

$$\dot{Q} = U\cdot\Delta T_m \cdot A$$



$\Delta T_2 = 180-60 = 120^{o}C$

$\Delta T_1 = 142.05-40 = 102.05^{o}C$

$$\Delta T_m = \frac{\Delta T_2-\Delta T_1}{\ln\left(\frac{\Delta T_2}{\Delta T_1}\right)} = \frac{120-102.05}{\ln\left(\frac{120}{102.05}\right)} = 110.78^{o}C$$

$$A = \frac{Q}{U\cdot\Delta T_m} = \frac{464{,}556}{450\times 110.78} = 9.319 \text{ m}^2$$

ii.

Bulk mean temperature of hot fluid

$$T_{b,h} = \frac{T_{h1} + T_{h2}}{2} = \frac{180 + 142.05}{2} = 161^{o}\,C$$

From Table A-27:

$$\rho = 906.23 \text{ kg/m}^3 \qquad \text{at } T_{b,h}=161^{o}C.$$

$$\mu = 1.6804 \times 10^{-4} \text{ Pa}\cdot\text{s}$$

$$Re = \frac{\rho u_m d_i}{\mu}$$

$$\therefore u_m = \frac{Re \cdot \mu}{\rho \cdot d_i} = \frac{10{,}000 \times 1.6804 \times 10^{-4}}{906.23 \times 0.02} = 0.0927 \text{ m/s}$$

iii.

$$\dot{m}_h = \rho A_c u_m$$

$$A_c = \frac{\dot{m}_h}{\rho u_m} = \frac{2.7778}{906.23 \times 0.0927} = 0.033 \text{ m}^2$$

iv.

$$N_t = \frac{A_c}{\frac{\pi d_i^2}{4}} = \frac{0.033 \times 4}{\pi \times 0.02^2} = 105.04 \approx 106$$

$$L = \frac{A_o}{\pi d_i N_t} = \frac{9.319}{\pi \times 0.02 \times 106} = 1.4 \text{ m}$$

<u>SOLUTION 2</u> (USING ε-NTU METHOD)

a.

The heat capacities for the two fluids are:

* Cold fluid

$$C_c = \dot{m}_c c_{p,c} = \frac{20{,}000}{3600} \times 4181 = 23228 \text{ W/K}$$

* Hot fluid:

$$C_h = \dot{m}_h c_{p,h} = \frac{10{,}000}{3600} \times 4407 = 12242 \text{ W/K}$$

The heat transfer rate with respect to the cold fluid is:

$$\dot{Q} = \dot{m}_c c_{p,c}(T_{c2} - T_{c1}) = \frac{20{,}000}{3600} \times 4181 \times (60 - 40) = 464{,}556 \text{ W}$$

So, in terms of heat balance:

$$\dot{Q} = \dot{m}_c c_{p,c}(T_{c2} - T_{c1}) = \dot{m}_h c_{p,h}(T_{h1} - T_{h2})$$

$$T_{h2} = T_{h1} - \frac{\dot{Q}}{\dot{m}_h c_{p,h}} = 180 - \frac{464{,}556}{\frac{10{,}000}{3600} \times 4407} = 142.05^{o}\,C$$

b.

From previous calculation, we can conclude that

$$C_{min} = C_h$$

and the effectiveness can be written as

$$\varepsilon = \frac{T_{h1} - T_{h2}}{T_{h1} - T_{c1}} = \frac{180 - 142.05}{180 - 40} = 0.271$$

The factor of non-equilibrium is:

$$C^* = \frac{C_{min}}{C_{max}} = \frac{C_h}{C_c} = \frac{12242}{23228} = 0.527$$

From Table 2.2, we have:

$$NTU = \frac{1}{1 - C^*} \ln \frac{1 - C^* \varepsilon}{1 - \varepsilon} = \frac{1}{1 - 0.527} \ln \frac{1 - 0.527 \times 0.271}{1 - 0.271}$$

$$\therefore NTU = 0.342$$

The heat transfer area is:

$$A_o = \frac{C_{min} \cdot NTU}{U} = \frac{12242 \times 0.342}{450} = 9.304 \ \ m^2$$

From the imposed Renolds number:

$$Re = \frac{\rho u_m d_i}{\mu}$$

$$\therefore u_m = \frac{Re \cdot \mu}{\rho \cdot d_i} = \frac{10{,}000 \times 1.6804 \times 10^{-4}}{906.23 \times 0.02} = 0.0927 \ \ m/s$$

The cross-sectional area of the tubes is:

$$\dot{m}_h = \rho A_c u_m$$

$$A_c = \frac{\dot{m}_h}{\rho u_m} = \frac{2.7778}{906.23 \times 0.0927} = 0.033 \ \ m^2$$

The total number of tubes necessary fro assuring the required mass flow rate is:

$$N_t = \frac{A_c}{\frac{\pi d_i^2}{4}} = \frac{0.033 \times 4}{\pi \times 0.02^2} = 105.04 \approx 106$$

The length of the heat exchanger can be calculated as

$$L = \frac{A_o}{\pi d_i N_t} = \frac{9.304}{\pi \times 0.02 \times 106} = 1.4 \ \ m$$

Problem 2.16

An oil cooler is used to cool lubricating oil from 70°C to 30°C. The cooling fluid is treated cooling water entering the exchanger at 15°C and leaving at 25°C. The specific heat capacities of the oil and water, respectively, are 2 and 4.2 kJ/(kg.K) and the oil flow rate is 4 kg/s.

a. Calculate the water flow rate $\dot{m}_c$ required.

b. Calculate the true mean temperature difference ΔT_m for a two-shell-pass-and-four-tube passes, and one-shell-pass-and-two-tube passes shell-and-tube heat exchanger and an unmixed cross-flow configuration, respectively.

c. Find the effectiveness of the heat exchangers.

GIVEN

-A shell-and-tube heat exchanger

-Cooling fluid inlet temperature (T_{c1}) = 15°C

-Cooling fluid outlet temperature (T_{c2}) = 25°C

-Lubricating oil inlet temperature (T_{h1}) = 70°C

-Lubricating oil outlet temperature(T_{h2}) = 30°C

-Mass flow rate of oil ($\dot{m}_h$) = 4 kg/s.

-The specific heat capacity of cooling water ($c_{p,c}$) = 4.2 kJ/(kg.K)

-The specific heat capacity of oil ($c_{p,h}$) = 2 kJ/(kg.K)

FIND

a. The water mass flow rate $\dot{m}_c$

b. ΔT_m for

i. two-shell-passes-and-four-tube passes

ii. one-shell-pass-and-two-tube-passes

iii. unmixed cross-flow configuration

c. Effectiveness of the heat exchangers.

SOLUTION:

a.

Heat balance equation:

$$\dot{Q} = \dot{m}_c c_{p,c}\left(T_{c2} - T_{c1}\right) = \dot{m}_h c_{p,h}\left(T_{h1} - T_{h2}\right)$$

$$\therefore \dot{m}_c = \frac{\dot{m}_h c_{p,h}\left(T_{h1} - T_{h2}\right)}{c_{p,c}\left(T_{c2} - T_{c1}\right)} = \frac{4 \times 2 \times (70-30)}{4.2 \times (25-15)} = 7.62 \ \ \text{kg/s}$$

b.

$$\Delta T_{lm,cf} = \frac{\left(T_{h1} - T_{c2}\right) - \left(T_{h2} - T_{c1}\right)}{\ln\left(\dfrac{T_{h1} - T_{c2}}{T_{h2} - T_{c1}}\right)} = \frac{(70-25)-(30-15)}{\ln\left(\dfrac{70-25}{30-15}\right)} = 27.3 \ \ ^{o}C$$

$$\Delta T_m = F \cdot \Delta T_{lm,cf}$$

where F is the correction factor.

$$R = \frac{\dot{m}_c c_{p,c}}{\dot{m}_h c_{p,h}} = \frac{7.62 \times 4.2}{4 \times 2} = 4$$

$$P = \frac{T_{c2} - T_{c1}}{T_{h1} - T_{c1}} = \frac{25-15}{70-15} = 0.18$$

i. Two-passes-shell-and-four-tube-passes:

F=0.96 (from Fig 2.8 in chapter 2)

$$\therefore \Delta T_m = F \cdot \Delta T_{lm,cf} = 0.96 \times 27.3 = 26.21 \ \ ^{c}C$$

ii. One-shell-pass-and-two-tube-passes:

F=0.90 (from Fig 2.7 in chapter 2)

$$\therefore \Delta T_m = F \cdot \Delta T_{lm,cf} = 0.90 \times 27.3 = 24.57 \ \ ^{c}C$$

iii. Unmixed cross flow

F=0.93 (from Fig 2.13 in chapter 2)

$$\therefore \Delta T_m = F \cdot \Delta T_{lm,cf} = 0.93 \times 27.3 = 25.39 \ \ ^{c}C$$

c.

Effectiveness of heat exchanger is equal to

$$\varepsilon = \frac{Q}{Q_{max}}$$

$$C_c = \dot{m}_c c_{p,c} = 7.62 \times 4200 = 32004 \ \ \text{W/K}$$

$$C_h = \dot{m}_h c_{p,h} = 4 \times 2000 = 8000 \ \ \text{W/K}$$

$$\therefore C_h = C_{min}$$

For all flow arrangements:

$$\varepsilon = \frac{C_h\left(T_{h1} - T_{h2}\right)}{C_{min}\left(T_{h1} - T_{c1}\right)} = \frac{C_c\left(T_{c2} - T_{c1}\right)}{C_{min}\left(T_{h1} - T_{c1}\right)}$$

$$\varepsilon = \frac{\left(T_{h1} - T_{h2}\right)}{\left(T_{h1} - T_{c1}\right)} = \frac{70-30}{70-15} = 0.727$$

Problem 2.17

For the oil cooler described in problem 2.16, calculate the surface area required for the shell-and-tube and unmixed cross-flow exchangers assuming the overall heat transfer coefficient U=90 W/(m^2.K). For the shell-and-tube exchanger, calculate the stream outlet temperature and compare it with the given values.

GIVEN

-A shell-and-tube heat exchanger

-Cooling fluid inlet temperature (T_{c1}) = 15°C

-Cooling fluid outlet temperature (T_{c2}) = 25°C

-Lubricating oil inlet temperature (T_{h1}) = 70°C

-Lubricating oil outlet temperature(T_{h2}) = 30°C

-Mass flow rate of oil ($\dot{m}_h$) = 4 kg/s.

-The specific heat capacity of cooling water ($c_{p,c}$) = 4.2 kJ/(kg.K)

-The specific heat capacity of oil ($c_{p,h}$) = 2 kJ/(kg.K)

-The overall heat transfer coefficient (U) = 90 W/(m^2.K)

FIND

a. The surface area A for

i. two-shell-passes-and-four-tube-passes

ii. one-shell-pass-and-two-tube-passes

iii. unmixed cross-flow configuration

b. Stream outlet temperature for shell-and-tube exchanger.

SOLUTION:

a.

$$A = \frac{Q}{U\Delta T_m}$$
$$= \frac{\dot{m}_h c_{p,h} \Delta T_h}{U\Delta T_m}$$
$$= \frac{4 \times 2000 \times (70 - 30)}{90 \cdot \Delta T_m}$$

i. Two-passes-shell and four tube passes:

$$A = \frac{320{,}000}{90 \times 26.21} = 135.7 \ \ m^2$$

ii. One shell pass and two tube passes:

$$A = \frac{320{,}000}{90 \times 24.57} = 144.7 \ \ m^2$$

iii. Unmixed cross flow

$$A = \frac{320{,}000}{90 \times 25.39} = 140.0 \ \ m^2$$

b.

$$NTU = \frac{AU}{C_{min}} = \frac{90 \times 144.7}{4 \times 2000} = 1.63$$

$$C^* = \frac{C_{min}}{C_{max}} = \frac{8000}{32004} = 0.25$$

$$\varepsilon = \frac{2}{1 + C^* + (1 + C^{*2})^{1/2} \dfrac{1 + \exp[-NTU \cdot \left(1 + C^{*2}\right)^{1/2}]}{1 - \exp[-NTU \cdot \left(1 + C^{*2}\right)^{1/2}]}} = 0.727$$

$$\varepsilon = \frac{\left(T_{h1} - T_{h2}\right)}{\left(T_{h1} - T_{c1}\right)}$$

$$T_{h2} = T_{h1} - \varepsilon\left(T_{h1} - T_{c1}\right)$$
$$= 70 - 0.727 \times (70 - 15)$$
$$= 30.02$$

The value obtained correspond to the given value.

Problem 2.18

In an oil cooler, oil flow through the heat exchanger with a mass flow rate of 8 kg/s and inlet temperature of 70°C. Specific heat of oil is 2 kJ/(kg.K). The cooling stream is treated cooling water that has specific heat capacity of 4.2 kJ/kg.K, flow rate of 20 kg/s, and inlet temperature of 15°C. Assuming a total heat exchanger surface area of 150 m^2 and an overall heat transfer coefficient of 150 W/(m^2.K), calculate the outlet temperature for two-pass shell-and tube and unmixed-unmixed cross flow units, respectively. Estimate the respective F-correction factor.

GIVEN

-A shell-and-tube heat exhanger

-Cooling fluid inlet temperature (T_{c1}) = 15°C

-Mass flow rate of water ($\dot{m}_c$) = 20 kg/s.

-Oil inlet temperatre (T_{h1}) = 70°C

-Mass flow rate of oil ($\dot{m}_h$) = 8 kg/s.

-The specific heat capacity of cooling water ($c_{p,c}$) = 4.2 kJ/(kg.K)

-The specific heat capacity of oil ($c_{p,h}$) = 2 kJ/(kg.K)

-Total surface area of heat exchanger (A) = 150 m^2

-Overall heat transfer coefficient (U) = 150 W/(m^2.K)

FIND

The outlet temperture T_{c2}, T_{h2} for two-pass shell-and-tube and unmixed cross-flow units, respectively.

SOLUTION:

$$C_c = \dot{m}_c c_{p,c} = 15 \times 1050 = 15750 \quad W/K$$

$$C_h = \dot{m}_h c_{p,h} = 5 \times 2000 = 10000 \quad W/K$$

$$\therefore C_h = C_{min}$$

$$C^* = \frac{C_{min}}{C_{max}} = \frac{10000}{15750} = 0.635$$

$$NTU = \frac{AU}{C_{min}} = \frac{600 \times 60}{10000} = 3.6$$

$$\varepsilon = \frac{2}{1 + C^* + (1 + C^{*2})^{1/2} \dfrac{1 + \exp[-NTU \cdot (1 + C^{*2})^{1/2}]}{1 - \exp[-NTU \cdot (1 + C^{*2})^{1/2}]}} = 0.702$$

Also,

$$\varepsilon = \frac{C_h(T_{h1} - T_{h2})}{C_{min}(T_{h1} - T_{c1})} = \frac{C_c(T_{c2} - T_{c1})}{C_{min}(T_{h1} - T_{c1})}$$

$$\therefore \varepsilon = \frac{(T_{h1} - T_{h2})}{(T_{h1} - T_{c1})}$$

and

$$T_{h2} = T_{h1} - \varepsilon(T_{h1} - T_{c1})$$
$$= 70 - 0.702(70 - 15)$$
$$= 31.39 \ ^oC$$

For outlet temperture of cooling water:

$$R = \frac{\dot{m}_c c_{p,c}}{\dot{m}_h c_{p,h}} = \frac{15 \times 1.05}{5 \times 2} = 1.575$$

also

$$R = \frac{T_{h1} - T_{h2}}{T_{c2} - T_{c1}}$$

thus,

$$T_{c2} = \frac{(T_{h1} - T_{h2})}{R} + T_{c1}$$
$$= \frac{70 - 31.39}{5.25} + 15$$
$$= 22.35 \ ^oC$$

To determine the correction factor F, R and P need to be first determined.

$$P = \frac{T_{c2} - T_{c1}}{T_{h1} - T_{c1}} = \frac{22.35 - 15}{70 - 15} = 0.134$$

so, the correction factor for two-pass shell-and-tube is found to be (from Fig 2.8 in chapter 2):

F=0.98

$$\Delta T_m = F \cdot \Delta T_{lm,cf}$$

$$\Delta T_{lm,cf} = \frac{(T_{h1} - T_{c2}) - (T_{h2} - T_{c1})}{\ln\left(\dfrac{T_{h1} - T_{c2}}{T_{h2} - T_{c1}}\right)} = \frac{47.65 - 16.39}{\ln\left(\dfrac{47.65}{16.39}\right)} = 29.29 \ ^oC$$

$$Q = AU\Delta T_{lm,cf} F = C_c(T_{c2} - T_{c1}) = C_h(T_{h1} - T_{h2})$$

$$T_{c2} = \frac{AU\Delta T_{lm,cf} F}{C_c} + T_{c1} = \frac{150 \times 150 \times 29.29 \times 0.98}{84000} + 15 = 22.69$$

$$T_{h2} = T_{h1} - \frac{AU\Delta T_{lm,cf}\,F}{C_h} = 70 - \frac{150 \times 150 \times 29.29 \times 0.98}{16000} = 29.63$$

For unmixed crossflow units, C^*, NTU are same as the two-pass shell-and-tube unit, so from Fig 2.15(c):

$$\varepsilon = 0.70$$

$$T_{h2} = T_{h1} - \varepsilon\left(T_{h1} - T_{c1}\right)$$

$$= 70 - 0.7(70 - 15)$$

$$= 31.5\ ^{o}C$$

$$T_{c2} = \frac{\left(T_{h1} - T_{h2}\right)}{R} + T_{c1}$$

$$= 22.33\ ^{o}C$$

$$P = \frac{T_{c2} - T_{c1}}{T_{h1} - T_{c1}} = \frac{22.35 - 15}{70 - 15} = 0.133$$

From Fig 2.13:

$$F = 0.95$$

Problem 2.19

An air blast cooler with a surface area for heat transfer of 600 m^2 and an overall heat transfer coefficient of 60 W/(m^2.K) is fed with the following streams:

Air: $\dot{m}_c$ =15 kg/s, $c_{p,c}$=1050 J/(kg.K), $T_{c,in}$=25°C

Oil: $\dot{m}_h$ =5 kg/s, $c_{p,h}$=2000 J/(kg.K), $T_{h,in}$=90°C

Calculate the stream exit temperatures, the F-factors, and the effectiveness for a two-shell-pass and four-tube-pass heat exchanger.

GIVEN

-A shell-and-tube heat exhanger

-Air inlet temperature (T_{c1}) = 25°C

-Mass flow rate of air ($\dot{m}_c$) = 15 kg/s.

-Oil inlet temperatre (T_{h1}) = 90°C

-Mass flow rate of oil ($\dot{m}_h$) = 5 kg/s.

-The specific heat capacity of air ($c_{p,c}$) = 1.05 kJ/(kg.K)

-The specific heat capacity of oil ($c_{p,h}$) = 2 kJ/(kg.K)

-Total surface area of heat exchanger (A) = 600 m^2

-Overall heat transfer coefficient (U) = 60 W/(m^2.K)

FIND

a. The outlet temperture T_{c2}, T_{h2}

b. F-facotr F

c. Effectiveness ε

for two-pass shell-and-tube and unmixed cross-flow units, respectively.

SOLUTION:

$$C_c = \dot{m}_c c_{p,c} = 15 \times 1050 = 15750 \ \ W/K$$
$$C_h = \dot{m}_h c_{p,h} = 5 \times 2000 = 10000 \ \ W/K$$
$$\therefore C_h = C_{min}$$
$$C^* = \frac{C_{min}}{C_{max}} = \frac{10000}{15750} = 0.635$$
$$NTU = \frac{AU}{C_{min}} = \frac{600 \times 60}{10000} = 3.6$$

From Fig 2.15(j):

$$\varepsilon = 0.83$$

Also,

$$\varepsilon = \frac{C_h(T_{h1} - T_{h2})}{C_{min}(T_{h1} - T_{c1})} = \frac{C_c(T_{c2} - T_{c1})}{C_{min}(T_{h1} - T_{c1})}$$
$$\therefore \varepsilon = \frac{(T_{h1} - T_{h2})}{(T_{h1} - T_{c1})}$$

and

$$T_{h2} = T_{h1} - \varepsilon(T_{h1} - T_{c1})$$
$$= 90 - 0.83(90 - 25)$$
$$= 36.05 \ \ ^oC$$

For outlet temperture of cooling water:

$$R = \frac{\dot{m}_c c_{p,c}}{\dot{m}_h c_{p,h}} = \frac{15 \times 1.05}{5 \times 2} = 1.575$$

also

$$R = \frac{T_{h1} - T_{h2}}{T_{c2} - T_{c1}}$$

thus,

$$T_{c2} = \frac{(T_{h1} - T_{h2})}{R} + T_{c1}$$
$$= \frac{90 - 36.05}{1.575} + 25$$
$$= 59.25 \ \ ^oC$$

To determine the correction factor F, R and P need to be first determined.

$$P = \frac{T_{c2} - T_{c1}}{T_{h1} - T_{c1}} = \frac{59.25 - 25}{90 - 25} = 0.53$$

so, the correction factor is found to be (from Fig 2.8 in chapter 2):

F=0.67

Problem 2.20

Water at the rate of 230 kg/h at 35 °C is available for use as a coolant in a double-pipe heat exchanger whose total surface area is 2 m^2. The water is to be used to cool oil [c_p=2.1kJ/kg·K] from an initial temperature of 120 °C. Because of other circumstances, an exit water temperature greater than 99 °C cannot be allowed. The exit temperature of the oil must not be below 60 °C. The overall heat-transfer coefficient is 300 W/m^2·K. Estimate the maximum flow rate of oil which may be cooled, assuming the flow rate of water is fixed at 230 kg/h.

GIVEN

$T_{c_1} = 35°C$

$T_{c_2} = 99°C$

$T_{h_1} = 120°C$

$T_{h_2} = 60°C$

$A = 2\ m^2$

$U = 300\ W/m^2K$

$c_{p_h} = 2.1\ kJ/kgK$

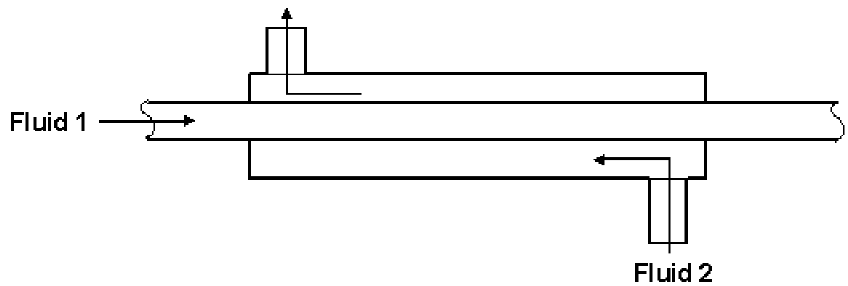
$\dot{m}_c = 230\ kg/h = 0.0639\ kg/s$

Find

$\dot{m}_h = ?$

SOLUTION:

Because of the reason T_{c_2} can not exceed T_{h_2} for parallel flow, the heat exchanger must be a counter-flow heat exchanger.



Counter-flow double-pipe heat exchanger.

Assuming water's heat capacity (c_{p_c}) is $\mathbf{4.187\ kJ/kgK}$

$$T_{h_2} - T_{c_1} = (T_{h_1} - T_{c_2})\exp\left[\text{UA}\left(\frac{1}{C_c} - \frac{1}{C_h}\right)\right]$$

$$C_c = c_{p_c}\dot{m}_c = 4.187 \times 0.0639 = 0.2675\ kW/K = 267.5\ W/K$$

$$60 - 35 = (120 - 99)\exp\left[300 \times 2\left(\frac{1}{267.5} - \frac{1}{C_h}\right)\right]$$

$$\frac{25}{21} = \exp\left[600\left(\frac{1}{267.5} - \frac{1}{C_h}\right)\right]$$

$$\ln\frac{25}{21} = 600\left(\frac{1}{267.5} - \frac{1}{C_h}\right)$$

$$C_h = 290\ W/K$$

$$C_h = c_{h_c}\dot{m}_h$$

$$290 = 2100 \times \dot{m}_h$$

$$\boxed{\dot{m}_h = 0.138\ kg/s\ = 497.2\ kg/h}$$

Problem 2.21

A counterflow double-pipe heat exchanger is used to heat liquid ammonia from 10 to 30 °C with hot water that enters the exchanger at 60 °C. The flow rate of the water is 4.0 kg/s and the overall heat-transfer coefficient is 600 W/m²·K. The area of the heat exchanger is 30 m². Calculate the flow rate of ammonia.

GIVEN

T_{h1} = 60 °C

T_{h2} = Unknown

T_{c1} = 10 °C

T_{c2} = 30 °C

$\dot{m}_{water}$ = 4 kg/s

U = 600 W/m²K

A=30 m²

$c_{p,c}$ = 4,74448 kJ/kgK (for ammonia)

$c_{p,h}$ = 4,178 kJ/kgK

Counter Flow

FIND

$\dot{m}_{ammonia}$ **= ?**

SOLUTION:

$$Q = \dot{m}_c c_{p,c}(T_{c2} - T_{c1}) = \dot{m}_h h c_{p,h}(T_{h2} - T_{h1}) \qquad (1)$$

$$\dot{m}_c \times 4.74448\frac{kJ}{kgK} \times (303K - 283K) = 4\frac{kg}{s} \times 4.178\frac{kJ}{kgK} \times (333K - T_{h2})$$

$$T_{h2} - T_{c1} = (T_{h1} - T_{c2})\exp\left[UA\left(\frac{1}{m_c\, c_{p,c}} - \frac{1}{m_h c_{p,h}}\right)\right] \quad (2)$$

$$T_{h2} - 283K = (333K - 303K)\exp\left[600\,\frac{W}{m^2 K} \times 30\, m^2 \times \left(\frac{1}{\dot{m}_c \;x\, 4.74448\ kJ/kgK} - \frac{1}{4\frac{kg}{s} x\, 4.178\ kJ/kgK}\right)\right]$$

Equation 1 and 2 are solved together, then ;

$\dot{m}_{ammonia}$ = 3.5224 kg/s

Problem 2.22

Hot water enters a counterflow heat exchanger at 90 °C. It is used to heat a cool stream of water from 5 to 30 °C. The flow rate of the cool stream is 1.3 kg/s, and the flow rate of the hot stream is 2.6 kg/s. The overall heat-transfer coefficient is 800 $W/m^2 \cdot K$. What is the area of the heat exchanger? Calculate the effectiveness of the heat exchanger.

GIVEN

T_{h1} = 90ºC

Tc_1 = 5ºC

Tc_2 = 30ºC

m_c = 1.3kg/s

m_h = 2.6 kg/s

U = 800 W/(m^2*K)

FIND

a. The area of the heat exchanger (A) =?

b. The effectiveness of the heat exchanger (ε) =?

SOLUTION:

Q = $(m * c_p)_c * \Delta Tc = (m * c_p)_h * \Delta Th$

$(c_p)_c$ = 4180 J/(kg*K) (tablo B-4)

$(c_p)_h$ = 4200 J/(kg*K) (tablo B-4)

Q = 1.3 * 4180 * (30-5) = 2.6 * 4200 * ΔTh

Q = 135.9 kW => ΔTh = 12.44 => T_{h2} = 77.56 ºC

$$\Delta Tm = \frac{\Delta T1 - \Delta T2}{\ln\left(\frac{\Delta T1}{\Delta T2}\right)}$$

$$\Delta T_m = \frac{(90-30)-(77.56-5)}{\ln\left[(90-30)/(77.56-5)\right]}$$

ΔTm = 66.08

$$A = \frac{Q}{U * \Delta Tm}$$

A=135.9*10^3/(800*66.08)

a. <u>A=2.57m^2</u>

ε = Q/Qmax

Cc = 4180 * 1.3 = 5434

Ch = 4200 * 2.6 =10920

Ch > Cc => Qmax = $(m * c_p)_c * (Th_1 - Tc_1)$

Qmax = 461.9 kW

b. <u>ε = 0.29</u>

Problem 2.23

A cross-flow finned-tube heat exchanger uses hot water to heat an appropriate quantity of air from 15 to 25 °C. The water enters the heat exchanger at 70 °C and leaves at 40 °C, and the total heat-transfer rate is to be 30 kW. The overall heat-transfer coefficient is 50 W/m^2·K. Calculate the area of the heat exchanger.

SOLUTION:

$$Q = U \times A \times F \times \Delta T_m$$

$$P = \frac{T_{c2} - T_{c1}}{T_{h1} - T_{c1}} = \frac{25 - 15}{70 - 15} = 0.18$$

$$R = \frac{T_{h1} - T_{h2}}{T_{c2} - T_{c1}} = \frac{70 - 40}{25 - 15} = 3$$

From 2.13; F = 0.94

$$\Delta T_m = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{45 - 25}{\ln\left(\frac{45}{25}\right)} = 34.03\ K$$

$$Q = U \times A \times F \times \Delta T_m$$

$$30000 = 50 \times A \times 0.94 \times 34.03$$

Hence,

$$A = 18.756\ m^2$$

Problem 2.24

A counterflow double-pipe heat exchanger is used to heat water from 20 °C to 40 °C with a hot oil which enters the exchanger at 180 °C and leaves at 140 °C. The flow rate of water is 3.0 kg/s and the overall heat-transfer coefficient is 200 W/m^2·K. Assume the specific heat for oil is 2100 J/ kg·K. Suppose the water flow rate is cut in half. What new oil flow rate would be necessary to maintain a 40 °C outlet water temperature? (The oil flow is *not* cut in half.)

GIVEN

Counterflow

T_{c1} = 20 C

T_{h1} = 180 C

T_{c2} = 40 C

T_{h2} = 140 C

m_{c1} = 3 kg/s

m_{c2} = 1.5 kg/s

m_{h2} = ?

c_{pc} = 4179 J/kgK(@30 C)

c_{ph}= 2100 J/kgK

U_c = 200 W/m2K

SOLUTION:

For cold fluid;

$$Q = U \times A \times \Delta T_M$$

$$Q = m_{c1} \times c_{pc} \times (T_{c2} - T_{c1})$$

$$\Delta T_m = \frac{\Delta T_1 - \Delta T_2}{ln\frac{\Delta T_1}{\Delta T_2}}$$

$$\Delta T_m = \frac{(180-40)-(140-20)}{ln\frac{(180-40)}{(140-20)}} = 129.74\ C$$

$$200 \times A \times 129.74 = 3 \times 4179 \times (40 - 20)$$

$$A = 9.66\ m^2$$

If water flow is cut in half, to obtain same outlet temperature for cold fluid it is needed to be changed hot fluid's outlet temperature. In this case;

$m_{c2} \times c_{pc} \times (T_{c2} - T_{c1}) = m_{h2} \times c_{ph} \times (T_{h1} - T_{h2})$

$1.5 \times 4179 \times (40 - 20) = m_{h2} \times 2100 \times (180 - T_{h2})$

$59.7 = m_{h2} \times (180 - T_{h2})$ [**]

$$200 \times 9.66 \times \frac{(180 - 40) - (T_{h2} - 20)}{ln\frac{(180 - 40)}{T_{h2} - 20}} = 1.5 \times 4179 \times (40 - 20)$$

$$ln\frac{(180 - 40)}{T_{h2} - 20} = (160 - T_{h2}) \times 0.0154$$

$$e^{(160 - T_{h2}) \times 0.0154} = \frac{140}{T_{h2} - 20}$$

$e^{0.0154 \times T_{h2}} = 0.084 \times T_{h2} - 1.68$ (*)

T_{h2} in star equation found with MATLAB.

$T_{h2} = 43\ C$

If T_{h2} is put in the double star equation;

$59.7 = m_{h2} \times (180 - T_{h2})$

$59.7 = m_{h2} \times (180 - 43)$

$m_{h2} = 0.436\ kg/s$

Problem 2.25

A shell-and-tube heat exchanger having one shell pass and four tube passes is used to heat 10 kg/s of ethylene glycol from 20 to 40 °C on the shell side; 15 kg/s of water entering at 70 °C is used in the tubes. The overall heat-transfer coefficient is 50 $W/m^2 \cdot K$. Calculate the area of the heat exchanger.

GIVEN

Th_1 = 70ºC

Tc_1 = 20ºC

Tc_2 = 40ºC

m_c = 10kg/s

m_h = 15 kg/s

U = 50 $W/(m^2*K)$

FIND

a. The area of the heat exchanger (A) =?

SOLUTION:

$Q = (m * c_p)_c * \Delta Tc = (m * c_p)_h * \Delta Th$

$(c_p)_c$ = 2483.9 J/(kg*K) (www.engineeringtoolbox.com)

$(c_p)_h$ = 4200 J/(kg*K) (tablo B-4)

Q = 10 * 2483.9 * (40-20) = 15 * 4200 * ΔTh

Q = 496.78 kW => ΔTh = 7.89 => T_{h2} = 62.11 ºC

From Figure 2.7;

P = (40-20) / (70-20) =0.4

R = (70-62.11) / (40-20) = 0.39

F = 0.975

$$\Delta Tm = \frac{\Delta T1 - \Delta T2}{\ln\left(\frac{\Delta T1}{\Delta T2}\right)}$$

$$\Delta Tm = \frac{(62{,}11 - 20) - (70 - 40)}{\ln\left(\frac{62{,}11 - 20}{70 - 40}\right)}$$

$\Delta T_m = 35.71$

$$A = \frac{Q}{U * F * \Delta Tm}$$

$$A = \frac{496780}{50 * 35{,}71 * 0{,}975}$$

<u>$A = 285.36\ m^2$</u>

Problem 3.1

A fluid flows steadily with a velocity of 6 m/s through a commercial iron rectangular duct whose sides are 1 in. by 2 in. and the length of the duct is 6 m. The average temperature of the fluid is 60°C. The fluid completely fills the duct. Calculate the surface heat transfer coefficient if the fluid is
a. Water;
b. Air at atmospheric pressure;
c. Engine oil (ρ=864 kg/m^3, c_p=2047 J/(kg.K), υ=0.0839×10^{-3} m^2/s, Pr=1050, k=0.140 W/m.K).

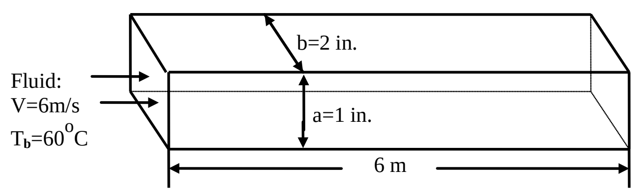
GIVEN

-Fluid flowing through a rectangular duct
-Fluid temperature (T_b) =60°C
-Fluid velocity (u_m) =6 m/s
-Inside size of pipe (a×b×L) =1 in. × 2 in. × 6 m
-Properties of Engine oil: ρ=864 kg/m^3, c_p=2047 J/(kg.K), υ=0.0839×10-3 m^2/s, Pr=1050, k=0.140 W/m.K).

FIND

The Surface heat transfer coefficient of
a. Water;
b. Air at atmospheric temperature;
c. Engine oil.

SKETCH



ASSUMPTIONS

Steady state.
Fully developed flow L/D>60.
Uniform and constant wall surface temperature.
Pipe wall is smooth.

SOLUTION:

PROPERTIES AND CONSTANTS

Properties of water at 60°C: (Table B.2 in appendix B)

$\rho = 983.1\ \text{kg/m}^3$ $\quad k = 0.651\ \text{W/(m}\cdot\text{K)}$

$\mu = 4.66\times10^{-4}\ \text{kg/m}\cdot\text{s}$ $\quad \text{Pr} = 3.0$

$c_p = 4.184\ \text{kJ/(kg}\cdot\text{K)}$

Properties of Air at 60°C: (Table B.1 in appendix B)

$\rho = 1.0595\ \text{kg/m}^3$ $\quad k = 0.0285\ \text{W/(m}\cdot\text{K)}$

$\mu = 2.0\times10^{-5}\ \text{kg/(m}\cdot\text{s)}$ $\quad \text{Pr} = 0.709$

$c_p = 1.009\ \text{kJ/(kg}\cdot\text{K)}$

Hydraulic diameter is given by:

$$D_h = \frac{4A}{P_w} = \frac{4\times2}{6} = \frac{4}{3}\ \text{in.} = 0.0339\ \text{m}$$

a.

$$\text{Re} = \frac{\rho\cdot u_m\cdot D_h}{\mu} = \frac{983.1\times6\times0.0339}{4.66\times10^{-4}} = 429104 \quad \text{(Turbulent)}$$

Using Gnielinski's correlation:

$$\text{Nu}_b = \frac{(f/2)(\text{Re}_b - 1000)\text{Pr}_b}{1.07 + 12.7(f/2)^{1/2}\left(\text{Pr}_b^{2/3} - 1\right)}$$

$$f = (1.58\ln\text{Re}_b - 3.28)^{-2}$$

$$\therefore\quad f = (1.58\times\ln(429104) - 3.28)^{-2} = 0.003376$$

$$\text{Nu}_b = \frac{(0.003376/2)\times(429104-1000)\times3}{1.07+12.7\times(0.003376/2)^{1/2}\times(3^{2/3}-1)} = 1330.1$$

$$h = \text{Nu}\cdot\frac{k}{D_h} = 1330.1\times\frac{0.651}{0.0339} = 25542.6\ \ \text{W/(m}^2\text{K)}$$

b.

$$\text{Re} = \frac{\rho\cdot u_m\cdot D_h}{\mu} = \frac{1.0595\times6\times0.0339}{2\times10^{-5}} = 10775.1 \quad \text{(Turbulent)}$$

$$f = (1.58\ln\text{Re}_b - 3.28)^{-2} = (1.58\times\ln(10775.1) - 3.28)^{-2} = 0.0077$$

$$\text{Nu}_b = \frac{(f/2)(\text{Re}_b - 1000)\text{Pr}_b}{1+12.7(f/2)^{1/2}\left(\text{Pr}_b^{2/3}-1\right)} = \frac{(0.0077/2)\times(10775.1-1000)\times0.709}{1+12.7\times(0.0077/2)^{1/2}(0.709^{2/3}-1)} = 31.82$$

$$h = \text{Nu}\cdot\frac{k}{D_h} = 31.82\times\frac{0.0285}{0.0339} = 26.75\ \ \text{W/(m}^2\text{K)}$$

c.

$$\text{Re} = \frac{\rho\cdot u_m\cdot D_h}{\mu} = \frac{u_m D_h}{\nu} = \frac{6\times0.0339}{0.0839\times10^{-3}} = 2424.3 \quad \text{(Transition)}$$

$$f = (1.58\ln\text{Re}_b - 3.28)^{-2} = (1.58\times\ln(2424.3) - 3.28)^{-2} = 0.01225$$

$$Nu_b = \frac{(f/2)(Re_b - 1000)Pr_b}{1 + 12.7(f/2)^{1/2}(Pr_b^{2/3} - 1)} = \frac{(0.01225/2) \times (2424.3 - 1000) \times 1050}{1 + 12.7 \times (0.01225/2)^{1/2}(1050^{2/3} - 1)} = 89.22$$

$$h = Nu \cdot \frac{k}{D_h} = 89.22 \times \frac{0.14}{0.0339} = 368.5 \quad W/(m^2K)$$

Problem 3.2

Air at 1.5 atm and 40°C flows through a 10-m long rectangular duct of 40 by 25 cm with a velocity of 5 m/s. The duct surface temperature is maintained at 120°C and the average air temperature at exit is 80°C. Calculate the total heat transfer using Gnielinski's correlation and check your result by energy balance.

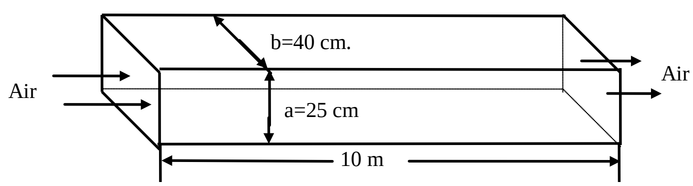
GIVEN

- -Air flowing through a rectangular duct
- -Air inlet temperature (T_{inlet}) = 40°C
- -Air outlet temperature(T_{outlet}) = 80°C
- -Inside size of pipe (a×b×L) = 40cm × 25cm × 10 m
- -Duct surface temperature(T_w)=120°C
- -Air pressure (P_{air}) =1.5 atm
- -Air velocity u_m=5 m/s

FIND

Total heat transfer coefficient.

SKETCH



ASSUMPTIONS

Steady state.
Fully developed flow L/D>60.
Uniform and constant wall surface temperature.
Pipe wall is smooth.

PROPERTIES AND CONSTANTS

The bulk mean temperature is:

$$T_b = \frac{T_{inlet} + T_{outlet}}{2} = \frac{40+80}{2} = 60^o C$$

Properties of Air at 60°C at 1 atm : (Table B.1 in Appendix B)

$\rho = 1.0595 \text{ kg/m}^3$ $\quad$ k = 0.0285 W/(m·K)

$\mu = 2.0\times10^{-5} \text{ m}^2\text{/s}$ $\quad$ Pr = 0.709

$c_p = 1.009 \text{ kJ/(kg·K)}$

Then at 1.5 atm, $\rho = 1.5 \times 1.0596 = 1.58925 \text{ kg/m}^3$

SOLUTION:

Hydraulic diameter of the duct is given by:

$$D_h = \frac{4A}{P_w} = \frac{4ab}{2(a+b)} = \frac{4\times 40\times 25}{2\times(40+25)} = 30.77 \text{ cm} = 0.3077 \text{ m}$$

$$Re = \frac{\rho u_m D_h}{\mu} = \frac{1.58925\times 5\times 0.3077}{2.0\times 10^{-5}} = 122253 \quad \text{(Turbulent)}$$

Using the simplified Gnielinski's correlation:

$$Nu_b = 0.0214(Re^{0.8} - 100)\, Pr^{0.4} \qquad \text{for } 0.5 < Pr < 1.5$$

$$Nu_b = 0.0214\times(122253^{0.8} - 100)\times(0.709)^{0.4} = 217.15$$

The heat transfer coefficient is:

$$h = \frac{Nu\cdot k}{D_h} = \frac{(217.15)\times(0.0285)}{0.3077} = 20.11 \quad W/m^2K$$

The heat transfer rate is:

$$Q = Ah(T_w - T_b) = 2(a+b)Lh(T_w - T_b)$$

$$Q = 2\cdot(40+25)\times 10^{-2}\times 10\times 20.11\times(120-60) = 15686 \quad W$$

$$Q = 15.69 \text{ kW}$$

To check your result by energy balance, we assume a volume of water with 10m length in the pipe. When it goes through the 10m duct, it travels 20m long. As the velocity is 5m/s, it takes 4s. So the total heat transfer energy is

$$W_1 = Q\times 4s = 15.69kW\times 4s = 62.76kJ$$

Meanwhile, the volume of the water is heated from 40 ^{o}C to 80 ^{o}C, So the energy contained by this volume of water is

$$W_2 = c_p m\Delta T = 1.009\times 1.58925\times(10\times 0.4\times 0.25)\times(80-40) = 64.34kJ$$

So the calculation is correct.

Problem 3.3

Calculate the heat transfer coefficient for water flowing through a 2-cm diameter tube with a velocity of 1 m/s. The average temperature of water is 60°C and the surface temperature is
a. Slightly over 60°C
b. 120°C.

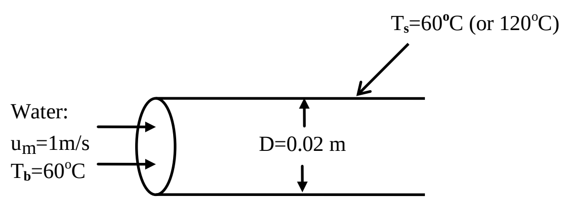
GIVEN

- -Water flowing through a circular tube
- -Water bulk mean temperature (T_b) = 60°C
- -Tube surface temperature (T_w) : (a) 60°C; (b) 120°C
- -Water velocity (u_m) = 1 m/s
- -Diameter of Tube (D) = 2 cm

FIND

Heat transfer coefficient, when surface temperature is a. slightly over 60°C and . 120°C.

SKETCH



ASSUMPTIONS

Steady state.
Fully developed flow L/D>60.
Uniform and constant wall surface temperature.
Pipe wall is smooth.

PROPERTIES AND CONSTANTS

Properties of water at 60°C: (Table B.1 in Appendix B)

$\rho = 983.2 \text{ kg/m}^3$ $\quad$ $k = 0.651 \text{ W/(m}\cdot\text{K)}$

$\mu = 0.467 \times 10^{-3} \text{ kg/(m}\cdot\text{s)}$ $\quad$ $Pr = 3.00$

SOLUTION:

Reynolds number

$$Re = \frac{\rho u_m d_i}{\mu} = \frac{983.2 \times 1 \times 0.02}{0.466 \times 10^{-3}} = 42197 \quad \text{(Turbulent)}$$

a. Using the Petukhov and Kirillov correlation:

$$Nu_b = \frac{(f/2)\,Re\,Pr}{1.07 + 12.7\sqrt{\frac{f}{2}}\left(Pr^{2/3} - 1\right)}$$

where $f = (1.58 \ln Re - 3.28)^{-2} = \left[1.58 \ln(42197) - 3.28\right]^{-2} = 0.0054$

$$Nu_b = \frac{(0.0054/2) \times 42197 \times 3}{1.07 + 12.7\sqrt{\frac{0.0054}{2}}\left(3^{2/3} - 1\right)} = 193.1$$

The heat transfer coefficient is:

$$h = \frac{Nu \cdot k}{d_i} = \frac{(193.1) \times (0.651)}{0.02} = 6285 \quad W/m^2K$$

b. At T_s=120°C, μ_w=0.23×10^{-3} kg/ms

Using the Petukhov and kirillov correlation:

$$Nu_b = \frac{(f/8)\,Re\,Pr}{1.07 + 12.7\sqrt{\frac{f}{8}}\left(Pr^{2/3} - 1\right)}\left(\frac{\mu_b}{\mu_w}\right)^n$$

Heating ⇒ n = 0.11

$$f = (1.82 \log Re_b - 1.64)^{-2}$$

$$f = [1.82 \log(42197) - 1.64]^{-2} = 0.02177$$

$$Nu_b = \frac{(0.02177/8) \times 42105.3 \times 3}{1.07 + 12.7\sqrt{\frac{0.02177}{8}}\left(3^{2/3} - 1\right)}\left(\frac{0.467}{0.235}\right)^{0.11} = 208.5$$

The heat transfer coefficient, in this case, is:

$$h = \frac{Nu \cdot k}{d_i} = \frac{(208.5) \times (0.651)}{0.02} = 6787 \quad W/m^2K$$

Problem 3.4

An oil with k=0.120 W/m.K, c_p=2000 J/kg.K, ρ=895 kg/m^3, μ=0.0041 kg/m.s flows through a 2-cm diameter tube which is 2-m long. The oil is cooled from 60°C to 30°C. The mean flow velocity is 0.4 m/s, and the tube wall temperature is maintained at 24°C (μ_w=0.021 kg/m.s). Calculate the heat transfer rate.

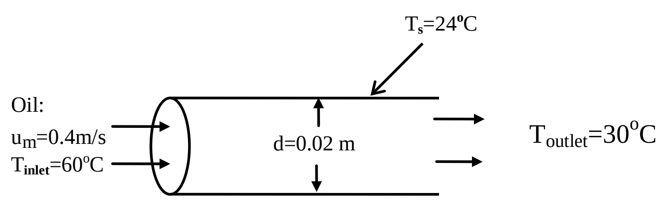
GIVEN

-Oil flowing through a circular tube
-Oil inlet temperature (T_{inlet}) = 60°C
-Oil outlet temperature (T_{outlet}) =30°C
-Tube surface temperature (T_w) : 20°C
-Water velocity (u_m) = 0.4 m/s
-Diameter of Tube (d_i) = 2 cm

FIND

Heat transfer rate.

SKETCH



ASSUMPTIONS

Steady state.
Fully developed flow L/D>60.
Uniform and constant wall surface temperature.
Pipe wall is smooth.

PROPERTIES AND CONSTANTS

The bulk mean oil temperature:

$$T_b = \frac{T_{inlet} + T_{outlet}}{2} = \frac{60+30}{2} = 45^o\,C$$

Properties of Oil at 45°C is given:

$$\rho = 895\ kg/m^3 \qquad k = 0.120\ W/(m\cdot K)$$
$$c_p = 2000\ \ J/(kg\cdot K) \qquad \mu = 0.0041\ kg/(m\cdot s)$$

$$Pr = \frac{\nu}{\alpha} = \frac{\mu c_p}{k} = \frac{(0.0041)\times(2000)}{0.12} = 68.3$$

<u>SOLUTION:</u>

The Reynolds number is:

$$\mathrm{Re} = \frac{\rho u_m \cdot d_i}{\mu} = \frac{(895)\times(0.4)\times(0.02)}{0.0041} = 1746.34 \quad \text{(laminar)}$$

Pe=Re·Pr =(1746.34)×(68.3)=119,275

$$\frac{d_i}{L} = \frac{0.02}{2} = 0.01$$

$$\left(\mathrm{Pe}\frac{d_i}{L}\right)^{\frac{1}{3}}\left(\frac{m_b}{m_w}\right)^{0.14} = \left[(119252)\times(0.01)\right]^{1/3}\left(\frac{0.0041}{0.021}\right)^{0.14} = 8.44 \quad > 2$$

Using Sieder and Tate correlation, we have:

$$\mathrm{Nu}_b = 1.86\left(\mathrm{Pe}\frac{d_i}{L}\right)^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.14} = (1.86)(8.44) = 15.7$$

The heat transfer coefficient is:

$$h = \frac{\mathrm{Nu}\cdot k}{d_i} = \frac{(15.7)\times(0.12)}{0.02} = 94.2 \quad \mathrm{W/m^2K}$$

The heat transfer rate is:

$$Q = Ah(T_b - T_w) = \pi d_i L h(T_b - T_w) = \pi(0.02)\cdot(2)\cdot(94.2)\cdot(45-24) = 248.46 \quad \mathrm{W}$$

Problem 3.5

A shell-and-tube type condenser is to be made with 3/4 in. O.D.(0.654 in. I.D.)brass tubes, and the length of the tubes between tube plates is 3 m. Under the worst conditions, cooling water is available at 21°C and the outlet temperature is to be 31°C. Water velocity inside the tubes is to be approximately 2 m/s. Vapor side film coefficient can be taken as 10,000 w/m^2.K. Calculate the overall heat transfer coefficient for this heat exchanger.

GIVEN

-Water flowing through the inner tube of a shell-and-tube condenser.
-Water inlet temperature (T_{inlet}) = 21°C
-Water outlet temperature (T_{outlet}) =31°C
-Water velocity (u_m) = 2 m/s
-Vapor side film coefficient (h_o) = 10,000 W/m^2.K
-Diameter of Tube: (D_o) = 3/4 in.;
(D_i) = 0.654 in.
-Length of the tube between the plate: (L) = 3 m.

FIND

Overall heat transfer coefficient.

ASSUMPTIONS

Steady state.
Fully developed flow L/D>60.
Uniform and constant wall surface temperature.
Pipe wall is smooth.

PROPERTIES AND CONSTANTS

The bulk mean water temperature:

$$T_b = \frac{T_{inlet} + T_{outlet}}{2} = \frac{21+31}{2} = 26^o C$$

Properties of water at 26°C is:

$\rho = 997 \text{ kg/m}^3$ $\quad$ $k = 0.608 \text{ W/(m}\cdot\text{K)}$

$c_p = 4180 \text{ J/(kg}\cdot\text{K)}$ $\quad$ $\mu = 8.72\times10^{-4} \text{ kg/(m}\cdot\text{s)}$

Pr = 6.01

SOLUTION:

d_o = 3/4 in.= 0.01905 m $\qquad$ d_i = 0.654 in. = 0.01661 m
The Reynolds number is:

$$Re = \frac{\rho u_m \cdot d_i}{\mu} = \frac{(997)\times(2)\times(0.01661)}{8.72\times10^{-4}} = 37982 \quad \text{(Turbulent)}$$

$$\because \frac{L}{d_i} = \frac{3}{0.01661} = 180.6 > 60 \qquad \therefore \text{It is fully developed flow.}$$

Using Gnielinski correlation yields:

$$Nu_b = \frac{(f/2)(Re-1000)\,Pr}{1.+12.7\left(\frac{f}{2}\right)^{1/2}\left(Pr^{2/3}-1\right)}$$

$$f = (1.58\ln Re - 3.28)^{-2}$$

$$f = (1.58\ln(37982-3.28)^{-2} = 0.00559$$

$$Nu_b = \frac{(0.00559/2)(37982-1000)(6.01)}{1.+12.7\left(\frac{0.00557}{2}\right)^{1/2}\left(5.93^{2/3}-1\right)} = 245.7$$

The inside heat transfer coefficient is:

$$h_i = \frac{Nu \cdot k}{d_i} = \frac{(245.7)\times(0.608)}{0.01661} = 8993.71 \quad W/m^2K$$

The overall heat transfer coefficient is:

$$U_o = \frac{1}{\frac{d_o}{d_i}\frac{1}{h_i} + \frac{d_o \ln(d_o/d_i)}{2k_{brass}} + \frac{1}{h_o}}$$

The thermal conductivity for brass is k_{brass}=104 W/(mK).

$$U_o = \frac{1}{\frac{0.01905}{0.01661}\frac{1}{8993.71} + \frac{0.01905\ln(0.01905/0.01661)}{2(104)} + \frac{1}{10000}}$$

$$U_o = 4165.36 \ \ W/m^2K$$

Problem 3.6

Water at 1.15 bar and 30°C is heated as it flows through a 1 in. I.D. tube at a velocity of 3m/s. The pipe surface temperature is kept constant by condensing steam outside the tube. If water outlet temperature is 80°C, calculate the surface temperature of the tube by assuming the inner surface of the tube to
a. Be smooth
b. Have a rough surface with a relative roughness of 0.001.

GIVEN

-Water flowing through the tube
-Water pressure (p) = 1.15 bar
-Water inlet temperature (T_{inlet}) = 30°C
-Water outlet temperature (T_{outlet}) =80°C
-Water velocity (u_m) = 3 m/s
-Diameter of Tube: (D_i) = 1 in.

FIND

The surface temperature of the tube, assuming the inner surface of the tube to be
a. smooth;
b. a rough surface with a relative roughness of 0.001.

ASSUMPTIONS

Steady state.
Fully developed flow L/D>60.
Uniform and constant wall surface temperature.
Tube length is L=5m.

PROPERTIES AND CONSTANTS

The bulk mean water temperature:

$$T_b = \frac{T_{inlet} + T_{outlet}}{2} = \frac{30+80}{2} = 55^o\,C$$

Properties of water at 55°C is:

$\rho = 985\ kg/m^3$ $\quad$ $k = 0.645\ W/(m\cdot K)$

$c_p = 4183\ \ J/(kg\cdot K)$ $\quad$ $\mu = 5.09\times 10^{-4}\ \ Pa.s$

$Pr = 3.31$

<u>SOLUTION:</u>

$$d_i = 1\text{ in.} = 0.0254\text{ m}$$

The Reynolds number is:

$$Re = \frac{\rho u_m \cdot d_i}{\mu} = \frac{985\times 3\times 0.0254}{5.09\times 10^{-4}} = 147460 \quad \text{(Turbulent)}$$

Mass flow rate:

$$\dot{m} = \rho u_m \frac{\pi d_i^{\,2}}{4} = 985 \times 3 \frac{\pi (0.0254)^2}{4} = 1.5 \ \ \text{kg/s}$$

Heat transfer rate:

$$\dot{Q} = \dot{m} c_p (T_{outlet} - T_{inlet}) = (1.5)(4183)(80-30) = 313725 \ \ \text{W}$$

$$\dot{Q} = hA(T_w - T_b) = h\pi d_i L (T_w - T_b)$$

Wall temperature:

$$T_w = \frac{\dot{Q}}{h\pi d_i L} + T_b$$

a. Using the Colburh analogy we have:

$$St \cdot Pr^{2/3} = \frac{f}{2}$$

where f, the friction factor, is taken from the Moody diagram (Fig. 4.1).

For smooth tubes, f=0.004

For a relative roughness of 0.001, f=0.00525

$$St = \frac{h}{\rho c_p u_m} = \frac{f}{2} Pr^{-2/3}$$

$$h = \rho c_p u_m \frac{f}{2} Pr^{-2/3}$$

For smooth tubes:

$$h = (985)(4183)(3)\left(\frac{0.004}{2}\right)(3.31)^{-2/3} = 11131 \quad \text{W}/(\text{m}^2 \cdot \text{K})$$

The wall temperature is:

$$T_w = \frac{313725}{(11131)\pi(0.0254)(5)} + 55 = 125.64 \ ^{o}C$$

b. For rough tubes

$$h = (985)(4183)(3)\left(\frac{0.00525}{2}\right)(3.28)^{-2/3} = 14698 \quad \text{W}/(\text{m}^2 \cdot \text{K})$$

In this case, the wall temperature is:

$$T_w = \frac{313725}{(14698)\pi(0.0254)(5)} + 55 = 108.5^{o}\ C$$

Another way to solve the problem is to use the Petukhov and Kirillov correlation, which is valid for both smooth and rough tubes.

Problem 3.7

A counter flow double pipe heat exchanger is used to cool the lubricating oil for a large industrial gas turbine engine. The flow rate of cooling water through the inner tube is m_c=0.2kg/s, while the flow rate of oil through the outer annulus is m_h=0.4 kg/s. The oil and water enter at temperatures of 60°C and 30°C respectively. The heat transfer coefficient in the annulus is calculated to be 8 W/m.K. The I.D. diameter of the tube of the annulus is 25mm and 45mm respectively. The outlet temperature of the oil is 40°C. Take c_p=4178 J/kg.K for water and c_p=2006 J/kg.K for oil.

a. Calculate the heat transfer coefficient in the inner tube.

b. Neglect the wall resistance and the curvature of the tube wall (assume a flat plate surface), calculate the overall heat transfer coefficient assuming water is the city water with a fouling factor of 0.000176 m^2.K/W inside the tube. Oil side is clean.

GIVEN

-A counterflow double-pipe heat exchanger
-Mass flow rate of cooling water ($\dot{m}_c$) = 0.2 kg/s
-Mass flow rate of lubricating oil ($\dot{m}_h$) = 0.4 kg/s
-Water inlet temperature (T_{c1}) = 30°C
-Oil inlet temperature (T_{h1}) = 60°C
-Oil outlet temperature (T_{h2}) =40°C
-Heat transfer coefficient in annulus side (h_o) = 8 W/m.K
-Tube diameter (d_o) = 45 mm
(d_i) = 25 mm
-Specific heat of water ($c_{p,water}$) = 4178 J/kg.K
-Specific heat of oil ($c_{p,oil}$) = 2006 J/kg.K
-Fouling factor inside the tube ($R_{f,i}$) = 0.000175 m^2.K/W

FIND

a. Heat transfer coefficient in the inner tube;
b. Overall heat transfer coefficient.

ASSUMPTIONS

Steady state.
Fully developed flow L/D>60.
Uniform and constant wall surface temperature.

<u>SOLUTION:</u>

Heat balance:

$$\dot{Q} = \dot{m}_c c_{p,c} \Delta T_c = \dot{m}_h c_{p,h} \Delta T_h$$

$$(0.2)(4178)(T_{c2}-30) = (0.4)(2006)(20)$$

$$\therefore T_{c2} = 49.21^oC$$

The bulk mean water temperature:

$$T_b = \frac{T_{c1} + T_{c2}}{2} = \frac{30 + 49.21}{2} = 39.61^o\ C$$

Properties of water at 39.61°C is:

$\rho = 992.3\ kg/m^3$ $\qquad$ $k = 0.626\ W/(m\cdot K)$

$\mu = 6.59\times10^{-4}\ m^2/s$ $\qquad$ $Pr = 4.36$

Flow area:

$$A_t = \frac{\pi d_i^2}{4} = \frac{\pi(0.025)^2}{4} = 4.9087\times10^{-4}\,m^2$$

Mass velocity:

$$G_t = \frac{\dot{m}}{A_t} = \frac{0.2}{4.9087\times10^{-4}} = 407.437\ \ kg/(m^2\cdot s)$$

Reynolds number:

$$Re = \frac{G_t d_i}{\mu} = \frac{(407.437)(0.025)}{6.59\times10^{-4}} = 15457$$

$\therefore$ Flow is turbulent.

Using Gnielinski Correlation:

$$f = (1.58\ln(Re_b) - 3.28)^{-2}$$
$$= (1.58\ln(15457) - 3.28)^{-2}$$
$$= 0.00699$$

$$Nu_b = \frac{(f/2)(Re_b - 1000)Pr_b}{1 + 12.7(f/2)^{1/2}\left(Pr^{2/3} - 1\right)}$$

$$= \frac{\left(\frac{0.00699}{2}\right)(15457 - 1000)(4.36)}{1 + 12.7\left(3.495\times10^{-3}\right)^{\frac{1}{2}}\left(4.36^{2/3} - 1\right)}$$

$$= 97.78$$

a. Heat transfer coefficient in inner tube:

$$h_i = \frac{Nu_b\cdot k}{d_i} = \frac{97.78\times0.6263}{0.025}$$
$$= 2448.4\ \ W/m^2\cdot K$$

b. The overall heat transfer coefficient neglecting tube wall resistance and tube curvature:

$$U = \frac{1}{\frac{1}{h_i} + R_{f,i} + \frac{1}{h_o}} = \frac{1}{\frac{1}{2448.4} + 0.000176 + \frac{1}{8}} = 7.96\ W/m^2\cdot K$$

Problem 3.8

City water flowing at 0.5 kg/s will be heated from 20^{o}C to 35^{o}C by hot treated boiler water at 140^{o}C. A 15^{o}C hot water temperature drop is allowed. A number of 4.50 m hairpins of 3 in. by 2 in. (schedule 40, see Table 8.2) double-pipe heat exchangers with annuli and tubes each connected in series will be used. Hot water flows through the inner tube. Calculate the heat transfer coefficient in the tube and in the annulus.

GIVEN

-Double pipe heat exchanger. City water in annulus, and hot water in inner tube.
-Mass flow rate of city water ($\dot{m}_c$) = 0.5 kg/s
-City water inlet temperature (T_{c1}) = 20^{o}C
-City water outlet temperature (T_{c2}) = 35^{o}C
-Hot water inlet temperature (T_{h1}) = 140^{o}C
-Hot water temperature drop (ΔT_h) =15^{o}C
-Hairpin: L = 4.5 m, 3 in.× 2 in. (schedule 40)
-Fouling factor ($R_{f,i}$) = 0.000175 m^2.K/W
($R_{f,o}$) = 0.000352 m^2.K/W

FIND

Heat transfer coefficient in the tube and in the annulus.

SOLUTION:

Schedule 40:

	Nominal (in.)	d_o (m)	d_i (m)
Annulus	3	0.0889	0.0779
Inner pipe	2	0.0603	0.0525

Properties:

Bulk mean temperatures:

Cold city water: $$T_{b,c} = \frac{T_{c1} + T_{c2}}{2} = \frac{20+35}{2} = 27.5^o\ C$$

Hot water: $$T_{h,c} = \frac{T_{h1} + T_{h2}}{2} = \frac{140+(140-15)}{2} = 132.5^o\ C$$

Annulus - cold city water at 27.5^{o}C	Inner tube - hot water at 132.5^{o}C
ρ_c = 996 kg/m^3	ρ_h = 932 kg/m^3
k = 0.61 W/(m.K)	k = 0.688 W/(m.K)
μ = 0.843×10^{-3} Pa.s	μ = 0.207×10^{-3} Pa.s
$c_{p,c}$=4.179 kJ/(kg.K)	$c_{p,h}$=4.269 kJ/(kg.K)
Pr = 5.79	Pr = 1.28

a. Annulus:

Velocity of cold water u_c:

$$A_{annulus} = \frac{\pi}{4}\left(D_i^{\,2} - d_o^{\,2}\right) = \frac{\pi}{4}\left(0.0779^2 - 0.0603^2\right) = 1.91\times10^{-3}\ \ m^2$$

$$u_c = \frac{\dot{m}_c}{A_{annulus}\rho_c} = \frac{0.5}{\frac{\pi}{4}\left(0.0779^2 - 0.0603^2\right)\left(996.412\right)} = 0.263 \ \ m/s$$

Hydraulic diameter of annulus:

$$D_h = \frac{4A_c}{P_w} = D_i - d_o = 0.0779 - 0.0603 = 0.0176 \ \ m$$

$$D_e = \frac{4A_c}{\pi d_o} = \frac{4 \times 1.91 \times 10^{-3}}{\pi(0.0603)} = 0.0403 \ \ m$$

Reynolds number:

$$Re = \frac{\rho_c \cdot u_c \cdot D_h}{\mu} = \frac{996 \times 0.263 \times 0.0176}{0.843 \times 10^{-3}} = 5469 \quad \text{(Turbulent)}$$

Using Gnielinski's Correlation:

$$f = (1.58 \ln Re_b - 3.28)^{-2}$$

$$f = (1.58 \ln 50346 - 3.28)^{-2}$$

$$f = 0.00523$$

$$Nu = \frac{\left(f/2\right)(Re - 1000)\,Pr}{1. + 12.7\left(\frac{f}{2}\right)^{1/2}\left(Pr^{\frac{2}{3}} - 1\right)}$$

$$Nu_b = \frac{(0.00523/2) \times 50346 \times 7.05}{1 + 8.7 \times (0.00523/2)^{1/2} \times (7.05 - 1)}$$

$$Nu_b = 251.5$$

$$\Rightarrow h_o = \frac{Nu \cdot k}{D_e} = \frac{41.4 \times 0.61}{0.0403} = 626.65 \text{ W/m}^2\text{K}$$

b. Inner tube:

$$c_{p,h}\dot{m}_h \Delta T_h = c_{p,c}\dot{m}_c \Delta T_c$$

$$\dot{m}_h = \frac{c_{p,c}\dot{m}_c \Delta T_c}{c_{p,h}\Delta T_h} = \frac{4.179 \times 0.5 \times (35 - 20)}{4.269 \times 15} = 0.489 \ kg/s$$

$$u_h = \frac{\dot{m}_h}{\rho_h \cdot \frac{\pi d_i^{\,2}}{4}} = \frac{0.489}{932 \times \frac{\pi \times 0.0525^2}{4}} = 0.2424 \ \ m/s$$

$$Re = \frac{\rho_h \cdot u_h \cdot d_i}{\mu} = \frac{(932)(0.2424)(0.0525)}{0.207 \times 10^{-3}} = 57298 \quad \text{(Turbulent)}$$

Using Gnielinski's Correlation:

$$f = (1.58 \ln Re - 3.28)^{-2}$$

$$f = (1.58 \ln(57298) - 3.28)^{-2}$$

$$f = 0.00508$$

$$Nu = \frac{\left(f/2\right)(Re - 1000)\,Pr}{1. + 12.7\left(\frac{f}{2}\right)^{1/2}\left(Pr^{\frac{2}{3}} - 1\right)}$$

$$Nu = \frac{(0.00508/2) \times (57298 - 1000) \times 1.28}{1 + 12.7 \times (0.00254)^{1/2} \times (1.28^{\frac{2}{3}} - 1)}$$

$$Nu = 164.2$$

$$\Rightarrow h_i = \frac{Nu \cdot k}{d_i} = \frac{164.2 \times 0.688}{0.0525} = 2151.8\ W/m^2K$$

Problem 3.9

It is proposed that the stationary diesel exhaust gases (CO_2) be used for heating air by the use of a double pipe heat exchanger. From the measurements of the velocity, it is calculated that the mass flow rate of gases from the engine is 100 kg/h.. The exhaust gas temperature is 600K. The air is available at 20°C and it will be heated to 80°C at a mass flow rate of 90 kg/h. Standard(Schedule 40) tube size 4×3 in. double pipe heat exchanger (hairpins) 2m in length (copper tubes) will be used. Air side heat transfer coefficient in tube is 25 W/m^2.K. Neglecting the tube wall resistance, and assuming clean and smooth surfaces, calculate the overall heat transfer coefficient.

GIVEN

-A double-pipe heat exchanger
-Mass flow rate of air ($\dot{m}_c$) = 90 kg/h = 0.025 kg/s
-Mass flow rate of exhausted gases ($\dot{m}_h$) = 100 kg/h = 0.02778 kg/s
-Air inlet temperature (T_{c1}) = 20°C
-Air outlet temperature (T_{c2}) = 80°C
-Exhausted gas temperature (T_{h1}) = 600 K
-Heat transfer coefficient in tube side (h_i) = 25 W/m^2.K
-Copper hairpins: L=2 m; 4×3 in.

FIND

Overall heat transfer coefficient.

ASSUMPTIONS

Steady state.
Fully developed flow L/D>60.
Clean and smooth surfaces.

<u>SOLUTION:</u>

Assume the average temperature of CO_2 as 500K, thus the following properties:

$\rho = 1.0732 \text{ kg/m}^3$ $k = 0.03352 \text{ W/(m}\cdot\text{K)}$

$\nu = 21.67 \times 10^{-6} \text{ m}^2\text{/s}$ $Pr = 0.702$

$c_p = 1013 \text{ J/kg.K}$

The bulk mean air temperature:

$$T_b = \frac{T_{c1} + T_{c2}}{2} = \frac{20+80}{2} = 50^o \text{ C}$$

Properties of air at 50°C is:

$\nu = 17.94 \times 10^{-6} \text{ m}^2\text{/s}$ $c_p = 1009 \text{ J/kg.K}$ $\rho = 1.093$

Geometry for seamless copper tubing:

4 in. Schedule 40: D_i = 4.026 in. = 0.10226 m
3 in. Schedule 40: d_o = 3.5 in. = 0.0889 m
d_i = 3.068 in. = 0.07793 m

Hydraulic diameter of annulus:

$$D_h = D_i - d_o = 0.10226 - 0.0889 = 0.01336 \text{ m}$$

$$D_e = \frac{D_i^{\,2} - d_o^{\,2}}{d_o} = \frac{0.10226^2 - 0.0889^2}{0.0889} = 0.02873 \ \ \text{m}$$

Velocity in annulus:

$$u_a = \frac{\dot{m}_h}{\rho A_c} = \frac{0.02778}{(1.0732)(2.006 \times 10^{-3})} = 12.9 \ \ \text{m/s}$$

Flow area:

$$A_c = \frac{\pi}{4}\left(D_i^{\,2} - d_o^{\,2}\right) = \frac{\pi}{4}\left(0.10226^2 - 0.0889^2\right) = 2.006 \times 10^{-3} \quad \text{m}^2$$

Reynolds number:

$$Re_a = \frac{u_a D_h}{\nu} = \frac{(12.9)(0.01336)}{21.67 \times 10^{-6}} = 7953$$

Using Kays and Crawford Correlation (for gases: Pr≈0.5~1.0 and Re≥5000)

$$Nu = 0.022 Re^{0.8} Pr^{0.5}$$
$$= 0.022 \times 7953^{0.8} \times 0.702^{0.5}$$
$$= 24.32$$

Heat transfer coefficient in annulus:

$$h_o = \frac{Nu \cdot k}{D_e} = \frac{24.32 \times 0.03352}{0.02873}$$
$$= 28.37 \ \ \text{W/m}^2 \cdot \text{K}$$

The overall heat transfer coefficient neglecting tube wall resistance:

$$U = \left[\frac{d_o}{d_i h_i} + \frac{1}{h_o}\right]^{-1}$$
$$= \left[\frac{3.5}{(3.068)(25)} + \frac{1}{28.37}\right]^{-1}$$
$$= 12.36 \ \ \text{W/m}^2 \cdot \text{K}$$

Problem 3.10

Consider the laminar flow of an oil inside a duct with a Reynolds number of 1000. The length of the duct is 2.5m and the diameter is 2 cm. The duct is heated electrically by the use of its walls as an electrical resistance. Properties of the oil at the average oil temperature are: ρ=870 kg/m^3, μ=0.004 N.s/m^2, and c_p=1959 J/kg.K, and k=0.128 W/m.K. Obtain the local Nusselt number at the end of the duct.

GIVEN

-Oil flows through a duct
-Reynolds number (Re) = 1000
-Length of duct (L) = 2.5m
-Diameter of duct (d) = 2 cm

FIND

Local Nusselt number at the end of the duct

SOLUTION:

Properties of the Oil:

$$\rho = 870\ \text{kg/m}^3 \qquad k = 0.128\ \text{W/(m}\cdot\text{K)}$$
$$\mu = 0.004\ \text{Pa.s} \qquad c_p = 1959\ \text{J/kg.K}$$

$$\text{Pr} = \frac{c_p \mu}{k} = \frac{(1959)(0.004)}{0.128} = 61.22$$

Peclet number:

$$\text{Pe} = \text{Re}\cdot\text{Pr} = (1000)(61.22) = 61220$$

$$\frac{\text{Pe}\cdot d}{L} = \frac{(61220)(0.02)}{2.5} = 489.76$$

Using the Nusselt and Graetz correlation for laminar flow and $\frac{\text{Pe}\cdot d}{L} > 10^2$; constant heat flux:

$$\text{Nu} = 1.953\left(\frac{\text{Pe}\cdot d}{L}\right)^{1/3}$$

$$\text{Nu} = (1.953)(489.76)^{1/3} = 15.39$$

Problem 3.11

In a crossflow heat exchanger, hot air at atmosphere with an average velocity of 3m/s flows across a bank of tubes in an array with $X_l = X_t$ = 5cm (see Figure 3.2) . The tube diameter is 2.5cm. The array has 20 rows in the direction of flow. The tube wall temperature is 30° C, and the average air temperature in the bundle is assumed to be 300°C. Calculate the average heat transfer coefficient and repeat the calculation if the array has 6 rows in the flow direction.

GIVEN

-Tube wall temperature (T_w) = 30^{o}C
-Average air temperature in the bundle (T_a) = 300^{o}C
-Face velocity U_∞= 3 m/s, In-tube arrangement
-$X_l = X_t$ = 0.05 m

FIND

Average heat transfer coefficient h.

SOLUTION:

From Appendix B (Table B.1), the properties of dry air at t =300 °C and atmosphere pressure are

$$\rho = 0.6157 \text{ kg/m}^3 \qquad k = 0.045 \text{ W/(m}\cdot\text{K)}$$

$$\mu = 2.95\times10^{-5} \text{ m}^2\text{/s} \qquad \Pr = 0.68$$

$$c_p = 1047 \text{ J/kg.K}$$

At wall temperature T_w = 30°C, Pr_w = 0.712
For calculating Reynolds number, we must determine the smallest flow section 1-1,
According to mass conservation equation

$$\rho U_\infty A_{face} = \rho U_0 A_{min}$$

$$\therefore U_0 = U_\infty \frac{A_{face}}{A_{min}} = 3\times\frac{X_t}{X_t - d_0} = 3\times\frac{0.05}{0.05-0.025} = 6\text{m / s}$$

Thus $Re = \dfrac{u_0 d_0 \rho_b}{\mu_b} = \dfrac{6\times0.025\times0.6157}{2.95\times10^{-5}} = 3130.68$

So we can use Equation (3.42c) to calculate Nu_b

Situation I: (n = 20)

n=20, C_n =10 (n>16)

$$\overline{Nu_b} = 0.27 C_n Re_b^{0.63} Pr_b^{0.36} (\frac{Pr_b}{Pr_w})^{0.25}$$

$$= 0.27\times1\times3130.68^{0.63}\times0.68^{0.36}\times\left(\frac{0.68}{0.712}\right)^{0.25}$$

=37.0

$$\therefore h = \frac{\kappa \overline{Nu_b}}{d_0} = \frac{0.045\times37.0}{2.5\times10^{-2}} = 66.6 \quad \text{W/m}^2\cdot\text{K}$$

Situation II: (n=6)

From Figure 3.3, it can be seen C_n=0.95

$$\overline{Nu_b} = 0.27C_n \, Re_b^{0.63} \, Pr_b^{0.36} \left(\frac{Pr_b}{Pr_w} \right)^{0.25}$$

$$= 0.27 \times 0.95 \times 3130.68^{0.63} \times 0.68^{0.36} \times \left(\frac{0.68}{0.712} \right)^{0.25}$$

=35.16

$$\therefore \quad h = \frac{k\overline{Nu_b}}{d_0} = \frac{0.045 \times 35.16}{2.5 \times 10^{-2}} = 63.3 \ \text{W/m}^2 \cdot \text{K}$$

Problem 3.12

Repeat Problem 3.11, if the heat exchanger employs a bank of staggered bare tubes with a longitudinal pitch of 4 cm and transverse pitch of 5 cm.

GIVEN
X_l = 0.04m, X_t = 0.05 m, Staggered-tubes

FIND
Average heat transfer coefficient h.

SOLUTION:

Air thermal properties are similar to Problem 3.11.
At first, we need to determine the maximum velocity across the tube-bank, which is related to the minimum area in Figure 3.2.

$$2(X_d - d_0) = 2(\sqrt{X_l^2 + (X_t/2)^2} - d_0) = 2(\sqrt{0.04^2 + 0.025^2} - 0.025) = 0.044$$

$$X_t - d_0 = 0.05 - 0.025 = 0.025$$

So,

$$X_t - d_0 < 2(X_d - d_0)$$

$$\frac{A_{\min}}{A_{face}} = \frac{X_t - d_0}{X_t} = \frac{0.05 - 0.025}{0.05} = 0.5$$

$$\therefore U_0 = U_\infty \frac{A_{face}}{A_{\min}} = 3 \times \frac{1}{0.5} = 6 \text{ m/s}$$

$$\mathrm{Re}_b = \frac{u_0 d_0 \rho}{\mu} = \frac{6 \times 0.025 \times 0.6157}{2.95 \times 10^{-5}} = 3130.68$$

For staggered bundles at Reynolds number = 3130.68, we can use Equation (3.43c) 40 calculate Nu_b

$$\overline{\mathrm{Nu}_b} = 0.35 C_n \mathrm{Re}_b^{0.6} \mathrm{Pr}_b^{0.36} \left(\frac{\mathrm{Pr}_b}{\mathrm{Pr}_w}\right)^{0.25} \left(\frac{X_t}{X_l}\right)^{0.2}$$

Situation I: (n=20)

n=20, $\because n > 16$, $\therefore C_n = 1$

$$\overline{Nu_b} = 0.35 \times 1 \times 3130.68^{0.6} \times 0.684^{0.36} \times \left(\frac{0.684}{0.712}\right)^{0.25} \times \left(\frac{0.05}{0.04}\right)^{0.2}$$

=39.55

$$\therefore \quad h = \frac{\kappa \overline{Nu_b}}{d_0} = \frac{39.55 \times 0.045}{2.5 \times 10^{-2}} = 71.2 \text{ W/m}^2 \cdot \text{K}$$

Situation II: (n=6)

n=6, $\therefore C_n = 0.95$ (From Figure 3.3)

$$\overline{Nu_b} = 0.35 \times 0.95 \times 3130.68^{0.6} \times 0.68^{0.36} \times \left(\frac{0.68}{0.712}\right)^{0.25} \times \left(\frac{0.05}{0.04}\right)^{0.2}$$

= 37.57

$$\therefore \quad h = \frac{\kappa \overline{Nu_b}}{d_0} = \frac{0.045 \times 37.57}{2.5 \times 10^{-2}} = 67.6 \ \text{W/m}^2 \cdot \text{K}$$

Problem 3.13

A shell-and-tube heat exchanger is to be used to cool 20 kg/s of water from 40°C to 20°C. The exchanger has one shell-side pass and two tube-side passes. The hot water flows through the tubes and the cooling water flows through the shell. The cooling water enters at 15°C and leaves at 25°C. The maximum permissible pressure drop is 10 kPa. The tube wall thickness is 1.25 mm and is selected as 18 Birmingham wire gauge (BWG) copper. The length of the heat exchanger is 5 m. Assume that the pressure losses at the inlet and outlet are equal to two of a velocity head ($\rho u_m^2/2$). Find the number of tubes and the proper tube diameter to expand the available pressure drop. (Hint: assume a tube diameter and average velocity inside the tubes.)

GIVEN

-A shell-and-tube heat exchanger, with hot flowing through tubes.
-Hot water inlet temperature (T_{h1}) = 40 °C
-Hot water outlet temperature (T_{h2}) = 20 °C
-Mass flow rate of hot water ($\dot{m}_h$) = 20 kg/s
-Cooling water inlet temperature (T_{c1}) = 15 °C
-Cooling water outlet temperature (T_{c2}) = 25 °C
-Maximum permissible pressure drop (Δp) = 10 kPa
-Length of heat exchanger (L) = 5 m
-Pressure losses at the inlet and outlet (Δp_{io}) = ρu_m^2
-18 BWG copper tube with wall thickness of 1.25 mm

FIND

Number of tubes and proper tube diameter to expand the available pressure drop.

SOLUTION:

Properties of water at $T_m = \dfrac{20+40}{2} = 30\ ^{o}C$:

$$\rho = 995.6\ \ kg/m^3 \qquad \mu = 7.98\times10^{-4}\ \ Pa.s$$
$$c_{p,h} = 4179\ \ J/kg.K \qquad k = 0.614\ \ W/m.K$$
$$Pr = 5.44$$

Assume that the average velocity u_m = 1.5 m/s, and try 3/4 18BWG tube with geometrical dimensions as:

3/4 18 BWG tube: I.D. = 0.652 in.; Internal area A_c = 0.3339 in.^{2};

So,

$$N_t = \frac{\dot{m}_h}{\rho u_m A_c} = \frac{20}{995.6\times1.5\times(0.000064516\times0.3339)} = 63$$

$$Re = \frac{\rho u_m d_i}{\mu} = \frac{995.6\times1.5\times(0.652\times0.0254)}{\left(7.98\times10^{-4}\right)} = 30992 \quad \text{(Turbulent)}$$

Flow regime is turbulent flow.

Using Blasius correlation in Table 3.4:

$$f = 0.0791\,Re^{-0.25} = 0.0791\times30992^{-0.25} = 0.0596$$

$$f = \frac{\tau_w}{\frac{1}{2}\rho u_m^2}$$

$$\tau_w = f\cdot\left(\frac{1}{2}\rho u_m^2\right) = 0.00596\times0.5\times995.6\times1.5^2 = 6.68\ \ N/m^2$$

$$\Delta p \cdot A_c \cdot N_t = \tau_w \cdot A_o \cdot N_t$$

$$\therefore \quad \Delta p \cdot = \frac{4\tau_w L}{d_i} = \frac{4 \times 6.68 \times 5}{0.0254 \times 0.652} = 8.067 \ \text{kPa}$$

The total pressure drop is:

$$\Delta p_t = \Delta p + \Delta p_{io} = \Delta p + \rho u_m{}^2 = 8.067 + 995.6 \times 1.5^2 = 8.067 + 2.24 = 10.307 \ \text{kPa} \ > \ 10 \ \text{kPa}$$

Since the total pressure drop is more than permissible pressure drop, so another type of tube is tried:

7/8 18 BWG tube: I.D. = 0.777 in.; Internal area A_c = 0.4742 in.^{2};

and we following the same procedure as previous to see if the pressure drop is less than permissible pressure drop.

$$N_t = \frac{\dot{m}_h}{\rho u_m A_c} = \frac{20}{995.6 \times 1.5 \times (0.000064516 \times 0.4742)} = 43$$

$$Re = \frac{\rho u_m d_i}{\mu} = \frac{995.6 \times 1.5 \times (0.777 \times 0.0254)}{\left(7.98 \times 10^{-4}\right)} = 36934 \quad \text{(Turbulent)}$$

Flow regime is turbulent flow.

Using Blasius correlation in Table 3.4:

$$f = 0.0791 Re^{-0.25} = 0.0791 \times 36934^{-0.25} = 0.0057$$

$$f = \frac{\tau_w}{\frac{1}{2}\rho u_m{}^2}$$

$$\tau_w = f \cdot \left(\frac{1}{2}\rho u_m{}^2\right) = 0.0057 \times 0.5 \times 995.6 \times 1.5^2 = 6.38 \ \text{N/m}^2$$

$$\Delta p \cdot A_c \cdot N_t = \tau_w \cdot A_o \cdot N_t$$

$$\therefore \ \Delta p \cdot = \frac{4\tau_w L}{d_i} = \frac{4 \times 6.38 \times 5}{0.0254 \times 0.777} = 6.465 \ \text{kPa}$$

The total pressure drop is:

$$\Delta p_t = \Delta p + \Delta p_{io} = \Delta p + \rho u_m{}^2 = 6.465 + 995.6 \times 1.5^2 = 6.465 + 2.24 = 8.705 \ \text{kPa} \ < 10 \ \text{kPa}$$

Problem 3.14 (*)

Repeat Problem 3.13, assuming that an overall heat transfer coefficient is give or estimated as 2000 W/m^2.K.

GIVEN
-A shell-and-tube heat exchanger, with hot flowing through tubes.
-Hot water inlet temperature (T_{h1}) = 40 ^{o}C
-Hot water outlet temperature (T_{h2}) = 20 ^{o}C
-Mass flow rate of hot water ($\dot{m}_h$) = 20 kg/s
-Cooling water inlet temperature (T_{c1}) = 15 ^{o}C
-Cooling water outlet temperature (T_{c2}) = 25 ^{o}C
-Length of heat exchanger (L) = 5 m
-18 BWG copper tube with wall thickness of 1.25 mm

FIND
Number of tubes.

SOLUTION:

Properties of water at $T_m = \frac{20+40}{2} = 30\ ^oC$:

$$\rho = 995.6\ \ kg/m^3 \qquad \mu = 7.98\times 10^{-4}\ \ Pa.s$$
$$c_{p,h} = 4179\ \ J/kg.K \qquad k = 0.614\ \ W/m.K$$
$$Pr = 5.44$$

The log temperature difference under counter current case

$$\Delta T_{lm,cf} = \frac{\Delta T_2 - \Delta T_1}{\ln\frac{\Delta T_2}{\Delta T_1}} = \frac{(40-25)-(20-15)}{\ln\frac{40-25}{20-15}} = 9.1°C$$

For one shell pass and two tube pass

$$P = \frac{\Delta T_c}{T_{h1} - T_{c1}} = \frac{10}{40-15} = 0.4$$

$$R = \frac{T_{h1}-T_{h2}}{T_{c2}-T_{c1}} = \frac{40-20}{25-15} = 2$$

From Fig. 2.7, the correction factor

F = 0.6

From energy balance

$$Q = \dot{m}_h c_{p,h}(T_{h1} - T_{h2})$$
$$= (20)(4179)(20) = 1671.6 \text{ KW}$$

The total heat transfer area

$$Q = UAF\Delta T_{lm,cf}$$

$$A = \frac{Q}{UF\Delta T_{lm,cf}} = \frac{1671.6\times 10^3}{2000\times 0.6\times 9.1} = 153.1 m^2$$

If we choose 7/8 18 BWG tube: I. D. = 0.777 in; Internal area A_c = 0.4742 in^2; the tube length l = 10m.

$$N_t = \frac{A}{\pi d_0 l} = \frac{153.1}{\pi(0.777 \times 0.0254 + 1.25 \times 10^{-3} \times 2) \times 10} = 220$$

The fluid velocity

$$u_m = \frac{\dot{m}}{N_t \rho \frac{1}{4} \pi d_0^2}$$

$$= \frac{20}{220 \times 995.6 \times 0.25 \times \pi \times (0.777 \times 0.0254)^2} = 0.30 m/s$$

$$\text{Re} = \frac{\rho u_m d_i}{\mu} = \frac{995.6 \times 0.3 \times (0.777 \times 0.0254)}{7.98 \times 10^{-4}} = 7386.6(\text{Turbulent})$$

Flow regime is turbulent flow.

Using Blasius correlation in Table 3.4:

$$f = 0.0791 \text{Re}^{-0.25} = 0.0791 \times 7386.6^{-0.25} = 0.0085$$

$$f = \frac{\tau_w}{\frac{1}{2} \rho u_m^2}$$

$$\tau_w = f \cdot (\frac{1}{2} \rho u_m^2) = 0.0085 \times 0.5 \times 995.6 \times 0.3^2 = 0.38 N/m^2$$

$$\Delta p \cdot A_c \cdot N_t = \tau_w \cdot A_0 \cdot N_t$$

$$\therefore \Delta p = \frac{4 \tau_w L}{d_i} = \frac{4 \times 0.38 \times 5}{0.0254 \times 0.777} = 0.39 kPa$$

The total pressure drop is:

$$\Delta p_t = \Delta p + \Delta p_{io} = \Delta p + \rho u_m^2 = 0.39 + 995.6 \times 0.3^2 \times 10^{-3}$$

$$= 0.39 + 0.09 = 0.48 kPa < 10 kPa$$

The pressure drop is smaller than permissible pressure drop, so it can satisfy the design requirements.

Problem 3.15

Calculate the average heat transfer coefficient for water flowing at 1.5 m/s with an average temperature of 20°C in a long 2.5-cm I.D. pipe by four different correlation from Table 3.6 (No. 2,3,4, and 6) considering the effect of temperature-dependent properties. The inside wall temperature of the tube is 70°C.

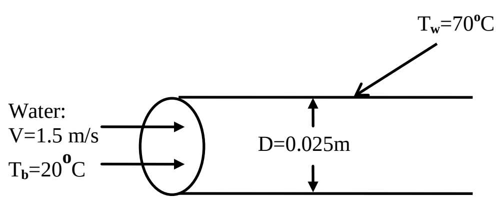
GIVEN

-Water flowing through the pipe
-Water temperature (T_b) =20°C
-Water velocity (V) = 4 m/s
-Inside diameter of pipe (D) =2.5cm = 0.025m
-Pipe surface temperature (T_s)=70°C

FIND

The average heat transfer coefficient ($\bar{h}_c$) by 4 different correlation.

SKETCH



ASSUMPTIONS

Steady state.
Fully developed flow L/D>60.
Uniform and constant wall surface temperature.
Pipe wall is smooth.

PROPERTIES AND CONSTANTS

Properties of water at 20°C:

$\rho = 998\ \text{kg/m}^3$ $\qquad$ $k = 0.599\ \text{W/m}\cdot\text{K}$

$\mu = 1.007\times10^{-3}\ \text{kg/m}\cdot\text{s}$ $\qquad$ $\text{Pr} = 7.05$

$\nu = 1.009\times10^{-6}\ \text{m}^2/\text{s}$

$\mu_w = \mu_{\text{Water at }70^\circ\text{C}} = 4.04\times10^{-4}\ \text{Pa.s}$

$\text{Pr}_w = \text{Pr}_{\text{Water at }70^\circ\text{C}} = 2.57$

<u>SOLUTION:</u>

$$\text{Re} = \frac{\rho\cdot u_m\cdot d_i}{\mu} = \frac{998\times1.5\times0.025}{1.007\times10^{-3}} = 37165 \quad \text{(Turbulent)}$$

The heat transfer coefficient is given by:

$$h_i = \frac{Nu_b \cdot k}{d_i}$$

Where the Nusselt number can be estimated using the turbulent forced convection correlation in circular ducts for liquids with variable properties:

a. Seider & Tate:

$$Nu_b = 0.023 Re^{0.8} Pr_b^{1/3} \left(\frac{\mu_b}{\mu_w} \right)^{0.14}$$

$$Nu_b = 0.023 \times 37165^{0.8} \times 7.05^{1/3} \left(\frac{1.007}{0.404} \right)^{0.14}$$

$$Nu_b = 227$$

$$\Rightarrow h_i = \frac{227 \times 0.599}{0.025} = 5439 \ \ W/m^2 \cdot K$$

b. Petuknov & Kirilov:

$$Nu_b = \frac{(f/8) Re_b Pr_b}{1.07 + 12.7\sqrt{\frac{f}{8}}\left(Pr_b^{2/3} - 1\right)} \left(\frac{\mu_b}{\mu_w} \right)^n$$

Heating $\Rightarrow$ n = 0.11

$$f = (1.82 \log Re_b - 1.64)^{-2}$$

$$f = (1.82 \log 37165 - 1.64)^{-2} = 0.0224$$

$$Nu_b = \frac{(0.0224/8) \times 37165 \times 7.05}{1.07 + 12.7\sqrt{\frac{0.0224}{8}}\left(7.01^{2/3} - 1\right)} \left(\frac{1.007}{0.404} \right)^{0.11}$$

$$Nu_b = 290$$

$$h_i = \frac{290 \times 0.599}{0.025} = 6948 \ W/m^2 \cdot K$$

c. Hufschmidt et al:

$$Nu_b = \frac{(f/8) Re_b Pr_b}{1.07 + 12.7\sqrt{\frac{f}{8}}\left(Pr_b^{2/3} - 1\right)} \left(\frac{Pr_b}{Pr_w} \right)^{0.11}$$

$$f = (1.82 \log Re_b - 1.64)^{-2}$$

$$f = (1.82 \log 37165 - 1.64)^{-2} = 0.0224$$

$$Nu_b = \frac{(0.0224/8) \times 37165 \times 7.05}{1.07 + 12.7\sqrt{\frac{0.0224}{8}}\left(7.05^{2/3} - 1\right)} \left(\frac{7.05}{2.57} \right)^{0.11}$$

$$Nu_b = 294$$

$$h_i = \frac{294 \times 0.599}{0.025} = 7044 \ W/m^2 \cdot K$$

d. Oskay & Kakac:

$$Nu_b = 0.023\,Re^{0.8}\;Pr_b^{\;0.4}\left(\frac{\mu_b}{\mu_w}\right)^{0.262}$$

$$Nu_b = 0.023 \times 37165^{0.8} \times 7.05^{0.4} \times \left(\frac{1.007}{0.404}\right)^{0.262}$$

$$Nu_b = 289$$

$$h_i = \frac{289 \times 0.599}{0.025} = 6924\ W/m^2 \cdot K$$

The values obtained for the heat transfer coefficient differed by less than 10%, when the correlation of Petukhov & Kirilov, Hufschmidt et al., and Oskay & Kakac where used which is very much expected. while the value obtained using sieder & Tate's differed by 22%.

Problem 3.16

A double pipe heat exchanger is used to condense steam at 40°C saturation temperature. Water at average bulk temperature of 20°C flows at 2m/s through the inner pipe (copper, 2.54 cm I.D., 3.05 cm O.D.). Steam at its saturation temperature flows in the annulus formed between the outer surface of the inner pipe and the outer pipe of 6-cm I.D. The average heat transfer coefficient of the condensing steam is 6000 w/m^{2}K, and the thermal resistance of a surface scale on the outer surface of the copper pipe is 0.000176 m^{2}K/W.

a. Determine the overall heat transfer coefficient between the steam and the water based on the outer area of the copper pipe.

b. Evaluate the temperature at the inner surface of the pipe.

c. Estimate the length required to condense 0.5 kg/s of steam.

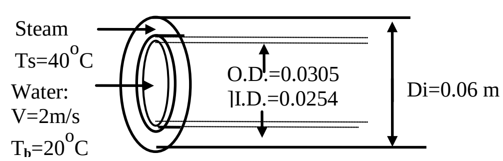
GIVEN

-Double pipe heat exchanger. Steam in annulus, and water in inner pipe.
-Steam temperature (Ts) =40°C
-Average bulk water temperature (T_b) =20°C
-Water velocity (V) = 2 m/s
-Inner pipe diameter: (I.D.) =0.0254m; (O.D.) = 0.0305m.
-outer pipe diameter: (I.D.)= 0.06 m.
-Heat transfer coefficient in annulus side (h_o) = 6000 W/m^{2}K.
-Thermal resistance caused by fouling (R_{fo}) = 1.76×10^{-4} m^{2}K/W.

FIND

a. Overall heat transfer coefficient (U).
b. Temperature of the inner surface of the copper pipe (T_{wi}).
c. The length (L) required to condense 0.5 kg/s of steam.

SKETCH



ASSUMPTIONS

Steady state.
Fully developed flow.
Constant steam temperature.

PROPERTIES AND CONSTANTS

Properties of water at 20°C:

$\rho = 998 \text{ kg/m}^3$ $\quad$ $k = 0.599 \text{ W/m}\cdot\text{K}$

$\mu = 1.007\times10^{-3} \text{ kg/m}\cdot\text{s}$ $\quad$ $Pr = 7.05$

$\nu = 1.009\times10^{-6} \text{ m}^2/\text{s}$

Properties of steam 40°C:

$$h_{fg} = 2573.45 - 167.5 = 2406 \text{ kJ/kg}$$

SOLUTION:

a.

$$Re = \frac{\rho\cdot u_m\cdot d_i}{\mu} = \frac{998\times2.0\times0.0254}{1.007\times10^{-3}} = 50346 \quad \text{(Turbulent)}$$

The overall heat transfer coefficient based on the pipes outside surface area can be expressed by:

$$U_oA_o = \frac{1}{\dfrac{1}{h_iA_i}+\dfrac{\ln(r_o/r_i)}{2\pi k_cL}+\dfrac{R_{fo}}{A_o}+\dfrac{1}{h_oA_o}}$$

$$h_i = \frac{Nu_bk}{d_i}$$

and from Table 3.3, Correlation #1:

$$Nu_b = \frac{(f/2)\,Re_b\,Pr_b}{1.+8.7\left(\dfrac{f}{2}\right)^{1/2}(Pr_b-1)}$$

$$f = (1.58\ln Re_b - 3.28)^{-2}$$

$$f = (1.58\ln 50346 - 3.28)^{-2}$$

$$f = 0.00523$$

$$Nu_b = \frac{(0.00523/2)\times50346\times7.05}{1+8.7\times(0.00523/2)^{1/2}\times(7.05-1)}$$

$$Nu_b = 251.5$$

$$\Rightarrow h_i = \frac{251.5\times0.599}{0.0254} = 5931 \text{ W/m}^2\text{K}$$

Thermal Conductivity of copper at 40°C: K_c=383 W/mK

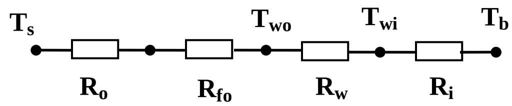
$$U_o = \frac{1}{\dfrac{A_o}{h_iA_i}+\dfrac{A_o\ln(r_o/r_i)}{2\pi k_cL}+R_{fo}+\dfrac{1}{h_o}}$$

$$U_o = \frac{1}{\dfrac{d_o}{h_id_i}+\dfrac{d_o\ln(r_o/r_i)}{2k_c}+R_{fo}+\dfrac{1}{h_o}}$$

$$U_o = \frac{1}{\dfrac{0.0305}{0.0254 \times 5931} + \dfrac{0.0305\ln\left(0.1525/0.0127\right)}{2 \times 383} + 0.000176 + \dfrac{1}{6000}}$$

$$U_o = 1827\ \mathrm{W/m^2K}$$

The thermal circuit can be sketched as follows:



R_o = Thermal resistance of steam [=1/($A_o h_o$)].
Ts = Temperature of the steam.
R_{fo} = Thermal resistance of the surface scale.
T_{wo} = Tube outside wall temperature.
R_{kt} = Thermal resistance of the copper tube.
T_{wi} = Tube inside wall temperature.
R_i = Water thermal resistance [=1/($A_i h_i$)].
T_b = Water bulk temperature.

b.

From the above circuit:

$$\frac{T_s - T_b}{R_o + \dfrac{R_{fo}}{A_o} + R_w + R_i} = \frac{T_{wi} - T_b}{R_i}$$

$$\Rightarrow T_{wi} = R_i\left(\frac{T_s - T_b}{R_o + \dfrac{R_{fo}}{A_o} + R_w + R_i}\right) + T_b$$

$$T_{wi} = \left(\frac{1}{5931}\right)\left(\frac{40-20}{\dfrac{0.0254}{0.0305}\times\dfrac{1}{6000} + \dfrac{0.0254}{0.0305}\times 0.000176 + \dfrac{0.0254\times\ln\left(0.01525/0.0127\right)}{2\cdot 383} + \dfrac{1}{5931}}\right) + 20$$

$$T_{wi} = 21.4\ ^{o}\mathrm{C}$$

c.

The heat released by the steam condensing at 40^{o}C is:

$$Q = m_s \cdot h_{fg} = 0.5 \times 2405$$
$$= 1203\ \ \mathrm{kJ/s}$$
$$Q = A_o U_o (T_s - T_b) = 2\pi \cdot r_o L U_o (T_s - T_b)$$
$$\Rightarrow L = \frac{Q}{2\pi \cdot r_o U_o (T_s - T_b)} = \frac{1203000}{2\pi \times 0.01525 \times 1827 \times (40-20)} = 344\ \mathrm{m}$$

Problem 3.17

Carbon dioxide gas at 1 atm pressure is to be heated fro 30 to 75 ^oC by pumping it through a tube bank at a velocity of 4 m/s. The tubes are heated by steam condensing within them at 200^{o}C. The tubes have a 2.5-cm O.D., are in an in-line arrangement, and have a longitudinal spacing of 4 cm and a transverse spacing of 4.5 cm. If 15 tube rows are required, what is the average heat transfer coefficient?

GIVEN

-Carbon dioxide being pumping through tube banks, and being heated by condensing steam.
-Co_2 inlet temperature (T_{in}) = 30 ^oC
-Co_2 outlet temperature (T_{out}) = 75 ^oC
-Co_2 velocity (u_m) = 4 m/s
-Steam temperature (T_s) = 200 ^oC
-Tube O.D. (d_o) = 2.5 cm
-X_l = 4 cm
-X_t = 4.5 cm
-Number of tube row (n) = 15

FIND

Average heat transfer coefficient h.

SOLUTION:

From Appendix B (Table B.1), the properties of Co_2 at $T_{average} = \frac{30+75}{2} = 52.5$ °C and atmosphere pressure are:

$$\rho = 1.641 \text{ kg/m}^3 \qquad k = 0.0185 \text{ W/(m}\cdot\text{K)}$$
$$\mu = 1.63\times10^{-5} \text{ m}^2\text{/s} \qquad Pr = 0.77$$
$$c_p = 876 \text{ J/kg.K}$$

Approximate wall temperature $T_w = \frac{52.5+200}{2} = 126.25$°C, then $Pr_w = 0.77$

For calculating Reynolds number, we must determine the smallest flow section,
According to mass conservation equation

$$\rho u_m A_{face} = \rho_0 u_0 A_{min}$$

$$\therefore u_0 = u_m \frac{A_{face}}{A_{min}} = 4\times\frac{X_t}{X_t - d_o} = 4\times\frac{0.045}{0.045-0.025} = 9 \text{ m/s}$$

Thus

$$Re = \frac{u_0 d_o \rho_b}{\mu_b} = \frac{9\times0.025\times1.641}{1.63\times10^{-5}} = 22651$$

n=15, $C_n \approx 1$.

So we can use Equation (3.42c) to calculate Nu_b:

$$Nu_b = 0.27C_n Re_b^{0.63} Pr_b^{0.36}\left(\frac{Pr_b}{Pr_w}\right)^{0.25}$$

$$= 0.27\times1\times22651^{0.63}\times0.77^{0.36}\left(\frac{0.77}{0.77}\right)^{0.25} = 136.2$$

$$h = \frac{Nu\cdot k}{d_o} = \frac{136.2\times0.0185}{0.025} = 100.8 \text{ W/m}^2\text{.K}$$

Problem 4.1

Consider the flow of 20°C water through a circular duct with an I.D. of 2.54 cm. The average velocity of the water is 4 m/s. Calculate the pressure drop per unit length (Δp/L).

GIVEN

- -Flow of water through circular duct
- -Water temperature (T) = 20°C
- -Water velocity (u) = 4 m/s
- -Inner diameter of duct (d_i) = 2.54 cm

FIND

The pressure drop per unit length Δp/L

SOLUTION:

Properties of water at T = 20°C = 293 K:

$\rho = 998 \text{ kg/m}^3 \qquad \mu = 10.07 \times 10^{-4} \text{ Pa.s}$

Reynolds number:

$$Re = \frac{\rho u d_i}{\mu} = \frac{998 \times 4 \times 0.0254}{10.07 \times 10^{-4}} = 100{,}692 \text{ (Turbulent)}$$

The friction factor can be obtained by Karman-Nikuradse correlation in Table 4.1:

$$f = 0.046 Re^{-0.2} \quad \text{for } 3 \times 10^4 < Re < 10^6$$

$$f = 0.046 \times (100{,}692)^{-0.2}$$

$$= 0.0046$$

Pressure drop per unit length:

$$\frac{\Delta p}{L} = 4f \frac{1}{d_i} \frac{\rho u^2}{2}$$

$$= 4 \times 0.0046 \times \frac{1}{0.0254} \times \frac{998 \times 4^2}{2}$$

$$= 5783.7 \text{ Pa/m}$$

Problem 4.2

Water at 5°C flows through a parallel-plate channel in a flat-plate heat exchanger. The spacing between the plates is 2 cm, and the mean velocity of water is 3.5 m/s. Calculate the pressure drop per unit length in the hydrodynamically fully developed region.

GIVEN

- -Flow of water through parallel-plate channel
- -Water temperature (T) = 5°C
- -Water velocity (u) = 3.5 m/s
- -Spacing between the plates is (B) = 2 cm

FIND

The pressure drop per unit length Δp/L

SOLUTION:

Properties of water at T = 5°C = 278 K:

$$\rho = 999 \text{ kg/m}^3 \qquad \mu = 15.21\times10^{-4} \text{ Pa.s}$$

The hydraulic diameter D_h is:

$$D_h = 0.02 \text{ m}$$

Reynolds number:

$$Re = \frac{\rho u D_h}{\mu} = \frac{999\times3.5\times0.02}{15.21\times10^{-4}} = 45{,}976 \text{ (Turbulent)}$$

The friction factor can be obtained by Blasius correlation in Table 4.1:

$$f = 0.0791 Re^{-0.25} \quad \text{for } 4\times10^3 < Re < 10^5$$

$$f = 0.0791\times(45{,}976)^{-0.25}$$

$$= 0.0054$$

Pressure drop per unit length:

$$\frac{\Delta p}{L} = 4f\frac{1}{D_h}\frac{\rho u^2}{2}$$

$$= 4\times0.0054\times\frac{1}{0.02}\times\frac{999\times3.5^2}{2}$$

$$= 6608.4 \text{ Pa/m}$$

Problem 4.3

It is proposed that the stationary diesel exhaust gas (CO_2) be used for heating air by the use of a double pipe heat exchanger. CO_2 flows through the annulus. From the measurements of the velocity, it is calculated that the mass flow rate of gases from the engine is 100 kg/h. The exhaust gas temperature is 600 K. The air is available at 20°C and it will be heated to 80°C at a mass flow rate of 90 kg/h. One standard (schedule 40) tube size of 4×3 in. double pipe exchanger (hairpin) of 2 m length (copper tubes) will be used.
a. Calculate the frictional pressure drop in the inner tube.
b. Calculate the pressure drop (Pa) in the annulus.
c. Calculate the pumping power for both streams.
Assume the average temperature of CO_2 at 500 K and the following properties:

$$\rho = 1.0732 \ \ kg/m^3, \quad c_p = 1013 \ \ J/kg.K, \ \ k = 0.03352 \ \ W/m.K, \ \ Pr = 0.702, \ \ \nu = 21.67 \times 10^{-6} \ \ m^2/s$$

For air: c_p = 1007 J/kg.K

GIVEN

-Double heat exchanger, with CO_2 flowing through the annulus and air through the inner tube
-Inlet CO_2 temperature (T_{h1}) = 600 K
-Mass flow rate of CO_2 ($\dot{m}_h$) = 100 kg/h = 0.0278 kg/s
-Inlet air temperature (T_{c1}) = 20°C
-Outlet air temperature (T_{c2}) = 80°C
-Mass flow rate of air ($\dot{m}_c$) = 90 kg/h = 0.025 kg/s
-4×3 double pipe exchanger (hairpin)
-length of hairpin (L) = 2 m

FIND

(1) Frictional pressure drop in the inner tube Δp_t;
(2) Pressure drop in the annulus Δp_a
(3) Pumping power P_t and P_a

SOLUTION:

Nominal pipe size of 3 in.	Nominal pipe size of 4 in.
d_o = 3.5 in. = 0.0889 m d_i = 3.068 in. = 0.0779 m	D_o = 4.5 in. = 0.1143 m D_i = 4.026 in. =0.1023 m

Properties of CO_2 at T = 500 K:

$$\rho = 1.0732 \ \ kg/m^3 \qquad \mu = 2.32 \times 10^{-5} \ \ Pa.s$$
$$c_p = 1013 \ \ J/kg.K \qquad \nu = 21.67 \times 10^{-6} \ \ m^2/s$$
$$k = 0.03352 \ \ W/m.K \qquad Pr = 0.702$$

Properties of air at T = 323 K: ($\dfrac{20+80}{2} = 50°C = 323\ K$)

$$\rho = 1.0931 \ \ kg/m^3 \qquad \mu = 1.96 \times 10^{-5} \ \ Pa.s$$
$$c_p = 1007 \ \ J/kg.K \qquad k = 0.0278 \ \ W/m.K$$
$$Pr = 0.71$$

a.

Reynolds number:

$$\mathrm{Re} = \frac{\rho u d_i}{\mu} = \frac{\dot{m}_c d_i}{\mu A_c} = \frac{0.025 \times 0.0779}{1.96 \times 10^{-5}\left(\frac{\pi}{4} \times 0.0779^2\right)} = 20848 \text{ (Turbulent)}$$

Using Eq. (4.7) for friction factor:

$$f = 0.079\mathrm{Re}^{-0.25} \qquad \text{for } 4 \times 10^3 < \mathrm{Re} < 10^5$$

$$f = 0.079 \times 21063^{-0.25}$$

$$= 0.00658$$

Pressure drop is:

$$\Delta p = 4f \frac{2L}{d_i} \frac{\rho u^2}{2}$$

where velocity is:

$$u = \frac{\dot{m}_c}{\rho A_c} = \frac{4\dot{m}_c}{\rho \pi d_i^2} = \frac{4 \times 0.025}{1.0931 \times \pi \times 0.0779^2} = 4.8 \quad \text{m/s}$$

$$\therefore \Delta p_t = 4 \times 0.00658 \times \frac{4}{0.0779} \frac{1.0931 \times 4.8^2}{2} = 17.01 \quad \text{Pa}$$

b.

The hydraulic diameter D_h is:

$$D_h = D_i - d_o = 0.1023 - 0.0889 = 0.0134 \text{ m}$$

The cross-sectional area of annulus is:

$$A_c = \frac{\pi\left(D_i^2 - d_o^2\right)}{4} = \frac{\pi \times \left(0.1023^2 - 0.0889^2\right)}{4} = 0.002 \text{ m}^2$$

Reynolds number:

$$\mathrm{Re} = \frac{\rho u D_h}{\mu} = \frac{\dot{m}_c D_h}{\mu A_c} = \frac{0.0278 \times 0.0134}{2.32 \times 10^{-5}(0.001996)} = 8028 \text{ (Transition flow)}$$

Using Eq. (4.7) for friction factor:

$$f = 0.079\mathrm{Re}^{-0.25} \qquad \text{for } 4 \times 10^3 < \mathrm{Re} < 10^5$$

$$f = 0.079 \times 8028^{-0.25}$$

$$= 0.00836$$

Pressure drop is:

$$\Delta p = 4f \frac{L}{D_h} \frac{\rho u^2}{2}$$

where velocity is:

$$u = \frac{\dot{m}_c}{\rho A_c} = \frac{0.0278}{1.0732 \times 0.002} = 12.95 \quad \text{m/s}$$

$$\therefore \Delta p_a = 4 \times 0.00836 \times \frac{2}{0.0134} \frac{1.0732 \times 12.95^2}{2} = 449.14 \quad \text{Pa}$$

c.

$$P = \frac{\dot{m}\Delta p}{\rho \eta_p}$$

$$P_t = \frac{0.025 \times 17.01}{1.0931 \times 0.8} = 0.49 \quad W$$ per hairpin.

$$P_a = \frac{0.0278 \times 449.14}{1.0732 \times 0.8} = 14.54 \quad W$$ per hairpin.

Problem 4.4

Calculate the pumping power for both streams given in Problem 3.8.

GIVEN

-Double pipe heat exchanger. City water in annulus, and hot water in inner tube.
-Mass flow rate of city water ($\dot{m}_c$) = 0.5 kg/s
-City water inlet temperature (T_{c1}) = 20°C
-City water outlet temperature (T_{c2}) = 35°C
-Hot water inlet temperature (T_{h1}) = 140°C
-Hot water temperature drop (ΔT_h) =15°C
-Hairpin: L = 4.5 m, 3 in.× 2 in. (schedule 40)

FIND

Pumping power P_t and P_a

SOLUTION:

Nominal pipe size of 3 in.	Nominal pipe size of 2 in.
d_o = 3.5 in. = 0.0889 m d_i = 3.068 in. = 0.0779 m	D_o = 2.375 in. = 0.0603 m D_i = 2.067 in. = 0.0525 m

Properties of hot and city water:

Bulk mean temperatures:

Cold city water: $T_{b,c} = \dfrac{T_{c1}+T_{c2}}{2} = \dfrac{20+35}{2} = 27.5^o\ C$

Hot water: $T_{h,c} = \dfrac{T_{h1}+T_{h2}}{2} = \dfrac{140+(140-15)}{2} = 132.5^o\ C$

Annulus - cold city water at 27.5°C	Inner tube - hot water at 132.5°C
ρ_c = 996 kg/m^3 k = 0.61 W/(m.K) $\mu = 0.843\times10^{-3}$ Pa.s $c_{p,c}$=4.179 kJ/(kg.K) Pr = 5.79	ρ_h = 932 kg/m^3 k = 0.688 W/(m.K) $\mu = 0.207\times10^{-3}$ Pa.s $c_{p,h}$=4.269 kJ/(kg.K) Pr = 1.28

Heat balance:

$$\dot{m}_h c_{p,h}\left(T_{h1}-T_{h2}\right) = \dot{m}_c c_{p,c}\left(T_{c2}-T_{c1}\right)$$

$$\dot{m}_h = \frac{\dot{m}_c c_{p,c}\left(T_{c2}-T_{c1}\right)}{c_{p,h}\left(T_{h1}-T_{h2}\right)} = \frac{0.5\times4.179\times(35-20)}{4.269\times15} = 0.49\ \ kg/s$$

The overall heat transfer coefficient U_o (h_i and h_o are obtained in Problem 3.8, and we neglect the wall resistance here):

$$U_o = \frac{1}{\frac{1}{h_i}\frac{d_o}{d_i} + \frac{1}{h_o}} = \frac{1}{\frac{1}{2151.8} \times \frac{0.0603}{0.0525} + \frac{1}{626.65}} = 469.6 \ \ W/m^2.K$$

$$\Delta T_{lm} = \Delta T_1 = \Delta T_2 = 105°C$$

$$Q = U_o(\pi d_o L) N_t \Delta T_{lm} = \dot{m}_c c_{p,c}(T_{c2} - T_{c1})$$

$$N_t = \frac{\dot{m}_c c_{p,c}(T_{c2} - T_{c1})}{U_o(\pi d_o L)\Delta T_{lm}} = \frac{0.5 \times 4179 \times (35-20)}{469.6 \times \pi \times 0.0603 \times 4.5 \times 105} = 0.75 \approx 1$$

∴ 1 hairpin is needed.

* Pressure drop in annulus (cold water):

Re = 5469 (obtained from Problem 3.8)

using Karmen-Nikuradse correlation in table 4.1:

$f = 0.046\ Re^{-0.2} = 0.046 \times 5469^{-0.2} = 0.0082$

$$\Delta p = 4f\,\frac{L(N_t/2)}{D_h}\frac{\rho u^2}{2} = 4 \times 0.0082 \times \frac{4.5 \times 0.5}{0.0176} \times \left(\frac{1}{2} \times 996 \times 0.263^2\right)$$

$$= 145 \ \ Pa$$

The pumping power for annulus flow (assume pump efficiency as 0.8):

$$P_a = \frac{\dot{m}\Delta p}{\rho_c \eta_p} = \frac{0.5 \times 145}{996 \times 0.8} = 0.09 \ \ W$$

* Pressure drop in inner tube (hot water):

Re = 57298 (obtained from Problem 3.8)

using Dew, Koo and McAdams correlation in table 4.1:

$f = 0.0014 + 0.125\ Re^{-0.32} \quad (4 \times 10^3 < Re < 5 \times 10^6)$

$= 0.0014 + 0.125 \times 57298^{-0.32} = 0.0052$

$$\Delta p = 4f\,\frac{LN_t}{d_i}\frac{\rho u^2}{2} = 4 \times 0.0052 \times \frac{4.5 \times 1}{0.0525} \times \left(\frac{1}{2} \times 932 \times 0.2424^2\right)$$

$$= 48.8 \ \ Pa$$

The pumping power for annulus flow (assume pump efficiency as 0.8):

$$P_t = \frac{\dot{m}\Delta p}{\rho_c \eta_p} = \frac{0.49 \times 48.8}{932 \times 0.8} = 0.032 \ \ W$$

Problem 4.5

Consider a shell-and-tube heat exchanger. Air with a flow rate of 1.5 kg/s at 500°C and at atmospheric pressure flows through 200 parallel tubes. Each tube has an internal diameter of 2 cm and length of 4 m. Assume that the D/D_0 = 0.5 data can be used for the contraction and enlargement. The outlet temperature of the cooled air is 100°C. Calculate the pressure drop for:
a. The abrupt contraction
b. Friction
c. Acceleration
d. Enlargement
and compare the total pressure drop with the frictional pressure drop.

GIVEN

-A shell-and-tube heat exchanger, with air flowing through the inner tubes.
-Inlet air temperature (T_{h1}) = 500°C
-Outlet air temperature (T_{h2}) = 100°C
-Mass flow rate of Air ($\dot{m}$) = 1.5 kg/s
-The number of tubes (N_t) = 200
-The cross-sectional area of inlet tube plate is 40% larger than that of total number of tubes.
-Internal diameter of tubes (d_i) = 2 cm
-length of tubes (L) = 4 m

FIND

The pressure drop for:
a. the abrupt contraction;
b. friction;
c. acceleration;
d. enlargement,
and compare the total pressure drop with the frictional pressure drop.

SOLUTION:

Properties of air at T = 500°C: $\rho_i = 0.4564 \ \text{kg/m}^3$

Properties of air at T = 100°C: $\rho_o = 0.9458 \ \text{kg/m}^3$

Properties of air at 300°C:

$$\rho_m = 0.6157 \ \text{kg/m}^3 \qquad \mu = 2.95 \times 10^{-5} \ \text{Pa.s}$$

a.

Mean velocity:

$$u_m = \frac{\dot{m}}{\rho_m A_c N_t} = \frac{4\dot{m}}{\rho_m \pi d_i \cdot N_t} = \frac{4 \times 1.5}{0.6157 \times \pi \times 0.02^2 \times 200} = 38.8 \ \text{m/s}$$

Dynamic head:

$$\frac{1}{2}\rho_m u_m^2 = \frac{1}{2} \times 0.6157 \times 38.8^2 = 463.45 \ \text{Pa}$$

Abrupt contraction: ($k_c = 0.4$ at D/D_0 = 0.5 in Table 4.3)

$$\Delta p_c = k_c \frac{1}{2}\rho u_m^{\ 2} = 0.4 \times 463.45 = 185.38 \text{ Pa}$$

b.

Reynolds number:

$$Re = \frac{\rho_m u_m d_i}{\mu} = \frac{0.6157 \times 38.8 \times 0.02}{2.95 \times 10^{-5}} = 16{,}196 \text{ (Turbulent)}$$

Using Eq. (4.7) for friction factor:

$$f = 0.079 Re^{-0.25} \qquad \text{for } 4 \times 10^3 < Re < 10^5$$

$$f = 0.079 \times 16196^{-0.25}$$

$$= 0.007$$

Frictional pressure drop:

$$\Delta p_f = 4f \frac{L}{d_i} \frac{\rho_m u_m^{\ 2}}{2}$$

$$\Delta p_f = 4 \times 0.007 \times \frac{4}{0.02} \times 463.45 = 2.6 \quad \text{kPa}$$

c.

Acceleration pressure drop:

$$\Delta p_a = G^2 \left(\frac{1}{\rho_o} - \frac{1}{\rho_i} \right)$$

$$G = \frac{\dot{m}}{N_t A_c} = \frac{4\dot{m}}{N_t \pi d_i^{\ 2}} = \frac{4 \times 1.5}{200 \cdot \pi \cdot 0.02^2} = 23.87 \quad \text{kg/m}^2\text{.s}$$

$$\therefore \Delta p_a = 23.84^2 \times \left(\frac{1}{0.9458} - \frac{1}{0.4564} \right) = -646.2 \quad \text{Pa}$$

d.

Enlargement pressure drop: ($k_e = 0.37$ from Kays and London [26])

$$\Delta p_e = k_e \frac{1}{2}\rho u_m^{\ 2} = 0.37 \times 463.45 = 171.48 Pa$$

The total pressure drop

$$\Delta p_t = \Delta p_c + \Delta p_e + \Delta p_a + \Delta p_f$$

$$= 115.86 - 171.48 - 646.2 + 2600$$

$$= 1898.2 Pa < 2.6 kPa$$

Problem 4.6

The core of a shell-and-tube heat exchanger contains 60 tubes (single-tube pass) with an inside diameter of 2.5 cm and length of 2 m. The shell inside diameter is 35 cm. The air, which flows through the tubes at a flow rate of 1.5 kg/s, is to be heated from 100°C to 300°C.

a. Calculate the total pressure drop as the sum of the frictional pressure drop in the tube and the pressure drop due to acceleration, abrupt contraction (kc=0.25), and abrupt enlargement (ke=0.37). Is the total pressure drop smaller than the frictional pressure drop in the tube? Why?

b. Calculate the fan power needed.

GIVEN

- -A shell-and-tube heat exchanger
- -Inlet air temperature (T_{c1})= 100°C
- -Outlet air temperature (T_{c2}) = 300°C
- -Mass flow rate of CO_2 ($\dot{m}$) = 1.5 kg/s
- -The number of tubes (N_t) = 65
- -Internal diameter of tubes (d_i) = 2.5 cm
- -Shell inside diameter (D_i) = 35 cm
- -length of tubes (L) = 2 m

FIND

a. The pressure drop for:
- i. the abrupt contraction;
- ii. friction;
- iii. acceleration;
- iv. enlargement,
- v. and compare the total pressure drop with the fricitonal pressure drop.

b. the fan power needed

SOLUTION:

Properties of air at T = 300°C: $\rho_i = 0.6088 \text{ kg/m}^3$

Properties of air at T = 100°C: $\rho_o = 0.9380 \text{ kg/m}^3$

Properties of air at at 200°C:

$$\rho_m = 0.7383 \text{ kg/m}^3 \qquad \mu = 2.59\times10^{-5} \text{ Pa.s}$$

a.

i.

Velocity:

$$u_m = \frac{\dot{m}}{\rho_m A_c N_t} = \frac{4\dot{m}}{\rho_m \pi d_i \cdot N_t} = \frac{4\times1.5}{0.7383\times\pi\times0.025^2\times60} = 68.98 \text{ m/s}$$

Dynamic head:

$$\frac{1}{2}\rho_m u_m^2 = \frac{1}{2}\times0.7383\times68.98^2 = 1756.5 \text{ Pa}$$

Abrupt contraction:

$$\Delta p_c = k_c \frac{1}{2}\rho u_m^2 = 0.25 \times 1756.5 = 439.12 \text{Pa}$$

ii.

Frictional pressure drop:

$$\Delta p_f = 4f \frac{L}{d_i} \frac{\rho_m u_m^2}{2}$$

Renolds number:

$$\text{Re} = \frac{\rho_m u_m d_i}{\mu} = \frac{0.7383 \times 68.98 \times 0.025}{2.59 \times 10^{-5}} = 49{,}044.6 \text{ (Turbulent)}$$

Using Eq. (4.7) for friction factor:

$$f = 0.079 \text{Re}^{-0.25} \qquad \text{for } 4 \times 10^3 < \text{Re} < 10^5$$

$$f = 0.079 \times 49044.6^{-0.25}$$

$$= 0.0053$$

Frictional pressure drop is:

$$\Delta p_f = 4 \times 0.0053 \times \frac{2}{0.025} \times 1756.5 = 2981.27 \quad \text{Pa}$$

iii.

Acceleration pressure drop:

$$\Delta p_a = G^2 \left(\frac{1}{\rho_o} - \frac{1}{\rho_i} \right)$$

$$G = \frac{\dot{m}}{N_t A_c} = \frac{4\dot{m}}{N_t \pi d_i^2} = \frac{4 \times 1.5}{60 \cdot \pi \cdot 0.025^2} = 50.96 \text{ kg/m}^2\text{.s}$$

$$\therefore \Delta p_a = 50.96^2 \times \left(\frac{1}{0.6088} - \frac{1}{0.9380} \right) = 1497.1 \quad \text{Pa}$$

iv.

Enlargement pressure drop:

$$\Delta p_e = k_e \frac{1}{2}\rho u_m^2 = 0.37 \times 1756.5 = 649.91 Pa$$

v.

The total pressure drop

$$\Delta p_t = \Delta p_c + \Delta p_e + \Delta p_a + \Delta p_f$$

$$= 2981.27 + 439.12 - 649.91 + 1497.1$$

$$= 4267.6 Pa > 2981.27 Pa$$

b.

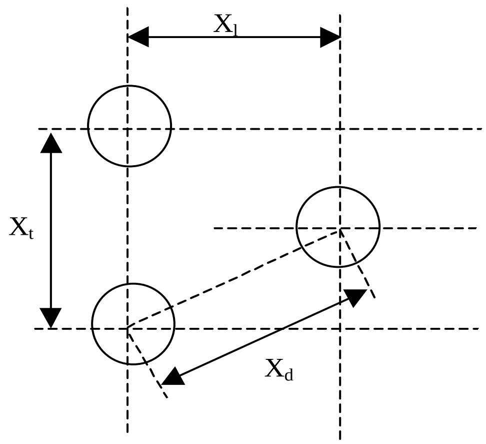
Power needed:

$$N = \frac{\dot{m}\Delta p}{\rho \eta_p} = \frac{1.5 \times 4267.6}{0.7383 \times 0.8} = 10.8 kW$$

η_p can be approximated as 80%.

Problem 4.7

Consider a tube bundle arrangement in crossflow heat exchanger. Tubes are staggered bare tubes with a longitudinal pitch of 20 mm, transverse pitch of 24.5 mm and 1/2 in. in diameter. The length of the heat exchanger is 0.80 m. The frontal area seen by the air stream is a 0.6 m × 0.6 m square. The air flows at 2 atm pressure with a mass flow rate of 1500 kg/h. Assume that the mean air temperature is 200°C. Calculate the air frictional pressure drop across the core of the heat exchanger.



GIVEN

- -Tube bundle in crossflow heat exchanger.
- -Mean temperature of air (T_m)= 200°C
- -Pressure of air flow (p) = 2 atm
- -Mass flow rate of air ($\dot{m}$) = 1500 kg/h
- -Pitch size of tube bundle (X_l) = 20 mm, and (X_t) = 24.5 mm.
- -Total frontal area: (A_f) = 0.6×0.6 m^2
- -Internal diameter of tubes (d_i) = 1/2 in. = 1.27 cm
- -length of heat exchanger (L) = 0.8 m

FIND

Frictional pressure drop across the core of the heat exchanger (Δp)

SOLUTION:

Properties of air at T = 200°C, p = 2 atm (Table B.1 in appendix B):

$$\rho = 1.495 \text{ kg/m}^3 \qquad \mu = 2.58\times 10^{-5} \text{ Pa.s}$$

Velocity:

$$X_d = \sqrt{X_l^2 + \left(\frac{X_t}{2}\right)^2} = \sqrt{20^2 + \left(\frac{24.5}{2}\right)^2} = 23.45mm$$

$$X_d - d_0 = 23.45 - 12.7 = 10.75$$

$$u_\infty = \frac{\dot{m}}{\rho A_f} = \frac{0.4166}{1.495 \times 0.36} = 0.775 \text{ m/s}$$

$$u_{max} = \frac{u_\infty X_t}{X_t - d_0} = \frac{0.775 \times 24.5}{24.5 - 12.7} = 1.607 m/s$$

$$\text{Re} = \frac{\rho u_m d_0}{\mu} = \frac{1.607 \times 1.495 \times 0.0127}{2.58 \times 10^{-5}} = 1183$$

$$X_l^* = \frac{X_l}{d_0} = \frac{0.02}{0.0127} = 1.575$$

$$X_t^* = \frac{X_t}{d_0} = \frac{0.0245}{0.0127} = 1.929$$

$$\frac{X_t^*}{X_l^*} = \frac{1.929}{1.575} = 1.22$$

From Fig. 4.5 for $\frac{X_t^*}{X_l^*} = 1.22$ and Re = 1183:

$$\chi = 1.0$$

From the main Fig. 4.5 for Re = 1183 and $X_t^* = 1.929$:

$$\frac{\text{Eu}}{\chi} = 0.60$$

Number of tube rows is approximately:

$$n = \frac{0.8}{X_l} = \frac{0.8}{0.02} = 40$$

Pressure drop:

$$\Delta p = \left(\frac{\text{Eu}}{\chi}\right) \cdot \chi \frac{\rho u_m^{\ 2}}{2} \cdot n = 0.60 \times 1.0 \times \frac{1}{2} \times 1.495 \times 1.607^2 \times 40 = 46.33 \text{ Pa}$$

Problem 4.8

Assume that the tube bundle arrangement given in Problem 3.11 is the core of a heat exchanger. Calculate the pressure drop caused by the tubes for in-line and staggered arrangements, and compare the two pressure drops and the two pumping powers.

SOLUTION:

* For in-line arrangement, n=20:

According to the solution of 3.11

$Re = 3130.68$

$$X_t^* = X_l^* = X_t/d_0 = X_l/d_0 = 0.05/0.025 = 2$$

thus, Eu/χ and χ can be determined from Figure 4.4

$Eu/\chi \cong 0.25$

$\chi = 1$

Pressure drop can be determined from Equation (4.20)

$$\Delta P_1 = \left(\frac{Eu}{\chi}\right)\chi\frac{1}{2}\rho u_0^2 \cdot n$$

$$= 0.25\times 1\times\frac{1}{2}\times 0.6157\times 6^2\times 20$$

$$= 55.4 \text{ Pa}$$

* For staggered arrangements, according to the solution of Problem 3.12

$Re_b = 3533.5$ and

$$X_t^* = X_t/d_0 = 0.05/0.025 = 2$$

$$X_l^* = X_l/d_0 = 0.04/0.025 = 1.6$$

$$\therefore X_t^*/X_l^* = 2/1.6 = 1.25$$

thus, Eu/χ and χ can be determined from Figure 4.4

$Eu/\chi \cong 0.25$

$\chi = 1$

Pressure drop can be determined from Equation (4.20)

$$\Delta P_2 = \left(\frac{Eu}{\chi}\right)\chi\frac{1}{2}\rho u_0^2 \cdot n$$

$$= 0.25\times 1\times\frac{1}{2}\times 0.6157\times 6.772^2\times 20$$

$$= 141.2 \text{ Pa}$$

$$\Delta P_2/\Delta P_1 = 141.2/55.4 = 2.55$$

* Pumping power comparison can be made from Equation (4.19)

$$\frac{W_{P2}}{W_{P1}} = \frac{\dfrac{1}{\eta_P} \cdot \dfrac{m}{\rho} \cdot \Delta P_2}{\dfrac{1}{\eta_P} \cdot \dfrac{m}{\rho} \cdot \Delta P_1} = \frac{\Delta P_2}{\Delta P_1} = 2.55$$

Problem 4.9

In a crossflow heat exchanger with in-line tubes, air flows across a bundle of tubes at 5°C and is heated to 32°C. The inlet velocity of air is 15 m/s. Dimensions of tubes are: d_o=25 mm, X_t=X_l=50 mm. There are 20 rows in the flow direction and 20 columns counted in the heat exchanger. The air mass flow rate is 0.5 kg/s. Air properties may be evaluated at 20°C and 1 atm. Calculate the frictional pressure drop and the pumping power.

GIVEN

- -Tube bundle in crossflow heat exchanger.
- -Inlet temperature of air (T_{c1}) = 5 °C
- -Outlet temperature of air (T_{c2}) = 32 °C
- -Pressure of air flow (p) = 1 atm
- -Velocity of air is (u_∞) = 15 m/s
- -Mass flow rate of air ($\dot{m}$) = 0.5 kg/s
- -Pitch size of tube bundle (X_l) = 50 mm, and (X_t) = 50 mm.
- -Diameter of tubes (d_o) = 25 mm
- -Number of rows (N_{row}) = 20
- -Number of columns (N_{column}) = 20

FIND

Frictional pressure drop (Δp) and pumping power (P_{pump})

<u>SOLUTION:</u>

Properties of air at T_m = 18.5 °C, p = 1 atm:

$$\rho = 1.2045 \ \text{kg/m}^3 \qquad \mu = 1.82\times10^{-5} \ \text{Pa.s}$$

$$u_{max} = \frac{u_\infty \times X_t}{(X_t - d_o)} = \frac{15\times0.05}{(0.05-0.025)} = 30 \ m/s$$

Reynolds number:

$$\text{Re} = \frac{\rho u_m d_o}{\mu} = \frac{1.2045\times30\times0.025}{1.82\times10^{-5}} = 49636$$

$$X_l^* = X_t^* = X_l/d_o = \frac{50}{25} = 2$$

From the main Fig. 4.4 for Re = 49,,636 and $X_t^* = 2$:

$$\frac{\text{Eu}}{\chi} = 0.21$$

Pressure drop:

$$\Delta p = \left(\frac{\text{Eu}}{\chi}\right)\frac{\rho u_m^2}{2}\cdot n = 0.21\times\frac{1}{2}\times1.2045\times30^2\times20 = 2277 \ \text{Pa}$$

$$P_{pump} = \frac{\dot{m}\Delta p}{\rho\eta_p} = \frac{0.5\times2277}{1.2045\times0.8} = 1182 \ \text{W}$$

Problem 4.10

In a heat exchanger for two different fluids, power expenditure per heating surface area is to be calculated. Fluids flow through the channels of the heat exchanger. Hydraulic diameter of the channel is 0.0241 m.
a. For air at an average temperature of 30°C, 2 atm pressure, heat transfer coefficient is 55 W/m^2.K
b. For water at an average temperature of 30°C, heat transfer coefficient is 3850 W/m^2.K.
Comparing the results of these two fluids, outline your conclusions.

GIVEN

-Hydraulic diameter (D_h) = 0.0241 m
-Air temperature (T_{air}) = 30 °C
-Air pressure = 2 atm
-Heat transfer coefficient for air (h_{air}) = 55 W/m^2.K
-Water temperature (T_{water}) = 30 °C
-Heat transfer coefficient for water (h_{water}) = 3850 W/m^2.K

FIND

Power expenditure per heating surface area (P)

SOLUTION:

Properties of air at T_m = 30°C, p = 2 atm:

$$\rho_a = 2.3312 \text{ kg/m}^3 \qquad \mu_a = 1.86\times10^{-5} \text{ Pa.s}$$
$$c_{p,a} = 1007 \text{ J/kg.K} \qquad k_a = 0.0264 \text{ W/m.K}$$

Properties of water at T_m = 30°C:

$$\rho_w = 996 \text{ kg/m}^3 \qquad \mu_w = 7.98\times10^{-4} \text{ Pa.s}$$
$$c_{p,w} = 4179 \text{ J/kg.K} \qquad k_w = 0.614 \text{ W/m.K}$$

Assume the pump efficiency η_p = 0.8:

a.

$$\frac{P}{A} = \frac{Ch^{3.5}\mu^{1.83}D_h^{\ 0.5}}{k^{2.33}c_p^{\ 1.17}\rho^2\eta_p} \qquad (C = 1.2465\times10^4)$$

$$\frac{P}{A} = \frac{(1.2465\times10^4)\times55^{3.5}(1.86\times10^{-5})^{1.83}\times0.0241^{0.5}}{0.0264^{2.33}\times1007^{1.17}\times2.3312^2\times0.8}$$
$$= 1.77 \text{ W/m}^2$$

b.

$$\frac{P}{A} = \frac{Ch^{3.5}\mu^{1.83}D_h^{\ 0.5}}{k^{2.33}c_p^{\ 1.17}\rho^2\eta_p} \qquad (C = 1.2465\times10^4)$$

$$\frac{P}{A} = \frac{(1.2465\times10^4)\times3850^{3.5}(7.98\times10^{-4})^{1.83}\times0.0241^{0.5}}{0.614^{2.33}\times4179^{1.17}\times996^2\times0.8}$$
$$= 3.34 \text{ W/m}^2$$

The power expenditure per heating surface area for given water is greater than the given air flow.

Problem 4.11

City water will be cooled in a heat exchanger by sea water entering at 15°C. The outlet temperature of the sea water is 20°C. City water will be re-circulated to reduce water consumption. The suction line of the pump has an I.D. of 154 mm, is 22m long, and has two 90° bends and a hinged check valve. The pipe from the pump to the heat exchanger has an I.D. of 127 mm, is 140 m long, and has six 90° bends. The 90° bends are all made of steel with a radius equal to the I.D. of the pipe, R/d=1.0. The heat exchanger has 62 tubes in parallel, each tube 6 m long. The I.D. of the tubes is 18 mm. All pipes are made of drawn mild steel (ε=0.0445 mm). The sea water flow rate is 120 m^3/h. Assume that there is one velocity head loss at the inlet and 0.5 velocity head loss at the outlet of the heat exchanger. The elevation difference is 10.5 m.

Calculate:

a. the total pressure drop in the system (kPa and m liquid head = Hm);

b. the power of the sea water pump (pump efficiency η=60%);

plot the pumping power as a function of the sea water flow rate (for three flow rate values: 2000 L/min., 2,500 L/min. and 3,000 L/min.).

GIVEN

-Sea water inlet temperature (T_{in}) = 15°C

-Sea water outlet temperature (T_{out}) = 20 °C

-Suction pipe has two 90° bends(R/d=1) and a hinged check valve, and

$d_{i,1}$ = 154 mm, L_1 = 22 m

-Pipe from pump to H.E. has six 90° bends(R/d=1), and

$d_{i,2}$ = 127 mm, L_2 = 140 m

-# of tubes in H.E. (N_t) = 62, and

$d_{i,3}$ = 18 mm, L_3 = 6 m

-Mass flow rate of sea water ($\dot{m}$) = 120 m^3/h

-Inlet loss (Δp_{inlet}) = $\frac{1}{2}\rho u^2$

-Outlet loss (Δp_{outlet}) = $0.5 \times \left(\frac{1}{2}\rho u^2\right)$

-Elevation (h) = 10.5 m

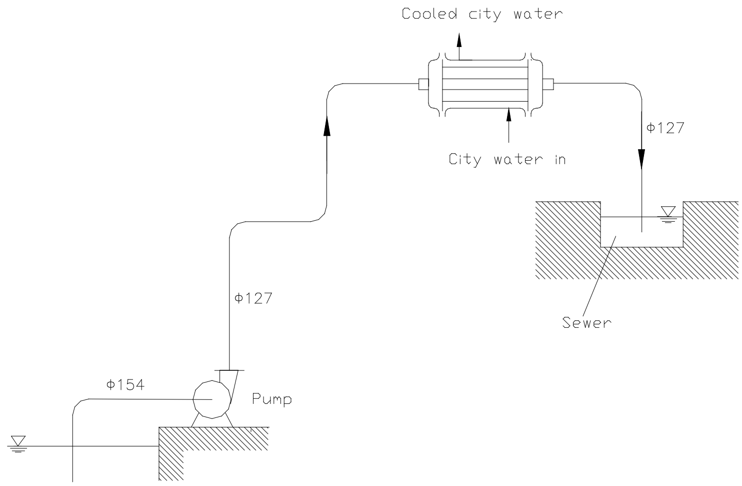
FIND

a. Total pressure drop in the system (Δp_{total})

b. Sea water pump power (P_{pump})

c. Plot the pumping power as a function of the sea water flow rate.

SKETCH



SOLUTION:

Properties of water at $T_m = \dfrac{15+20}{2} = 17.5\,^oC$:

$$\rho = 998.7 \text{ kg/m}^3 \qquad \mu = 10.67\times 10^{-4} \text{ Pa.s}$$

Velocities:

$$\dot{V} = 120 \text{ m}^3/\text{hr} = \frac{120}{3600} \text{ m}^3/\text{s} = 0.033 \text{ m}^3/\text{s}$$

$$\dot{m} = \rho\dot{V} = 998.7\times 0.033 = 33.29 \text{ kg/s}$$

* In suction line:

$$u_s = \frac{\dot{V}}{\frac{\pi d_s^2}{4}} = \frac{4\times 0.033}{\pi\times 0.154^2} = 1.79 \text{ m/s}$$

* In the pipe from pump to heat exchanger:

$$u_p = \frac{\dot{V}}{\frac{\pi d_p^2}{4}} = \frac{4\times 0.033}{\pi\times 0.127^2} = 2.63 \text{ m/s}$$

* In the heat exchanger tubes:

$$u_t = \frac{\dot{V}}{\frac{\pi d_t^2}{4}\cdot N_t} = \frac{4\times 0.033}{\pi\times 0.018^2\times 62} = 2.11 \text{ m/s}$$

Friction coefficient:

* In suction line:

$$Re = \frac{\rho u_s d_s}{\mu} = \frac{998.7\times 1.79\times 0.154}{10.67\times 10^{-4}} = 258014 \qquad \text{(Turbulent)}$$

$$f_s = 0.046\,Re^{-0.2} = 0.046\times 258014^{-0.2} = 0.0038$$

* In the pipe from pump to heat exchanger:

$$Re = \frac{\rho u_p d_p}{\mu} = \frac{998.7 \times 2.63 \times 0.127}{10.67 \times 10^{-4}} = 312629 \quad \text{(Turbulent)}$$

$$f_p = 0.046\, Re^{-0.2} = 0.046 \times 312629^{-0.2} = 0.0037$$

* In heat exchanger tubes:

$$Re = \frac{\rho u_t d_t}{\mu} = \frac{998.7 \times 2.11 \times 0.018}{10.67 \times 10^{-4}} = 35549 \quad \text{(Turbulent)}$$

$$f_t = 0.046\, Re^{-0.2} = 0.046 \times 35549^{-0.2} = 0.0056$$

a.

Pressure drop:

* In suction line:

	L/d_i	
2×90° bends	2×16.5 = 33	(R/D=1.0)
1 hinged check valve	110	
straight pipe	$\left(\frac{22}{0.154}\right)$	

$$\therefore \sum \frac{L}{d_i} = 285.86$$

$$\Delta p_s = 4 f_s \rho \frac{u_s^2}{2} \sum \frac{L}{d_i} = 4 \times 0.0038 \times 998.7 \times \frac{1.79^2}{2} \times 285.86 = 6951.9 \quad N/m^2$$

* In the pipe from pump to heat exchanger:

	L/d_i	
6×90° bends	6×16.5 = 99	(R/D=1.0)
straight pipe	$\left(\frac{140}{0.127}\right) = 1102.36$	

$$\therefore \sum \frac{L}{d_i} = 1102.36$$

$$\Delta p_p = 4 f_p \rho \frac{u_p^2}{2} \sum \frac{L}{d_i} = 4 \times 0.0037 \times 998.7 \times \frac{2.63^2}{2} \times 1102.36 = 56351 \ N/m^2$$

* In heat exchanger pipes:

$$\Delta p_t = 4 f_t \rho \frac{u_t^2}{2} \frac{N_t \cdot L}{d_i} = 4 \times 0.0056 \times 998.7 \times \frac{2.11^2}{2} \times \frac{62 \times 6}{0.018} = 1029173.1 \ N/m^2$$

* In the pipe from heat exchanger to the sewer:

By assuming that L = 15 m

	L/d_i	
2×90° bends	2×16.5 = 33	(R/D=1.0)
straight pipe	$\left(\frac{15}{0.127}\right) = 151.11$	

$$\therefore \sum \frac{L}{d_i} = 151.11$$

$$\Delta p_p = 4 f_p \rho \frac{u_p^{\ 2}}{2} \sum \frac{L}{d_i} = 4 \times 0.0036 \times 998.7 \times \frac{2.63^2}{2} \times 151.11 = 7515.7 \ \ N/m^2$$

* Pressure loss at the inlet and outlet of the heat exchanger:

$$\Delta p_{loss} = (1 + 0.5)\rho \frac{u_p^{\ 2}}{2} = 1.5 \times 998.7 \times \frac{2.63^2}{2} = 5181 \ N/m^2$$

* Pressure loss due to the elevation difference:

$$\Delta p_{elev} = \rho g h = 998.7 \times 9.82 \times 10.5 = 102976 \ N/m^2$$

Total pressure drop:

$$\Delta p_{total} = \Delta p_s + \Delta p_p + \Delta p_t + \Delta p_{h \to s} + \Delta p_{loss} + \Delta p_{elev}$$

$$= 6951.5 + 56351 + 1029173 + 7515.7 + 5181 + 102976$$

$$= 1213.2096 \text{ kPa} \approx 1213 \text{ kPa}$$

b.

Pumping power:

$$P = \frac{\dot{m} \cdot \Delta p_{total}}{\rho \cdot \eta_p} = \frac{33.29 \times 1213}{998.7 \times 0.6} = 67.4 \quad kW$$

Problem 4.12

For the situation given in Problem 4.11,

a. Plot the total pressure head vs. flow rate (m liquid head vs. lt/min.).

b. Change the pipe diameter before and after the pump and repeat (a), (b), and (c) in Problem 4.11.

c. Obtain the optimum pipe diameter considering the cost of pumping power (operational cost) plus the fixed charges based on capital investment for the pipe installed.

GIVEN

This problem can be carried out as Problem 4.11.

Problem 4.13

A shell-and-tube heat exchanger is to be used to cool 25.3 kg/s of water from 38°C to 32°C. The exchanger has one shell-side pass and two tube-side passes. The hot water flows through the tubes, and the cooling water flows through the shell. The cooling water enters at 20°C and leaves at 30°C. The shell-side (outside) heat transfer coefficient is estimated to be 5678 W/m^2.K. Design specifications require that the pressure drop through the tubes be as close to 13.8 kPa as possible, that the tubes be 18 BWG copper tubing (1.24 mm wall thickness), and that each pass be 4.9 m long. Assume that the pressure losses at the inlet and outlet are equal to one and one half of a velocity heat ($\rho u_m^2/2$), respectively. For these specifications, what tube diameter and how many tubes are needed?

GIVEN

- Hot water inlet temperature (T_{h1}) = 38°C
- Hot water outlet temperature (T_{h2}) = 32 °C
- Mass flow rate of hot water ($\dot{m}_h$) = 25.3 kg/s
- Cooling water inlet temperature (T_{c1}) = 20°C
- Cooling water outlet temperature (T_{c2}) = 30 °C
- Heat transfer coefficient on shell side (h_o) = 5678 W/m^2.K
- # of tube passes (N_p) = 2
- Required pressure drop (Δp) = 13.8 kPa
- Tube specification: 18 BWG (1.24 mm wall thickness)
  Length (L) = 4.9 m
- Inlet loss (Δp_{inlet}) = $\frac{1}{2}\rho u_m^2$
- Outlet loss (Δp_{outlet}) = $0.5\times\left(\frac{1}{2}\rho u_m^2\right)$

FIND

Tube diameter d_i and the # of tubes.

SOLUTION:

Properties of water at $T_m = \frac{38+32}{2} = 35$ °C:

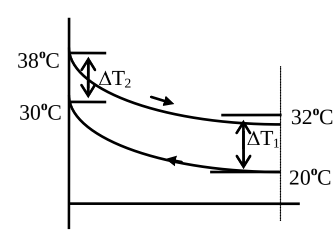
$$\rho = 994 \text{ kg/m}^3 \qquad \mu = 7.20\times10^{-4} \text{ Pa.s}$$
$$k = 0.62 \text{ W/m.K} \qquad Pr = 4.78$$
$$c_{p,h} = 4178 \text{ J/kg.K}$$

Heat duty:

$$\dot{Q} = \dot{m}_h c_{p,h}\left(T_{h1} - T_{h2}\right) = 25.3\times4178\times(38-32) = 634.22 \text{ kW}$$

Log mean temperature:

$$\Delta T_{lm,cf} = \frac{\Delta T_1 - \Delta T_2}{\ln\frac{\Delta T_1}{\Delta T_2}} = \frac{8-12}{\ln\frac{8}{12}} = 9.86 \ {}^{o}\text{C}$$



Correction factor:

$$P = \frac{T_{c2} - T_{c1}}{T_{h1} - T_{c1}} = \frac{30 - 20}{38 - 20} = 0.56 \quad ; \qquad R = \frac{T_{h1} - T_{h2}}{T_{c2} - T_{c1}} = \frac{38 - 32}{30 - 20} = 0.6$$

From figure: F = 0.83

$$\left.\begin{array}{l} Q = UAF\Delta T_{lm,cf} \\ A = \pi d_o L N_t = \pi(d_i + 2t) L N_t \end{array}\right\} \Rightarrow \pi(d_i + 2t) L N_t = \frac{Q}{UF\Delta T_{lm,cf}}$$

We can approximate the tube heat transfer coefficient as $h_i = 6000\ W/m^2.K$, and

$$U = \left(\frac{1}{h_o} + \frac{1}{h_i}\right)^{-1} = \left(\frac{1}{6000} + \frac{1}{5678}\right)^{-1} = 2917.3 \quad W/m^2 \cdot K$$

$$(d_i + 2t) N_t = \frac{Q}{\pi L U F \Delta T_{lm,cf}} = \frac{634.22 \times 10^3}{\pi \times 4.9 \times 2917.3 \times 0.88 \times 9.86} = 1.63$$

$$\therefore N_t = \frac{1.63}{d_i + 0.00248}$$

Pressure drop:

$$\Delta p = \frac{1}{2}\rho u_m^2 \left(1 + \frac{1}{2} + 4f\,\frac{L \cdot N_t}{d_i}\right) = \frac{1}{2}\rho u_m^2 \left(\frac{3}{2} + 4f\,\frac{L \cdot N_t}{d_i}\right)$$

Velocity:

$$u_m = \frac{\dot{m}}{\rho \dfrac{\pi d_i^2}{4} \cdot \dfrac{N_t}{N_p}} = \frac{4 \times 25.3 \times 2}{994 \times \pi} \cdot \frac{1}{d_i^2 N_t} = \frac{0.065}{d_i^2 N_t}$$

Reynolds number:

$$Re = \frac{\rho u_m d_i}{\mu} = \frac{\rho d_i}{\mu} \cdot \frac{4\dot{m} N_p}{\rho \pi d_i^2 N_t} = \frac{4\dot{m} N_p}{\pi \mu d_i N_t} = \frac{4 \times 25.3 \times 2}{\pi \times (7.20 \times 10^{04})} \cdot \frac{1}{d_i N_t} = \frac{89480.4}{d_i N_t}$$

Assuming a turbulent flow, the friction factor is:

$$f = [1.58 \ln(Re) - 3.28]^{-2} = [1.58 \ln(89480.4) - 1.58 \ln(d_i N_t) - 3.28]^{-2}$$
$$= [14.73 - 1.58 \ln(d_i N_t)]^{-2} = 0.4[9.32 - \ln(d_i N_t)]^{-2}$$

Pressure drop:

$$\Delta p = \frac{1}{2} \times 994 \times \left(\frac{0.065}{d_i^2 N_t}\right)^2 \left\{1.5 + 4 \times 0.4 \times [9.32 - \ln(d_i N_t)]^{-2} \cdot \frac{4.9 \cdot N_t}{d_i}\right\}$$

Δp is required as 13.8 kPa as given in the statement, so

$$\Delta p = \frac{3.15}{(d_i^2 N_t)^2} \left\{1 + \cdot \frac{5.23}{d_i} [9.32 - \ln(d_i N_t)]^{-2} \cdot N_t\right\} = 13.8 \times 10^3$$

Substituting $N_t = \dfrac{1.63}{d_i + 0.00248}$, we get the following equation with one unknown, d_i:

$$\frac{(d_i + 0.00248)^2}{d_i^4}(1.186)\left\{1 + \cdot \frac{5.23}{d_i}\left[9.35 - \ln\left(\frac{1.63 d_i}{d_i + 0.00248}\right)\right]^{-2} \cdot N_t\right\} = 13.8 \times 10^3$$

This equation can be solved using the trial method.

then $N_t = \dfrac{1.63}{0.015875 + 0.00248} = 88.8 \approx 89$ tubes

For d_i = 5/8 in. = 0.01587 m, the pressure drop is

$$\Delta p = \frac{1}{2} \times 994 \times \left(\frac{0.065}{d_i^{\,2} N_t} \right)^2 \left\{ 1.5 + 4 \times 0.4 \times \left[9.32 - \ln\left(d_i N_t \right) \right]^{-2} \cdot \frac{4.9 \cdot N_t}{d_i} \right\}$$

$$= \frac{1}{2} \times 994 \times \left(\frac{0.065}{0.015875^2 \times 89} \right)^2 \left\{ 1.5 + 4 \times 0.4 \times \left[9.32 - \ln\left(0.015875 \times 89 \right) \right]^{-2} \cdot \frac{4.9 \cdot 89}{0.015875} \right\}$$

$$= 6472.3 \text{ Pa}$$

Problem 4.14

Water is to be heated from 10°C to 30°C at the rate of 300 kg/s by atmospheric pressure steam in a single-pass shell-and-tube heat exchanger consisting of 1-in. schedule 40 steel pipe. The surface coefficient on the steam side is estimated to be 11,350 W/m^2.K. An available pump can deliver the desired quantity of water provided that the pressure drop through the pipes does not exceed 15 psi. Calculate the number of tubes in parallel and the length of each tube necessary to operate the heat exchanger with the available pump.

GIVEN

- Cooling water inlet temperature (T_{c1}) = 10°C
- Cooling water outlet temperature (T_{c2}) = 30 °C
- Mass flow rate of hot water ($\dot{m}_c$) = 300 kg/s
- Steam temperature (T_s) = 100°C
- Heat transfer coefficient on shell side (h_o) = 11,350 W/m^2.K
- Required pressure drop (Δp) = 15 psi = 103,442 Pa
- Tube specification: (d_o) = 1.315 in. = 0.0334 m
  (d_i) = 1.049 in. = 0.0266 m

FIND

The # of tubes and the length of tube L.

SOLUTION:

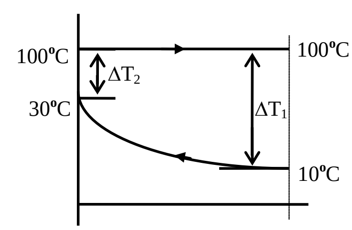
Properties of water at $T_m = \dfrac{10+30}{2} = 20\ ^{o}C$:

$$\rho = 998\ \text{kg/m}^3 \qquad \mu = 10.07\times10^{-4}\ \text{Pa.s}$$
$$c_{p,c} = 4182\ \text{J/kg.K} \quad k = 0.599\ \text{W/(m.K)}$$
$$Pr = 7.05$$

Heat duty:

$$\dot{Q} = \dot{m}_c c_{p,c}\left(T_{c1} - T_{c2}\right) = 300\times4182\times(30-10) = 25{,}092\ \text{kW}$$

Log mean temperature:



$$\Delta T_{lm,cf} = \frac{\Delta T_1 - \Delta T_2}{\ln\dfrac{\Delta T_1}{\Delta T_2}} = \frac{90-70}{\ln\dfrac{90}{70}} = 79.6\ ^{o}C$$

We can approximate the tube heat transfer coefficient as h_i = 6000 W/m^2.K, and the overall heat transfer coefficient is:

$$U = \left(\frac{1}{h_o} + \frac{1}{h_i}\right)^{-1} = \left(\frac{1}{6000} + \frac{1}{11350}\right)^{-1} = 3925\ \ W/m^2\cdot K$$

$$\left.\begin{aligned} Q &= UA\Delta T_{lm,cf} \\ A &= \pi d_o L N_t \end{aligned}\right\} \Rightarrow LN_t = \frac{Q}{\pi d_o U \Delta T_{lm,cf}} = \frac{25092\times10^3}{\pi\times0.0334\times3925\times79.6} = 765.4$$

Velocity:

$$u_m = \frac{\dot{m}}{\rho\dfrac{\pi d_i^{\ 2}}{4}\cdot N_t} = \frac{4\times300}{998\times\pi}\cdot\frac{1}{0.0266^2 N_t} = \frac{541}{N_t}$$

Reynolds number:

$$\text{Re} = \frac{\rho u_m d_i}{\mu} = \frac{\rho d_i}{\mu} \cdot \frac{541}{N_t} = \frac{998 \times 0.0266}{\left(10.07 \times 10^{-4}\right)} \cdot \frac{541}{N_t} = \frac{14{,}261{,}985}{N_t}$$

Assuming a turbulent flow, the friction factor is:

$$f = \left[1.58 \ln(\text{Re}) - 3.28\right]^{-2} = \left[1.58 \ln(14{,}261{,}985) - 1.58 \ln(N_t) - 3.28\right]^{-2}$$

$$= 0.4\left[14.4 - \ln(N_t)\right]^{-2}$$

Pressure drop:

$$\Delta p = 4f \cdot N_t \frac{L}{d_i}\left(\frac{1}{2}\rho u_m{}^2\right) = 4 \times 0.4\left[14.4 - \ln(N_t)\right]^{-2} \cdot N_t \cdot \frac{765.4}{N_t} \cdot \frac{1}{2}\rho\left(\frac{541}{N_t}\right)^2$$

$$\Delta p = 4 \times 0.4\left[14.42 - \ln(N_t)\right]^{-2} \frac{1}{2} \times 998 \times \left(\frac{541}{N_t}\right)^2 \times (765.4)$$

$$= \left(1.78856 \times 10^{11}\right)\left[14.4 - \ln(N_t)\right]^{-2} \cdot \frac{1}{N_t{}^2} \leq 15 \ \text{psi} = 103{,}422$$

For N_t = 140 tubes, Δp = 102004 Pa < $\Delta p_{admissible}$ and

L = 765.4/140 = 5.5 m

Problem 5.1

Using the Table 3.8 to calculate the Nusselt number in a rectangular channel having aspect ratio of 0.5, if the width of the channel is 100 μm and Kn number is 0.12, calculate the heat transfer coefficient for air at 40°C for two different boundary conditions.

GIVEN

- a rectangular channel aspect ratio = 0.5
- width of the channel = 100 μm
- Kn number is 0.12
- air at 40°C

FIND

a. The Nusselt number in a rectangular channel
b. The heat transfer coefficient for air at 40°C for two different boundary conditions

SOLUTION:

Properties of Air, from Appendix B (Table B.1), are

$$\rho = 1.1267 \text{ kg/m}^3$$
$$\mu = 1.91 \times 10^{-5} \text{ kg/(m} \cdot \text{s)}$$
$$c_p = 1.009 \text{ kJ/(kg} \cdot \text{K)}$$
$$c_v = 0.7185 \text{ kJ/(kg} \cdot \text{K)}$$
$$k = 0.0271 \text{ W/(m} \cdot \text{K)}$$
$$\text{Pr} = 0.711$$

Given parameters;

$$\frac{2b}{2a} = 0.5$$
$$2b = 100\ \mu m$$
$$2a = 200\ \mu m$$
$$D_h = \frac{4(100 \times 200)}{\left[(2 \times 100) + (2 \times 200)\right]} = 133.333\ \mu m$$
$$\text{Kn} = 0.12$$

From the Table 3.8, for the given dimension, Nusselt number for the constant heat flux condition can be obtained as

$$Nu = 4.123$$

So the heat transfer coefficient for the constant heat flux condition is

$$h = \frac{Nu \times k}{d} = \frac{4.123 \times 0.0271}{0.133 \times 10^{-3}} = 838 \text{ W/m}^2\text{K}$$

From the Table 3.8, for the given dimension, Nusselt number for the constant wall temperature condition can be obtained as

$$Nu = 3.391$$

So the heat transfer coefficient for the constant wall temperature condition is

$$h = \frac{Nu \times k}{d} = \frac{3.391 \times 0.0271}{0.133 \times 10^{-3}} = 690.95 \text{ W/m}^2\text{K}$$

Problem 5.2

A microchannel heat exchanger has been designed for cooling an electronic chip of size 30 mm by 30 mm. The transition Reynolds number is 800. The average Nusselt numbers, can be calculated from the following correlations:

$Nu = 0.0384 Re^{0.67} Pr_{nf}^{0.33}$ **for Laminar Flow**

$Nu = 0.00726 Re^{0.8} Pr_{nf}^{0.33}$ **for Turbulent Flow**

To cool the chips, a square microchannel of 250 µm is chosen where the permissible water velocity is 1 m/s and fluid temperature is 40°C.
a. Calculate the heat transfer coefficient
b. If the length of the microchannel has been given as 20 mm, calculate the friction coefficient and the pressure drop

GIVEN

- -A square microchannel of 250 µm
- -The transition Reynolds number = 800
- -water velocity = 1 m/s
- -fluid temperature = 40°C

FIND

a. the heat transfer coefficient
b. the friction coefficient and the pressure drop

SOLUTION:

$$D_h = \frac{2 \times a \times b}{a+b} = \frac{2 \times 250 \times 250}{250 + 250} = 250 \mu m$$

Properties of Water at $40^{\circ}C$, from Appendix B (Table B.2), are

$$\rho = 992.22 (kg/m^3)$$
$$\mu = 0.00065298 (kg/m \cdot s)$$
$$c_p = 4.1794 (kj/kg \cdot K)$$
$$c_v = 4.0734 (kj/kg \cdot K)$$
$$k = 0.63063 (W/m \cdot K)$$
$$\text{Pr} = 4.34$$

$$\text{Re} = \frac{u_m \times \rho \times D_h}{\mu} = \frac{1 \times 992.22 \times 250 \times 10^{-6}}{0.00065298} = 379.88$$

379.88 < 800, so flow is laminar.

$$L_h = 0.056 \times \text{Re} \times D_h = 0.056 \times 379.88 \times 250 \times 10^{-6} = 5.32 \times 10^{-3} m = 5.32 mm$$

5.32 mm < 20 mm so the flow is hydrodynamically fully developed.

$$L_t = 0.05 \times \text{Re} \times \text{Pr} \times D_h = 0.05 \times 379.88 \times 4.34 \times 250 \times 10^{-6} = 0.0205 m = 20.5 mm$$

20.5 mm > 20 mm ,but it is very close to the thermally fully developed region, so we can assume that the flow is thermally fully developed.

$$Nu_{\infty} = 0.0384 \times \mathrm{Re}^{0.62} \times \mathrm{Pr}^{0.33} = 0.0384 \times 379.88^{0.62} \times 4.34^{0.33} = 2.48$$

a.

$$h = \frac{Nu_{\infty} \times k}{d} = \frac{2.48 \times 0.63063}{250 \times 10^{-6}} = 6255.8 \;\; W / m^2 K$$

b.

Friction Factor;

f = 16 / Re = 16/379.88= 0.042

$$\Delta p = 4 \times f \times \frac{L}{D_h} \times \frac{\rho \times u_m^{\;2}}{2} = 4 \times 0.042 \times \frac{0.02}{250 \times 10^{-6}} \times \frac{992.22 \times 1}{2} = 6.667 kPa = 0.067 bar$$

Problem 5.3

Repeat the problem 5.2, if the permissible water velocity is changed to be 2 m/s but the fluid temperature is still the same. By using a proper correlation, determine the average heat transfer coefficient.

GIVEN

-A square microchannel of 250 µm
-The transition Reynolds number = 800
-water velocity = 2 m/s
-fluid temperature = 40°C

FIND

the average heat transfer coefficient

SOLUTION:

$$Re = \frac{u_m \times \rho \times D_h}{\mu} = \frac{2 \times 992.22 \times 250 \times 10^{-6}}{0.00065298} = 759.76$$

$$L_h = 0.056 \times Re \times D_h = 0.056 \times 759.76 \times 250 \times 10^{-6} = 10.64 \times 10^{-3} m = 10.64 mm$$

We can use the length of the micro channel as 20 mm.
10.64 mm < 20 mm so the flow is hydrodynamically fully developed.

$$L_t = 0.05 \times Re \times Pr \times D_h = 0.05 \times 759.76 \times 4.34 \times 250 \times 10^{-6} = 0.041 m = 41 mm$$

41 mm > 20 mm so the flow is not thermally fully developed.

Choi et al. correlation which is proper for the conditions given below;

$$Nu = 0.000972 \times Re^{1.17} \times Pr^{1/3} \qquad Re < 2000$$

$$Nu = 3.82 \times 10^{-6} \times Re^{1.96} \times Pr^{1/3} \qquad 2500 < Re < 20000$$

The Reynolds number is 759.76 < 2000 so the flow is laminar. We can use the Choi et al. correlation for laminar flow.

$$Nu_\infty = 0.000972 \times Re^{1.17} \times Pr^{1/3} = 0.000972 \times 759.76^{1.17} \times 4.34^{1/3} = 3.72$$

$$Nu_l = Nu_\infty \left[1 + \left(\frac{d}{l}\right)^{2/3}\right] = 3.72\left[1 + \left(\frac{250 \times 10^{-6}}{0.02}\right)^{2/3}\right] = 3.92$$

$$h = \frac{Nu \times k}{d} = \frac{3.92 \times 0.63063}{250 \times 10^{-6}} = 9889.2 \quad W/m^2K$$

Problem 5.4

If the microchannel from the problem 5.2 is changed from the square microchannel to be the trapezoidal microchannel. The width of the microchannel is 125 μm, the length of the microchannel is 62.5 μm, the side angle is 44 degree, the permissible water velocity is 1.5 m/s and the fluid temperature is 40°C. By selecting a proper correlation, calculate
a. the average heat transfer coefficient,
b. the pressure drop

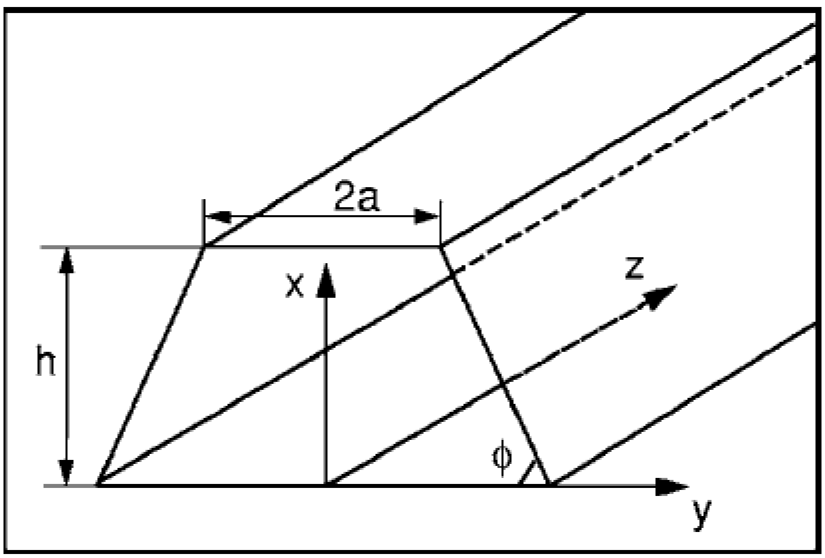
GIVEN

-the trapezoidal microchannel: width = 125 μm, the length = 62.5 μm, side angle = 44 degree
-water velocity = 1.5 m/s
-fluid temperature = 40°C

FIND

a. the average heat transfer coefficient,
b. the pressure drop

SOLUTION:



2a = 125 μm, h = 62.5 μm, φ = 45°, and long side = 250 μm

$$D_h = \frac{4 \times A_c}{wetted\ perimeter} = \frac{3 \times 62.5 \times 250}{125 + 250 + 2 \times {62.5}/{\cos 45}} = 84.9528\mu m$$

a.

$$\text{Re} = \frac{u_m \times \rho \times D_h}{\mu} = \frac{1.5 \times 992.22 \times 84.9528 \times 10^{-6}}{0.00065298} = 193.632$$

According to Renksizbulut et. al. correlation the entrance length is

$$\frac{L}{D_h} = \left(0.085Re + \frac{0,8}{Re^{0.3}}\right)\left(\frac{90°}{\varphi}\right)^{0.6} (1+\alpha)^{-0.24} \quad \text{for} \quad 0.5 \le \alpha \le 2 \text{ and } 10 \le \text{Re} \le 1000$$

where

$$\alpha = \frac{h}{2a} = \frac{62.5}{125} = 0.5 \text{ and Re} = 193.632$$

So

$$L = 84.9528 \times 10^{-6} \times \left(0.085 \times 193.632 + \frac{0,8}{193.632^{0.3}}\right)\left(\frac{90°}{45°}\right)^{0,6} (1+0.5)^{-0.24}$$
$$= 1.94 \times 10^{-3} m$$

Renksizbulut et. al. correlation for the Nusselt number is available for $0.1 \le \text{Re} \le 1000$

$$Nu = \left\{2.87\left(\frac{90°}{\varphi}\right)^{-0,26} + 4.8exp\left[-3.9\alpha\left(\frac{90°}{\varphi}\right)^{0.21}\right]\right\} \times G$$

$$G = [1 + 0.075(1+\alpha) \times exp(-0.45Re)]$$

Then

$$G = [1 + 0.075(1+0.5) \times exp(-0.45 \times 193.632)] = 1$$

and

$$Nu = \left\{2.87\left(\frac{90°}{45°}\right)^{-0,26} + 4.8exp\left[-3.9 \times 0.5 \times \left(\frac{90°}{45°}\right)^{0.21}\right]\right\} \times 1 = 2.90$$

$$h = \frac{Nu \times k}{d} = \frac{2.90 \times 0.63063}{84.9528 \times 10^{-6}} = 21527.57 W/m^2K$$

b.

Renksizbulut et. al. correlation for friction factor is;

$$f \times Re = 13.9\left(\frac{90°}{\varphi}\right)^{-0.07} + 10.4exp\left[-3.25\alpha\left(\frac{90°}{\varphi}\right)^{0.23}\right] \; for \; 0 \le \alpha \le 1$$

Then

$$f = \left\{13.9\left(\frac{90°}{45°}\right)^{-0.07} + 10.4exp\left[-3.25 \times 0.5 \times \left(\frac{90°}{45°}\right)^{0.23}\right]\right\} \times \frac{1}{193.632} = 0.07606$$

The pressure drop is

$$\Delta p = 4 \times f \times \frac{L}{D_h} \times \frac{\rho \times u_m{}^2}{2} = 4 \times 0.07606 \times \frac{0.02}{84.9528 \times 10^{-6}} \times \frac{992.22 \times 1.5^2}{2} = 79.95kPa = 0.8bar$$

Problem 5.5

Consider the properties of the nanofluid, Al_2O_3/water, at 30ºC, calculate the effective thermal conductivity of 6% volume fraction of nanofluid by using the Hamilton and Crosser model and assuming that the particles are spherical. Repeat the problem with 8% volume fraction nanofluid at the same water temperature.

GIVEN

-nanofluid: Al_2O_3/water
-temperature = 30ºC

FIND

a. the effective thermal conductivity of 6% volume fraction
b. the effective thermal conductivity of 8% volume fraction

SOLUTION:

$$\frac{k_{nf}}{k_f} = \frac{k_p + (n-1)k_f + (n-1)(k_p - k_f)\phi}{k_p + (n-1)k_f - (k_p - k_f)\phi}$$

$$\Psi = 1\ (n = 3)\ and\ (\phi = 0.06\ , \phi = 0.08)$$

$$k_p = 30\frac{W}{m.K}\ , k_f = 0.617\ \frac{W}{m.K}$$

$$\phi = 0.06 => k_{nf} = 0.728\ \frac{W}{m.K}$$

$$\phi = 0.08 => k_{nf} = 0.768\ \frac{W}{m.K}$$

Problem 5.6

Consider the Nusselt number for fully developed laminar flow under constant heat flux and constant wall temperature boundary conditions are constant as 4.36 and 3.66, respectively. If the nanofluid of Al_2O_3/water flows in a tube with a diameter of 0.5 cm at a bulk temperature of 40°C, calculate the heat transfer coefficients for both boundary conditions and compare these values with the heat transfer coefficients of pure water in both conditions to find the heat transfer enhancement, if the volume fraction is

a. 0.05% and sphericity value equals to 1,

b. 0.05% and sphericity value equals to 3.

GIVEN

-nanofluid: Al_2O_3/water

-Nu = 4.36; constant heat flux boundary condition

-Nu = 3.66; constant wall temperature boundary condition

-temperature = 40^{o}C

-tube diameter = 0.5 cm

FIND

a. the heat transfer enhancement; 0.05% and sphericity value equals to 1

b. the heat transfer enhancement; 0.05% and sphericity value equals to 3

SOLUTION:

from Table 5.8; Alumina (Al_2O_3)

k_f = 0.613 W/m-K (from Table 5.8; water)

ϕ = 5% = 5/100 = 0.05

The thermal conductivities of Nanofluid, Alumina (Al_2O_3)/Water, by using classical models; Maxwell (1873)

$$k_{nf} = \frac{k_p + 2k_f + 2(k_p - k_f)\phi}{k_p + 2k_f - (k_p - k_f)\phi} k_f$$

$$= \frac{(40) + (2)(0.613) + (2)(40 - 0.613)(0.05)}{(40) + (2)(0.613) - (40 - 0.613)(0.05)} (0.613)$$

k_{nf} = 0.705 W/m-K

The thermal conductivities of Nanofluid, Alumina (Al_2O_3)/Water, by using classical models; Hamilton and Crosser

a.

From $n = \dfrac{3}{\psi}$ $(\psi = 1, n = 3)$

$$k_{nf} = \frac{k_p + (n-1)k_f - (n-1)\phi(k_f - k_p)}{k_p + (n-1)k_f + \phi(k_f - k_p)} k_f$$

$$= \frac{(40) + (3-1)(0.613) - (3-1)(0.05)(0.613 - 40)}{(40) + (3-1)(0.613) + (0.05)(0.613 - 40)} (0.613)$$

$$k_{nf} = 0.705 \quad \text{W/m-K}$$

b.

From $$n = \frac{3}{\psi} \qquad (\psi = 3, n = 1)$$

$$k_{nf} = \frac{k_p + (n-1)k_f - (n-1)\phi(k_f - k_p)}{k_p + (n-1)k_f + \phi(k_f - k_p)} k_f$$

$$= \frac{(40) + (1-1)(0.613) - (1-1)(0.05)(0.613 - 40)}{(40) + (1-1)(0.613) + (0.05)(0.613 - 40)}(0.613)$$

$$k_{nf} = 0.645 \quad \text{W/m-K}$$

Problem 5.7

Consider the solved Example 3.3 where the heat transfer coefficient in the inner tube is calculated; if a hot nanofluid of Al_2O_3/water flows through the inner tube with 5% of particle volume fraction and sphericity value equals to 1, calculate the heat transfer coefficient by the use of an experimentally obtained nanofluid correlation given by $Nu_{nf} = 0.0059\left(1.0+7.6286\phi^{0.6886}Pe_d^{0.001}\right)Re_{nf}^{0.9238}Pr_{nf}^{0.4}$**, Eq. (5.97).**

GIVEN

-A hot nanofluid; Al_2O_3/water
-particle volume fraction = 5%
-Nusselt number; $Nu_{nf} = 0.0059\left(1.0+7.6286\phi^{0.6886}Pe_d^{0.001}\right)Re_{nf}^{0.9238}Pr_{nf}^{0.4}$

FIND

the heat transfer coefficient in the inner tube

SOLUTION:

$\Delta T_1 = \Delta T_2 = \Delta T_m = 105°C = 378K$

$$T_{bulk} = \frac{140+125}{2} = 132℃ = 405K\ , T_{wall} = 105℃ = 378K$$

Base fluid properties:
at wall temperature

$$\rho = \frac{1}{1.047*10^{-3}} = 955.11\frac{kg}{m^3}\ , \mu = 268*10^{-6}\ Pa.s$$

$$k = 682*10^{-3}\ \frac{W}{m.K}$$

at bulk temperature

$$\rho = \frac{1}{1.072*10^{-3}} = 932.836\frac{kg}{m^3}\ , \mu = 209*10^{-6}\ Pa.s$$

$$k = 688*10^{-3}\frac{W}{m.K}\ , Pr = 1.325\ , c_p = 4271\frac{J}{kg.K}$$

$$Re_b = \frac{\rho u_m di}{\mu} = \frac{4\dot{m}_{nf}}{\pi\mu_{nf}di} = \frac{4*1.36}{\pi*0.0525*0.209*10^{-3}} = 15485$$

$$Nu_f = 0.0059*\left(1.0+7.6286\phi^{0.6886}Pe_d^{0.001}\right)Re_{nf}^{0.9238}Pr_{nf}^{0.4} = 97.0211$$

Nanofluid properties:

$$\mu_{nf,b} = (1+7.3\phi+123\phi^2)\mu_{f,b} = 3.496*10^{-4}Pa.s$$

$$\mu_{nf,w} = (1+7.3\phi+123\phi^2)\mu_{f,w} = 4.48*10^{-4}Pa.s$$

$$(\dot{m}*c_p)_h*\Delta T_h = (\dot{m}*c_p)_c*\Delta T_c$$

$$\Delta T_h = \Delta T_c => \dot{m}_{h,nf} = \frac{\dot{m}_c*c_{p,c}}{c_{p,h}}$$

$$c_{p,c} = 4179\,\frac{J}{kg.K}$$

$$c_{p,h} = c_{p,nf} = \phi c_{p,p} + (1-\phi)c_{p,f} => c_{p,nf} = 4087.45\,\frac{J}{kg.K}$$

$$\dot{m}_{h,nf} = 1.42\,\frac{kg}{s}$$

$$Re_b = \frac{\rho u_m di}{\mu} = \frac{4\dot{m}_{nf}}{\pi \mu_{nf} di} = 98557.11 > 2300\ \ Turbulant\ flow$$

$$k_{static} = \frac{30 + 2*0.688 - 2*0.05*(0.688-30)}{30 + 2*0.688 + 0.05*(688-30)} * 0.688 = 0.7891\,\frac{W}{m.K}$$

$k_{Brownian}$ negligible => $k_{nf} = k_{static} = 0.7891$ W/m.K

$$Pr_{nf} = \frac{c_{p,nf}\mu_{nf}}{k_{nf}} = 1.81$$

$$\rho_{nf} = \phi\rho_p + (1-\phi)\rho_f = 1086.1942\,\frac{kg}{m^3}$$

$$Nu_{nf} = \left(\frac{Pe_{nf}}{Pe_f}\right)^{\frac{1}{3}} = \left(\frac{k_f \rho_{nf} c_{p,nf}}{k_{nf}\rho_f c_{p,f}}\right)^{\frac{1}{3}} * Nu_f = 342.68$$

$$enhancement => \frac{h_{nf}}{h_f} = (\frac{\rho_{nf} c_{p,nf}}{\rho_f c_{p,f}})^{1/3} * (\frac{k_{nf}}{k_f})^{2/3} = 1.136$$

Although Nusselt Number decreases in nanofluid, increase of thermal conductivities causes increase in convection coefficient. ($\frac{h_{nf}}{h_f} > 1$)

$$h_f = \frac{k_f * Nu_f}{d} = \frac{97.0211 * 0.688}{0.0525} = 1271.438\,\frac{W}{m^2.K}$$

$$\frac{h_{nf}}{h_f} = 1.136 => h_{nf} = 1444.354\,\frac{W}{m^2.K}$$

Problem 5.8

The CuO/water nanofluid, k_{CuO} = 401 $\mathrm{W/(m\cdot K)}$ and k_{water} = 0.613 $\mathrm{W/(m\cdot K)}$, flows in a tube under the fully developed turbulent flow condition at 30ºC. If an inside diameter of the tube is 0.5 cm,
a. using a proper correlation, determine the effective heat transfer coefficient of the nanofluid for the constant wall temperature and constant heat flux boundary conditions, assuming the volume fraction of the spherical nanoparticles is 8%,
b. compare the values with the heat transfer coefficients of water under the same conditions to find the enhancement.

SOLUTION:

$$properties\ of\ water: k_{water} = 0.613\frac{W}{m.K}\ ,\rho_f = \frac{1}{1.004*10^{-3}} = 996.016\frac{kg}{m^3}$$

$$properties\ of\ copper\ [1]: k_{cu} = 401\frac{W}{m.K}\ ,\rho = 8.94*10^3\frac{kg}{m^3}\ ,c_p = 384.6\frac{J}{kg.K}$$

$$\frac{k_{nf}}{k_f} = \frac{k_p + 2k_f + 2(k_p - k_f)\phi}{k_p + 2k_f - (k_p - k_f)\phi}\quad ,\Psi = 1\ (n = 3)\ and\ \phi = 0.08$$

$$k_{nf} = 0.772\frac{W}{m.K}$$

a.

Constant heat flux:

Nu_{nf} = 4.36

$$h_{nf} = \frac{Nu_{nf}k_{nf}}{di} = 673.184\frac{W}{m^2.K}$$

Seider-Tate Correlation

$$\frac{h_{nf}}{h_f} = \left(\frac{\rho_{nf}c_{p,nf}}{\rho_f c_{p,f}}\right)^{\frac{1}{3}} * \left(\frac{k_{nf}}{k_f}\right)^{\frac{2}{3}}$$

The Sieder-Tate Correlation neglecting the variation of viscosity enhancement of the fluid with temperature.

$$\rho_{nf} = \phi\rho_p + (1-\phi)\rho_f = 1631.534\frac{kg}{m^3}$$

$$c_{p,nf} = \phi\rho_p c_{p,p} + (1-\phi)c_{p,f} = 3874.528\frac{J}{kg.K}$$

$$enhancement;\ \frac{h_{nf}}{h_f} = \left(\frac{1631.534*3874.528}{996.016*4178}\right)^{\frac{1}{3}} * \left(\frac{0.772}{0.613}\right)^{\frac{2}{3}} = 1.341$$

$$Nu_{nf} = 4.36;\ h_{nf} = 673.184\frac{W}{m^2.K} \Rightarrow\ h_f = 502.002\frac{W}{m^2.K}$$

$$Nu_{nf} = 3.36;\ h_{nf} = 518.784\frac{W}{m^2.K} \Rightarrow\ h_f = 386.8635\frac{W}{m^2.K}$$

Problem 5.9

Repeat the Problem 5.6 with the Al_2O_3/ethylene glycol nanofluid flows in the tube, calculate heat transfer coefficients with
a. 0.05% of particle volume fraction and sphericity value equals to 1.
b. 0.05% of particle volume fraction and sphericity value equals to 3.

GIVEN

- -nanofluid: Al_2O_3/ethylene glycol
- -Nu = 4.36; constant heat flux boundary condition
- -Nu = 3.66; constant wall temperature boundary condition
- -temperature = 40°C
- -tube diameter = 0.5 cm

FIND

a. The heat transfer coefficient; 0.05% of particle volume fraction and sphericity value equals to 1.
b. The heat transfer coefficient;0.05% of particle volume fraction and sphericity value equals to 3.

SOLUTION:

$$h_{nf} = \frac{k_{nf} * Nu_{nf}}{d}$$
$$h_f = \frac{k_f * Nu_f}{d}$$

Thermal conductivity of ethylene glycol k_f = 0.258 W/m.K
Thermal conductivity of Al_2O_3 k_p = 30 W/m.K

a.

$$\frac{k_{nf}}{k_f} = \frac{k_p + 2k_f + 2(k_p - k_f)\phi}{k_p + 2k_f - (k_p - k_f)\phi} \quad ,\Psi = 1\ (n = 3)\ and\ \phi = 0.05$$
$$\frac{k_{nf}}{k_f} = 1.1537; \quad k_{nf} = 0.298\ \frac{W}{m.K}$$
$$h_{nf} = \frac{Nu_{nf}.k_{nf}}{d}$$
$$Nu_{nf} = 4.36;\ h_{nf} = \frac{4.36 * 0.298}{0.005} = 259.856\ \frac{W}{m^2.K}$$
$$Nu_{nf} = 3.66;\ h_{nf} = \frac{3.66 * 0.298}{0.005} = 218.136\ \frac{W}{m^2.K}$$

b.

$$\frac{k_{nf}}{k_f} = \frac{k_p}{k_p - (k_p - k_f)\phi} \quad ,\Psi = 3\ (n = 1)\ and\ \phi = 0.05$$
$$\frac{k_{nf}}{k_f} = 1.0522;\ k_{nf} = 0.271\ \frac{W}{m.K}$$

$$Nu_{nf} = 4.36;\ h_{nf} = \frac{4.36 * 0.271}{0.005} = 236.312\ \frac{W}{m^2.K}$$

$$Nu_{nf} = 3.66;\ h_{nf} = \frac{3.66 * 0.271}{0.005} = 198.372\ \frac{W}{m^2.K}$$

Problem 5.10

Determine the local heat transfer coefficient at 30 cm from the entrance of a heat exchanger where engine oil flows through tubes with a diameter of 0.5 in. The oil flows with a velocity of 0.5 m/s and a local bulk temperature of 30°C, while the local tube wall temperature is at 60°C. If the oil is replaced with the nanofluid of Al_2O_3/oil with 5% particle volume fraction under the same condition, find the heat transfer enhancement.

GIVEN

-diameter = 0.5 in
-oil velocity = 0.5 m/s
-particle volume fraction = 5%
-local tube wall temperature = 60 ^oC
-bulk temperature = 30 ^oC
-Heat transfer coefficient for water (h_{water}) = 3850 W/m^2.K

FIND

the heat transfer enhancement

SOLUTION:

Properties of Oil are

$$\rho = 882.3 \text{ kg/m}^3$$
$$\mu_b = 0.416 \text{ Pa}\cdot\text{s}$$
$$\mu_w = 0.074 \text{ Pa}\cdot\text{s}$$
$$c_p = 1.922 \text{ kJ/(kg}\cdot\text{K)}$$
$$\text{Pr} = 5550$$

$$Re_{f,b} = \frac{\rho u_m d_i}{\mu} = \frac{882.3 \times 0.5 \times 0.0127}{0.416} = 13.47$$

From the Sieder-Tate Correlation;

$$Nu_f = 1.86\left(Pe\frac{d}{L}\right)^{1/3}\left(\frac{\mu_{f,b}}{\mu_{f,w}}\right) = 34.8$$

$$h_f = \frac{k_f \times Nu_f}{d} = \frac{34.8 \times 0.144}{0.0127} = 394.6 \text{ W/m}^2\cdot\text{K}$$

Properties of Al_2O_3 are

$$k_p = 30 \ \mathrm{W/(m \cdot K)}$$
$$\phi = 0.05$$
$$\rho_p = 4000 \ \mathrm{kg/m^3}$$
$$c_{p,p} = 600 \ \mathrm{J/kg \cdot K}$$

The nanofluid properties can be obtained from

$$\begin{aligned} k_{nf} &= \frac{k_p + (n-1)k_f - (n-1)\phi(k_f - k_p)}{k_p + (n-1)k_f + \phi(k_f - k_p)} k_f \\ &= \frac{(30) + (3-1)(0.144) - (3-1)(0.05)(0.144 - 30)}{(30) + (3-1)(0.144) + (0.05)(0.144 - 30)} (0.144) \\ &= 0.166 \ \mathrm{W/(m \cdot K)} \end{aligned}$$

$$c_{p,nf} = \phi c_{p,p} + (1-\phi) c_{p,f} = 1855.9 \ \mathrm{J/kg \cdot K}$$

$$\rho_{nf} = \phi \rho_p + (1-\phi)\rho_f = 1038.185 \ \mathrm{kg/m^3}$$

$$\mu_{nf,b} = \left(1 + 7.3\phi + 123\phi^2\right) \mu_{f,b} = 0.696 \ \mathrm{Pa \cdot s}$$
$$\mu_{nf,w} = \left(1 + 7.3\phi + 123\phi^2\right) \mu_{f,w} = 0.124 \ \mathrm{Pa \cdot s}$$

$$Nu_{nf} = \left(\frac{Pe_{nf}}{Pe_f}\right)^{1/3} = \left(\frac{k_f \rho_{nf} c_{p,nf}}{k_{nf} \rho_f c_{p,f}}\right)^{1/3} = 342.68$$

$$\frac{Nu_{nf}}{Nu_f} = \left(\frac{k_f \rho_{nf} c_{p,nf}}{k_{nf} \rho_f c_{p,f}}\right)^{1/3} \left(\frac{\mu_{nf,b}/\mu_{nf,w}}{\mu_{f,b}/\mu_{f,w}}\right)^{0.14} = 1.07$$

$$Nu_{nf} = 34.8 \times 1.07 = 37.236$$

$$h_{nf} = \frac{37.236 \times 0.166}{0.0127} = 486.71 \ \mathrm{W/m^2 \cdot K}$$

Problem 5.11

Compare the Nusselt number values obtained from Eqs. (5.96) and (5.99), when the nanofluid, Al_2O_3/water, flows with the Reynolds and Prandtl numbers equal to 10000 and 7, respectively, and 1% volume fraction.

GIVEN

- $Re_{nf} = 10000$
- $Pr_{nf} = 7$
- particle volume fraction = 1%

FIND

Compare the Nusselt number values obtained from Eqs. (5.96) and (5.99)

SOLUTION:

From Eq. (5.96)

$$\begin{aligned} Nu_{nf} &= 0.021 Re_{nf}^{0.8} Pr_{nf}^{0.5} \\ &= (0.021)(10000)^{0.8}(7)^{0.5} \\ &= 88.058 \end{aligned}$$

From Eq. (5.99)

$$\begin{aligned} Nu_{fd,nf} &= 0.085 Re_{nf}^{0.71} Pr_{nf}^{0.35} \\ &= (0.085)(10000)^{0.71}(7)^{0.35} \\ &= 116.199 \end{aligned}$$

Problem 5.12

Using the experimental correlation with nanofluids given in Eq. (5.99) for turbulent flow conditions, calculate the heat transfer coefficient under the same nanofluid conditions as in Example 5.16 and compare the heat transfer coefficients from both cases.

GIVEN

-Eq. (5.99); $Nu_{fd,nf} = 0.085{Re_{nf}}^{0.71}{Pr_{nf}}^{0.35}\begin{cases}10000 < Re < 500000\\ 6.6 < Pr < 13.9\\ \phi < 10\%\end{cases}$

FIND

the heat transfer coefficient

SOLUTION:

$\Delta T_1 = \Delta T_2 = \Delta T_m = 105°C = 378K$

$$T_{bulk} = \frac{140+125}{2} = 132℃ = 405K\ , T_{wall} = 105℃ = 378K$$

Base fluid properties at wall temperature:

$$\rho = \frac{1}{1.047 * 10^{-3}} = 955.11\frac{kg}{m^3},$$

$$\mu = 268 * 10^{-6}\ Pa.s, k = 682 * 10^{-3}\ \frac{W}{m.K}$$

Base fluid properties at bulk temperature:

$$\rho = \frac{1}{1.072 * 10^{-3}} = 932.836\frac{kg}{m^3}\ , \mu = 209 * 10^{-6}\ Pa.s,$$

$$k = 688 * 10^{-3}\frac{W}{m.K}\ , c_p = 4271\frac{J}{kg.K}$$

Nanofluid properties:

$$\mu_{nf,b} = (1 + 7.3\phi + 123\phi^2)\mu_{f,b} = 3.496 * 10^{-4} Pa.s$$

$$\mu_{nf,w} = (1 + 7.3\phi + 123\phi^2)\mu_{f,w} = 4.48 * 10^{-4} Pa.s$$

$$(\dot{m} * c_p)_h * \Delta T_h = (\dot{m} * c_p)_c * \Delta T_c$$

$$\Delta T_h = \Delta T_c => \dot{m}_{h,nf} = \frac{\dot{m}_c * c_{p,c}}{c_{p,h}}$$

$$c_{p,c} = 4179\frac{J}{kg.K}$$

$$c_{p,h} = c_{p,nf} = \phi c_{p,p} + (1-\phi)c_{p,f} => c_{p,nf} = 4087.45\frac{J}{kg.K}$$

$$\dot{m}_{h,nf} = 1.42\frac{kg}{s}$$

$$Re_b = \frac{4\dot{m}_{nf}}{\pi\mu_{nf}di} = 98557.11 > 2300\ \ Turbulant\ flow$$

$$k_{static} = \frac{30 + 2 * 0.688 - 2 * 0.05 * (0.688 - 30)}{30 + 2 * 0.688 + 0.05 * (688 - 30)} * 0.688 = 0.7891\frac{W}{m.K}$$

$k_{nf} = k_{static} = 0.7891$ W/m.K

$$Pr_{nf} = \frac{c_{p,nf}\mu_{nf}}{k_{nf}} = 1.81$$
$$Nu_{fd,nf} = 0{,}085 * Re_{nf}^{0.71} * Pr_{nf}^{0.35}$$
$$Nu_{fd,nf} = 0{,}085 * 98557.11^{0.71} * 1.81^{0.35} = 540.2763$$

$$h_{nf} = \frac{k_{nf} * Nu_{nf}}{d} = \frac{540.2763 * 0.7891}{0.0525} = 8120.61\ \frac{W}{m^2.K}$$

Problem 6.1

The heat transfer coefficient of a steel (*k* = 43 W/m · K) tube (1.9 cm ID and 2.3 cm OD) in a shell-and-tube heat exchanger is 500 W/m^2 · K on the inside and 120 W/m^2 · K on the shell side, and it has a deposit with a total fouling factor of 0.000176 m^2 · K/W. Calculate

a. The overall heat transfer coefficient

b. The cleanliness factor, and percent over surface

GIVEN:

-Tube geometry: (d_i) = 1.9 cm

(d_o) = 2.3 cm

-Inside heat transfer coefficient (h_i) = 500 W/m^2.K

-Outside heat transfer coefficient (h_o) = 120 W/m^2.K

-Thermal conductivity of steel tube (k) = 43 W/m.K

-Total fouling factor (R_{ft}) = 0.000176 m^2.K/W

FIND:

a. Overall heat transfer coefficient (U)

b. Cleanliness factor (CF), and percent over surface (OS)

SOLUTION:

a.

The overall heat transfer coefficient is:

$$U_f = \left[\frac{d_o}{d_i}\frac{1}{h_i} + R_{ft} + \frac{d_o \ln(d_o/d_i)}{2k} + \frac{1}{h_o}\right]^{-1}$$

$$U_f = \left[\frac{2.3}{1.9}\frac{1}{500} + 0.000176 + \frac{0.023\ln(2.3/1.9)}{2\times 43} + \frac{1}{120}\right]^{-1}$$

$$= 91.06 \ \ W/m^2.K$$

b.

Clean surface overall heat transfer coefficient is:

$$\frac{1}{U_c} + R_{ft} = \frac{1}{U_f}$$

$$U_c = \left(\frac{1}{U_f} - R_{ft}\right)^{-1} = \left(\frac{1}{91.06} - 0.000176\right)^{-1} = 92.54 \ \ W/m^2 K$$

Cleanliness factor:

$$CF = \frac{U_f}{U_c} = \frac{91.06}{92.54} = 0.984$$

Percent over surface:

$$OS = 100 \cdot U_c \cdot R_{ft} = 100 \times 92.54 \times 0.000176 = 1.63$$

Problem 6.2

Consider a shell-and-tube heat exchanger having one shell and four tube passes. The fluid in the tubes enters at 200°C and leaves at 100°C. The temperature of the fluid is 20°C entering the shell and 90°C leaving the shell. The overall heat transfer coefficient based on the clean surface area of 12 m^2 is 300 $W/m^2.K$. During the operation of this heat exchanger for six months, assume that a deposit with a fouling factor of 0.000528 $m^2.K/W$ has built up. Estimate the percent decrease in the heat transfer rate between the fluids.

GIVEN:

-A shell-and-tube heat exchanger

-Hot fluid inlet temperature (T_{h1}) = 200°C

-Hot fluid outlet temperature (T_{h2}) = 100°C

-Cold fluid inlet temperature (T_{c1}) = 20°C

-Cold fluid outlet temperature (T_{c2}) = 90°C

-Clean surface area (A_o) = 12 m^2

-Overall heat transfer coefficient (U_c) = 300 $W/m^2.K$

-Fouling factor (R_{ft}) = 0.000528 $m^2.K/W$

FIND:

Percent decrease in the heat transfer rate between the fluids.

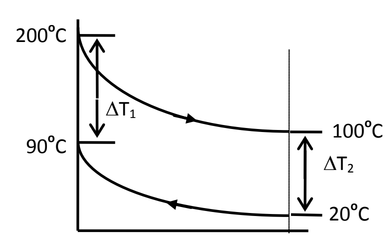
SOLUTION:

The fouled surface overall heat transfer coefficient is:

$$U_f = \left[\frac{d_o}{d_i}\cdot\frac{1}{h_i} + \frac{d_o}{d_i}\cdot R_{fi} + \frac{d_o \ln(d_o/d_i)}{2k} + \frac{1}{h_o}\right]^{-1}$$

$$U_f = \left(\frac{1}{U_c} + R_{ft}\right)^{-1} = \left(\frac{1}{300} + 0.000528\right)^{-1} = 258.97 \;\; W/m^2K$$

Log-mean temperature difference:



$$\Delta T_{lm,cf} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} = \frac{72.3 - 60}{\ln \frac{72.3}{60}} = 65.96 \ \ ^oC$$

Heat transfer rate (assume the same heat transfer area before and after fouling):

$$Q_c = U_f A_f F \Delta T_{lm,cf}$$

$$A_f = \frac{Q_f}{U_f F \Delta T_{lm,cf}} = \frac{15682.5}{100.78 \times 0.83 \times 65.96} = 2.84 \ \ m^2$$

The percent decrease in the heat transfer rate is:

$$CF = \frac{U_f}{U_c} = \frac{100.78}{102.6} = 0.98$$

Problem 6.3

Assume the water for a boiler is preheated using flue gases from the boiler stack. The flue gases are available at the rate of 0.25 kg/s at 150°C, with a specific heat of 1000 J/kg.K. The water entering the exchanger at 15°C at the rate of 0.05 kg/s is to be heated to 90°C. The heat exchanger is to be of the type with one-shell pass and four-tube passes. The water flows inside the tubes, which are made of copper (2.5 m I.D., 3.0 cm O.D.). The heat transfer coefficient on the gas side is 115 W/m^2.K, while the heat transfer coefficient on the water side is 1150 W/m^2.K. A scale on the water side and gas side offer an additional total thermal resistance of 0.000176 m^2.K/W.

- **a. Determine the overall heat transfer coefficient based on the outer tube diameter.**
- **b. Determine the appropriate mean temperature difference for the heat exchanger.**
- **c. Estimate the required tube length.**
- **d. Calculate the percent over surface design and the cleanliness factor.**

GIVEN:

-A shell-and-tube heat exchanger

-Hot fluid inlet temperature (T_{h1}) = 150°C

-Mass flow rate of hot flue gas ($\dot{m}_h$) = 0.25 kg/s

-Specific heat of hot gas ($c_{p,h}$) = 1000 J/kg.K

-Cold fluid inlet temperature (T_{c1}) = 15°C

-Cold fluid outlet temperature (T_{c2}) = 90°C

-Mass flow rate of cold water ($\dot{m}_c$) = 0.05 kg/s

-Heat transfer coefficient on gas side (h_o) = 115 W/m^2.K

-Heat transfer coefficient on water side (h_i) = 1150 W/m^2.K

-Fouling factor (R_f) = 0.000176 m^2.K/W

-Tube diameter (d_i) = 2.5 cm

(d_o) =3.0 cm

FIND:

(1) Overall heat transfer coefficient based on the outer tube diameter (U_o)

(2) Mean temperature (ΔT_m)

(3) Tube length (L)

(4) Percent over surface design (OS) and the cleanliness factor(CF).

SOLUTION:

Thermal conductivity of copper: k = 360 W/m.K

a.

The overall heat transfer coefficient is:

$$U_f = \left[\frac{d_o}{d_i}\cdot\frac{1}{h_i} + R_{ft} + \frac{d_o \ln(d_o/d_i)}{2k} + \frac{1}{h_o}\right]^{-1}$$

$$U_f = \left[\frac{3}{2.5}\cdot\frac{1}{1150} + 0.000176 + \frac{0.03\ln(3/2.5)}{2\times 360} + \frac{1}{115}\right]^{-1} = 100.78 \ \ W/m^2K$$

b.

Properties of water at $\frac{15+90}{2} = 52.5^o\,C = 326\ \ K$:

$c_{p,c}$ = 4182 J/kg.K

Heat duty:

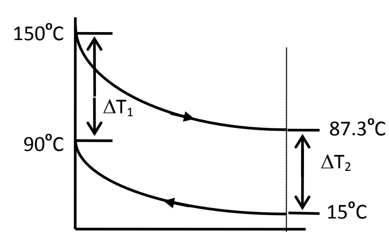
$$Q = \dot{m}_c c_{p,c}(T_{c2} - T_{c1}) = \dot{m}_h c_{p,h}(T_{h1} - T_{h2})$$

$$Q = 0.05 \times 4182 \times (90 - 15) = 15682.5 \ \ W$$

$$T_{h2} = T_{h1} - \frac{Q}{\dot{m}_h c_{p,h}} = 150 - \frac{15682.5}{0.25\times 1000} = 87.3 \ \ ^oC$$

Log-mean temperature difference:

$$\Delta T_{lm,cf} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} = \frac{72.3 - 60}{\ln\frac{72.3}{60}} = 65.96 \ \ ^oC$$



c.

For one shell and 4-tube passes, from figure (2.7):

$$P = \frac{T_{c2} - T_{c1}}{T_{h1} - T_{c1}} = \frac{90 - 15}{150 - 15} = 0.556, \qquad R = \frac{T_{h1} - T_{h2}}{T_{c2} - T_{c1}} = \frac{150 - 87.3}{90 - 15} = 0.836$$

Then, F = 0.83

$$Q_c = U_f A_f F \Delta T_{lm,cf}$$

Heat transfer area:

$$A_f = \frac{Q_f}{U_f F\Delta T_{lm,cf}} = \frac{15682.5}{100.78 \times 0.83 \times 65.96} = 2.84 \ \ m^2$$

$$A = \pi d_o L \cdot \frac{N_t}{N_p} \cdot N_p = \pi d_o L N_t$$

$$LN_t = \frac{A}{\pi d_o} = \frac{2.84}{\pi \times 0.03} = 30.13$$

For a number of tubes $N_t = 100$, L = 0.3 m

d.

Clean surface overall heat transfer coefficient:

$$U_c = \left(\frac{1}{U_f} - R_{ft}\right)^{-1} = \left(\frac{1}{100.78} - 0.000176\right)^{-1} = 102.6 \ \ W/m^2.K$$

Percent over surface:

$$OS = 100 U_c R_{ft} = 100 \times 102.6 \times 0.000176 = 1.8$$

Cleanliness factor:

$$CF = \frac{U_f}{U_c} = \frac{100.78}{102.6} = 0.98$$

Problem 6.4

A horizontal shell-and-tube heat exchanger is used to condense organic vapors. The organic vapors condense on the outside of the tubes, while water is used as the cooling medium on the inside of the tubes. The condenser tubes are 1.9 cm O.D., 1.6 cm I.D. copper tubes 2.4 m in length. There are total of 768 tubes. The water makes four passes through the exchanger. Tests data obtained when the unit was first placed into service are as follows:

water rate = 3700 liters/min

inlet water temperature = 29 ^{o}C

outlet water temperature = 49 ^{o}C

organic-vapor condensation temperature = 118 ^{o}C

After three months of operation, another test made under the same conditions as the first (i.e., same water rate and inlet temperature and same condensation temperature) showed that the exit water temperature was 46 ^{o}C.

a. What is the tube-side fluid (water) velocity?

b. By assuming no changes in either the inside heat transfer coefficient or the condensing coefficient, negligible shell-side fouling, and no fouling at the time of the first test, estimate the tube-side fouling coefficient at the time of the second test.

GIVEN:

-A shell-and-tube heat exchanger

-Organic vapor temperature (T_h) = 118^{o}C

-Cold fluid inlet temperature (T_{c1}) = 29^{o}C

-Cold fluid outlet temperature (T_{c2}) = 49^{o}C

-Cold fluid outlet temperature after fouling ($T_{c2,f}$) = 46 ^{o}C

-Mass flow rate of cold water ($\dot{m}_c$) = 3700 liters/min

-Tube diameter (d_i) = 1.6 cm

(d_o) = 1.9 cm

-Tube length (L) = 2.4 m

-# of tubes (N_t) = 768

-Tube-side passes (N_p) = 4

FIND:

a. Fluid velocity in tube (u_i)

b. Fouling coefficient at the time of second test.

SOLUTION:

a.

Flow area:

$$A_c = \frac{\pi d_i^2}{4} \cdot \frac{N_t}{N_p} = \frac{\pi \times (0.016)^2}{4} \cdot \frac{768}{4} = 0.0386 \ m^2$$

Water velocity:

$$u_i = \frac{\dot{V}}{A} = \frac{0.06166}{0.0386} = 1.597 \ m/s$$

b.

Fouled surface heat transfer coefficient is:

$$U_f = \left[\frac{d_o}{d_i} \cdot \frac{1}{h_i} + \frac{d_o}{d_i} \cdot R_{fi} + \frac{d_o \ln(d_o/d_i)}{2k} + \frac{1}{h_o}\right]^{-1}$$

Properties of water at $\frac{29+49}{2} = 39^\circ C = 312 \ K$:

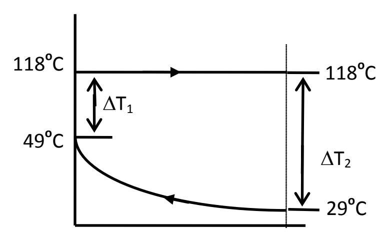
$$\rho = 993 \ kg/m^3 \qquad c_{p,c} = 4178 \ J/kg.K$$

Case 1 (before three month):

$$Q = \dot{m}_c c_{p,c}(T_{c2} - T_{c1}) = 993 \times 0.06166 \times 4178 \times (49 - 29) = 5116.24 \ kW$$

Log-mean temperature difference:

$$\Delta T_{lm,cf} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} = \frac{110 - 80}{\ln\frac{110}{80}} = 94.2 \ ^\circ C$$



$$P = \frac{T_{c2} - T_{c1}}{T_{h1} - T_{c1}} = \frac{49 - 29}{118 - 29} = 0.24, \qquad R = \frac{T_{h1} - T_{h2}}{T_{c2} - T_{c1}} = \frac{0}{20} = 0$$

F = 1

$$A_t = \pi d_o L \cdot \frac{N_t}{N_p} \cdot N_p = \pi d_o L N_t = \pi \times 0.019 \times 2.4 \times 768 = 110.021 \ \ m^2$$

Clean surface overall coefficient:

$$Q_c = U_c A_C F \Delta T_{lm,cf}$$

Heat duty for fouled condition:

$$Q = \dot{m}_c c_{p,c} \left(T_{c2,f} - T_{c1}\right) = 993 \times 0.06166 \times 4178 \times (46-29) = 4348.8 \ \ kW$$

Fouled surface heat transfer coefficient:

$$\Delta T_{lm,cf} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\dfrac{\Delta T_1}{\Delta T_2}\right)} = \frac{(118-46)-(118-29)}{\ln\dfrac{(118-46)}{(118-29)}} = 80.2\,^{o}C$$

$$U_f = \frac{Q_f}{A_t F \Delta T_{lm,cf}} = \frac{4348.8 \times 10^3}{110.021 \times 1 \times 80.2} = 492.86 \ \ W/m^2.K$$

But, $U_f = \left[\dfrac{1}{U_c} + \dfrac{d_o}{d_i} \cdot R_{fi}\right]^{-1}$

$$R_{fi} = \frac{d_i}{d_o}\left[\frac{1}{U_f} - \frac{1}{U_c}\right] = \left(\frac{1.6}{1.9}\right)\left(\frac{1}{502.89} - \frac{1}{492.86}\right) = 0.000285 \ \ m^2.K/W$$

Problem 6.5

In a double pipe heat exchanger, deposits of calcium carbonate with a thickness of 1.12 mm and magnesium phosphate with a thickness of 0.88 mm on the inside and outside of the inner tube respectively have formed over time. Tubes (ID = 1.9 cm, OD = 2.3 cm) are made of carbon steel (k = 43 W/m.K). Calculate the total fouling resistance based on the outside surface area of the heat exchanger.

GIVEN:

-A double pipe heat exchanger

-Tube diameter (d_i) = 1.6 cm

(d_o) = 1.9 cm

-Deposits thickness inside (δ_1) = 1.12 mm

-Deposits thickness outside (δ_2) = 0.88 mm

FIND:

Total fouling resistance based on the outside surface area of the heat exchanger (R_{ft})

SOLUTION:

From Table 6.3:

Calcium carbonate: k_1 = 2.941 W/m.K

Magnesium phosphate k_2 = 2.1625 W/m.K

Total fouling resistance is:

$$R_{ft} = \frac{d_o}{d_i} \cdot R_{fi} + R_{fo}$$

$$R_{fi} = A_i R_1 = \pi d_i L \cdot \frac{\ln\left(\dfrac{d_i}{d_i - 2\delta_1}\right)}{2\pi L k_1}$$

$$R_{fi} = \frac{d_i}{2k_1} \ln\left(\frac{d_i}{d_i - 2\delta_1}\right)$$

$$R_{fi} = \frac{0.019}{2 \times 2.941} \ln\left(\frac{0.019}{0.019 - 2 \times 0.00112}\right) = 0.000405 \ \ m^2.K/W$$

$$R_{fo} = A_o R_2 = \pi d_o L \cdot \frac{\ln\left(\frac{d_o + 2\delta_2}{d_o}\right)}{2\pi L k_2}$$

$$R_{fo} = \frac{d_o}{2k_2} \ln\left(\frac{d_o + 2\delta_2}{d_o}\right)$$

$$R_{fo} = \frac{0.023}{2 \times 2.1625} \ln\left(\frac{0.023 + 2 \times 0.00088}{0.023}\right) = 0.000392 \quad m^2.K/W$$

$$R_{ft} = \frac{2.3}{1.9} \cdot 0.000405 + 0.000392 = 0.000882 \quad m^2.K/W$$

Problem 6.6

In a shell-and-tube heat exchanger, water at a flow rate of 3 kg/s is heated from 20 °C to 90°C. On the shell side stream condenses to heat the water resulting in an overall heat transfer coefficient of 2100 W/m².K. After a period of six months of continuous operation, the temperature of tube inside water drops to 80 °C. Calculate

a. The overall heat transfer coefficient under six months operating conditions

b. The total fouling resistance under operating conditions.

GIVEN:

-A shell-and-tube heat exchanger

-Steam temperature (T_h) = 100°C

-Cold fluid inlet temperature (T_{c1}) = 20°C

-Cold fluid outlet temperature (T_{c2}) = 90°C

-Cold fluid outlet temperature after fouling ($T_{c2,f}$) = 80 °C

-Mass flow rate of cold water ($\dot{m}_c$) = 3 kg/s

-Overall heat transfer coefficient before fouling (U_c) = 2100 W/m².K

FIND:

a. Overall heat transfer coefficient after fouling (U_f)

b. Total fouling resistance under operating conditions (R_t)

SOLUTION:

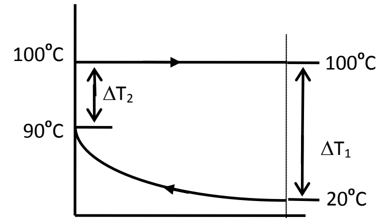
Properties of water at $\frac{20+90}{2} = 55^\circ C = 328 \text{ K}$:

$$\rho = 985 \text{ kg/m}^3 \qquad c_{p,c} = 4182 \text{ J/kg.K}$$

a.

Log-mean temperature difference:

$$\Delta T_{lm,cf} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} = \frac{80-10}{\ln\frac{80}{10}} = 33.7\ ^{o}C$$



Heat duty under clean conditions:

$$Q = \dot{m}_c c_{p,c}\left(T_{c2} - T_{c1}\right) = 3 \times 4182 \times (90-20) = 878.22\ \ kW$$

$$Q_c = U_c AF\Delta T_{lm,cf}$$

Heat transfer area:

$$A = \frac{Q_c}{U_c F\Delta T_{lm,cf}} = \frac{878.22 \times 10^3}{2100 \times 1 \times 33.7} = 12.409\ \ m^2$$

Heat duty under fouled conditions:

$$Q = \dot{m}_c c_{p,c}\left(T_{c2,f} - T_{c1}\right) = 3 \times 4182 \times (80-20) = 752.76\ \ kW$$

$$\Delta T'_{lm,cf} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} = \frac{80-20}{\ln\frac{80}{20}} = 43.3\ ^{o}C$$

Fouled surface overall heat transfer coefficient is:

$$U_f = \frac{Q_f}{A\Delta T'_{lm,cf}} = \frac{752.76 \times 10^3}{12.409 \times 43.3} = 1400.98\ W/\ m^2.K$$

b.

The total fouling resistance is:

$$R_{fi} = \left[\frac{1}{U_f} - \frac{1}{U_c}\right] = \left(\frac{1}{1400.98} - \frac{1}{2100}\right) = 0.000238\ \ m^2.K/W$$

Problem 6.7

In problem 6.1, if shell-side fluid is refined tube oil and tube fluid is sea water with a velocity of 2m/s, what value of the overall heat transfer coefficient should be used for design proposes if the shell-side heat transfer coefficient remains unchanged? Calculate the oversurface design. Is this acceptable?

GIVEN:

-Sea water flowing in tube

-Sea water velocity u_m = 2 m/s

-Other conditions are the same as those in Problem 6.1

FIND:

a. Overall heat transfer coefficient;

b. Percentage over surface.

SOLUTION:

The sea water properties:

$$\rho = 997.207 \ \ \text{kg/m}^3 \qquad k = 0.605\text{w/m.K}$$
$$\mu = 9.09\times10^{-4} \ \ \text{Pa.s} \qquad \Pr = 6.29$$

Thermal conductivity of steel: k_s = 43 W/m.K;

$$\text{Re} = \frac{\rho u_m d_i}{\mu} = \frac{997.207\times2\times0.019}{9.09\times10^{-4}} = 41687.4 \ \text{(Turbulent flow)}$$

The heat transfer coefficient inside the tube can be calculated by Equation (3.31):

$$f = (1.58\ln\text{Re} - 3.28)^{-2}$$
$$= [1.58\times\ln(41687.4) - 3.28]^{-2} = 0.00546$$

$$\text{Nu}_b = \frac{(f/2)(\text{Re}_b - 1000)\Pr}{1+12.7(f/2)^{0.5}(\Pr_{b0}{}^{2/3}-1)}$$
$$= \frac{(0.00546/2)\cdot(41687.4-1000)\cdot6.29}{1+12.7\times(0.00546/2)^{0.5}(6.29^{2/3}-1)}$$
$$= 269$$

$$h_i = \frac{Nu_b k}{d_i} = \frac{269 \times 0.605}{0.019} = 8564.8 \ \ W/m^2 \cdot K$$

The overall heat transfer coefficient is:

$$\frac{1}{U_f} = \frac{d_o}{d_i}\frac{1}{h_i} + R_{ft} + \frac{d_o \ln(d_o/d_i)}{2k_s} + \frac{1}{h_o} = \frac{1}{2445} + R_{ft} + \frac{1}{8}$$

$$U_f = \left[\frac{d_o}{d_i}\frac{1}{h_i} + R_{ft} + \frac{d_o \ln(d_o/d_i)}{2k_s} + \frac{1}{h_o} = \frac{1}{2445} + R_{ft} + \frac{1}{8}\right]^{-1}$$

$$= \left[\frac{0.023}{0.019}\frac{1}{8564.8} + 0.000176 + \frac{0.023 \ln(0.023/0.019)}{2 \times 43} + \frac{1}{120} = \right]^{-1}$$

$$= 114.9 \ \ W/m^2 \cdot K$$

Clean surface overall heat transfer coefficient is:

$$\frac{1}{U_f} = \frac{1}{h_i} + R_{ft} + \frac{1}{h_o} = \frac{1}{2445} + R_{ft} + \frac{1}{8}$$

Percentage over surface is:

$$OS = 100\ U_c R_{ft} = 100 \times 117.3 \times 0.000176 = 2.06$$

Problem 6.8

A counterflow doublepipe heat exchanger is designed to cool lubricating oil for a large industrial gas turbine engine. The flow rate of cooling water through the inner tube is 0.2 kg/s, and the water enters the tubes at 20°C, while the flow rate of oil through the outlet annulus is 0.4 kg/s. The oil and water enter the heat exchanger at temperatures of 60°C and 30°C, respectively. The heat transfer coefficients in the annulus and in the inner tube have been estimated as 8 W/m^2 · K and 2445 W/m^2 · K, respectively. The outer diameter of the inner tube is 25 mm, and the inner diameter of the outer tube is 45 mm. The total length of the double-pipe heat exchanger (total length per hairpin) is 15 m and total heat exchanger area is 325 m^2. In the analysis, the tube wall resistance and the curvature of the wall are neglected. What is the total value of the fouling resistance used in this design?

GIVEN:

-A double pipe heat exchanger

-Cold fluid inlet temperature (T_{c1}) = 20^{o}C

-Mass flow rate of cold water ($\dot{m}_c$) = 0.2 kg/s

-Cold fluid inlet temperature (T_{h1}) = 60^{o}C

-Cold fluid outlet temperature (T_{h2}) = 30^{o}C

-Mass flow rate of oil ($\dot{m}_h$) = 0.4 kg/s

-Heat transfer coefficient in annulus (h_o) = 8 W/m^2.K

-Heat transfer coefficient in inner tube (h_i) = 2445 W/m^2.K

-Outer diameter of inner tube (d_o) = 25 mm

-Inner diameter of outer tube (D_i) = 45 mm

-Total length per hairpin (L) = 15 m

- Total heat exchanger area (A) = 325 m^2

FIND:

Total fouling resistance (R_{ft})

SOLUTION:

Specific heat for lubricating oil: $c_{p,h}$ = 2006 J/kg.K, for water: $c_{p,c}$ = 4178 J/kg.K.

The fouled surface overall heat transfer coefficient is:

$$\frac{1}{U_f} = \frac{1}{h_i} + R_{ft} + \frac{1}{h_o} = \frac{1}{2445} + R_{ft} + \frac{1}{8}$$

$$\Rightarrow \ R_{ft} = \frac{1}{U_f} - \frac{1}{h_i} - \frac{1}{h_o} = \frac{1}{U_f} - 0.125409$$

Heat duty for fouled condition:

$$Q = \dot{m}_c c_{p,c}\left(T_{c2} - T_{c1}\right) = \dot{m}_h c_{p,h}\left(T_{h1} - T_{h2}\right) = 0.4 \times 2006 \times (60 - 30) = 24072 \ \ \text{W}$$

$$T_{c2} = T_{c1} + \frac{\dot{m}_h c_{p,h}\left(T_{h1} - T_{h2}\right)}{\dot{m}_c c_{p,c}}$$

$$= 20 + \frac{0.4 \times 2006 \times (60 - 30)}{0.2 \times 4178} = 48.8 \ {}^{o}\text{C}$$

$$\Delta T_m = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\Delta T_1 / \Delta T_2\right)}$$

$$= \frac{(60 - 48.8) - (30 - 20)}{\ln\left(\dfrac{60 - 48.8}{30 - 20}\right)} = 10.58 \ \ {}^{o}\text{C}$$

$$Q = U_f A \ \Delta T_m \ \text{cut} \ = U_f \left(\pi d_o L\right) \Delta T_m \ \text{out}$$

$$U_f = \frac{Q}{A\Delta T_m} = \frac{24076}{325 \times 10.58} = 7.002 \ \ \text{W/m}^2 \cdot \text{K} \ \text{corrected answer}$$

$$R_{ft} = \frac{1}{U_f} - \frac{1}{h_i} - \frac{1}{h_o} = \frac{1}{U_f} - 0.125409 = \frac{1}{7.002} - 0.125409 = 0.01740734 \ \text{corr}$$

ected answer

Problem 6.9

A shell-and-tube type condenser with one shell pass and four tube passes is used to condense organic vapor. The condensation occurs on the shell side, while the coolant water flows inside the tubes that are 1.9-cm O.D. and 1.6-cm I.D. copper tubes. The length of the heat exchanger is 3 m long. The total number of tubes is 840. The initial data of the condenser are recorded as:

Water rate, 70 kg/s

Water inlet temperature, 20 ^{o}C

Water outlet temperature, 45 ^{o}C

Condensation temperature, 105 ^{o}C

After 4 months of operation, under the same conditions, the exit temperature of water drop to 40 ^{o}C. By assuming shell-side fouling is negligible, there is no fouling at the time of the first operation, and the inside and outside heat transfer coefficients are unchanged, estimate the tube-side fouling factor after the operation of 4 months.

GIVEN:

-A shell-and-tube heat exchanger

-Water inlet temperature (T_{c1}) = 20^{o}C

-Water outlet temperature (T_{c2}) = 45^{o}C

-Steam temperature (T_h) = 105 ^{o}C

-Mass flow rate of cold water ($\dot{m}_c$) = 70 kg/s

-Outer diameter of tube (d_o) = 1.9 cm

-Inner diameter of tube (d_i) = 1.6 cm

-Total length of tube (L) = 3 m

-After four months, (T_{c2}) = 40 ^{o}C

FIND:

Total fouling resistance (R_{ft})

SOLUTION:

Specific heat for water at mean temperature $T_m = \frac{T_{c1}+T_{c2}}{2} = \frac{35+95}{2} = 65\ ^oC$:

$$c_{p,c} = 4178 \text{ J/kg.K}$$

Then:

$$Q = \dot{m}_c c_{p,c}\left(T_{c2} - T_{c1}\right) = 70 \times 4178 \times (45-20) = 7311.9 \text{ k W}$$

The total heat transfer area A_o is:

$$A_o = \pi d_o L \cdot N = \pi \times 0.019 \times 3 \times 840 = 150.3 \ \ m^2$$

The heat exchanger log mean temperature is:

$$\Delta T_{lm,cf} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\Delta T_1 / \Delta T_2\right)}$$

$$= \frac{(105-20)-(105-45)}{\ln\left(\frac{105-20}{105-45}\right)} = 71.8 \ \ ^oC$$

According to the heat transfer equation:

$$Q = U_c A_o \Delta T_m = U_c\left(\pi d_o L\right)\Delta T_m$$

$$U_c = \frac{Q}{\left(\pi d_o L\right)\Delta T_m} = \frac{7311.9\times 10^3}{150.3\times 71.8} = 677.1 \text{ W/m}^2\cdot\text{K}$$

After 4 months of operation, the outlet water temperature is decreased to 40 ^oC, so the mean temperature:

$$T_m = \frac{T_{c1}+T_{c2}'}{2} = \frac{35+90}{2} = 62.5\ ^oC$$

$$Q_f = \dot{m}_c c_{p,c}\left(T_{c2,f} - T_{c1}\right) = 70\times 4179\times(40-20) = 5850.6 \text{ k } W$$

$$\Delta T_{lm,cf} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\Delta T_1 / \Delta T_2\right)}$$

$$= \frac{(105-20)-(105-40)}{\ln\left(\frac{105-20}{105-40}\right)} = 74.6 \ \ ^oC$$

$$Q_f = U_f A_o \Delta T_m = U_f\left(\pi d_o L\right)\Delta T_m$$

$$U_f = \frac{Q_f}{\left(\pi d_o L\right)\Delta T_m} = \frac{5850.6\times 10^3}{150.4\times 74.6} = 521.5 \ \ \text{W/m}^2\cdot\text{K}$$

Because the shell-side fouling is negligible, so the tube-side fouling factor is:

$$R_{ft} = \frac{1}{U_f} - \frac{1}{U_c} = \frac{1}{521.5} - \frac{1}{677.1} = 0.00044 \ \ m^2\cdot K/W$$

Problem 6.10

In a single-phase double-pipe heat exchanger, water is to be heated from 35 to 95 ^oC. The water flow rate is 4 kg/s. Condensing steam at 200 ^oC in the annulus is used to heat the water. The overall heat transfer coefficient used in the design of this heat exchanger is 3500 $W/m^2.K$. After an operation of 6 months, the outlet temperature of the hot water drops to 90 ^oC. The maintaining of the outlet temperature of the water is essential for the purpose of this heat exchanger; therefore fouling is not acceptable. Calculate the total fouling factor under these operations and comment if the cleaning cycle must be extended.

GIVEN:

-A double-pipe heat exchanger

-Water inlet temperature (T_{c1}) = 35^oC

-Water outlet temperature (T_{c2}) = 95^oC

-Water outlet temperature after 6 months of fouling (T_{c2}') = 90 ^oC

-Steam temperature (T_h) = 200 ^oC

-Mass flow rate of cold water ($\dot{m}_c$) = 4 kg/s

-Heat transfer coefficient for clean surface (U_c) = 3500 $W/m^2.K$

FIND:

Total fouling resistance (R_{ft})

SOLUTION:

When the surface is clean:

Specific heat for water at mean temperature $T_m = \frac{T_{c1} + T_{c2}}{2} = \frac{35+95}{2} = 65\ ^oC$:

$$c_{p,c} = 4185 \text{ J/kg.K}$$

Then:

$$Q = \dot{m}_c c_{p,c}\left(T_{c2} - T_{c1}\right) = 4 \times 4185 \times (95-35) = 1004.4 \text{ k } W$$

The heat exchanger log mean temperature is:

$$\Delta T_{lm,cf} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\Delta T_1/\Delta T_2\right)}$$

$$= \frac{(200-35)-(200-95)}{\ln\left(\dfrac{200-35}{200-95}\right)} = 132.7 \quad {}^{o}C$$

The total heat transfer area A_o is:

$$A_o = \frac{Q}{U_c \Delta T_m} = \frac{1004.4\times10^3}{3500\times132.7} = 2.16 \ \ m^2$$

After 6 months of operation, the outlet water temperature is decreased to 90 ^{o}C, so the mean temperature:

$$T_m = \frac{T_{c1} + T_{c2}^{'}}{2} = \frac{35+90}{2} = 62.5 \ {}^{o}C$$

and,

$$c_{p,c} = 4185 \text{ W/kg.K}$$

$$\Delta T_{lm,cf} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\Delta T_1/\Delta T_2\right)}$$

$$= \frac{(200-35)-(200-90)}{\ln\left(\dfrac{200-35}{200-90}\right)} = 135.6 \quad {}^{o}C$$

$$Q_f = \dot{m}_c c_{p,c}\left(T_{c2}^{'} - T_{c1}\right) = 4\times4185\times(90-35) = 920.7 \text{ k}W$$

$$U_f = \frac{Q_f}{A_o \Delta T_m} = \frac{920.7\times10^3}{2.16\times135.6} = 3143.4 \ \ W/m^2\cdot K$$

$$R_{ft} = \frac{1}{U_f} - \frac{1}{U_c} = \frac{1}{3143.4} - \frac{1}{3500} = 0.000032 \ \ m^2\cdot K/W$$

Because the purpose of the heat exchanger is to maintain the outlet temperature of water at 95 ^{o}C, after 6 months of operation, the outlet temperature is only 90 ^{o}C, so the cleaning cycle must be shortened.

Problem 6.11

Distilled water enters the tubes of a shell-and-tube heat exchanger at 200°C and leaves at 100°C. The objective is to design a heat exchanger to heat city water from 20°C to 90°C on the shell side. The overall heat transfer coefficient based on the outside clean surface area of 12 m² is 1500 W/m².K. Tubes are carbon steel ID = 16 mm and OD = 19 mm. During the operation of this heat exchanger for six months, a deposit builds up on the shell-side only.

Estimate the percent decrease in the heat transfer rate between the fluids.

GIVEN:

-A shell-and-tube heat exchanger, with hot water flowing through tubes.

-Distilled hot water inlet temperture (T_{h1}) = 200°C

-Distilled hot water outlet temperature (T_{h2}) = 100°C

-Cold water inlet temperature (T_{c1}) = 20°C

-Cold water outlet temperature (T_{c2}) = 90°C

-Outside clean surface area (A_c) = 12 m².

-Overall heat transfer coefficient based on outside area (U_o) = 1500 W/m².K

-Tube diameter (d_i) = 16 mm

(d_o) = 19 mm

FIND:

Percent decrease in the heat transfer rate between the fluids.

SOLUTION:

From table 6.11: (for the tube side)

$R_{fi} = 0.176\times10^3\, m^2 \cdot K/W$

$R_{fo} = 0.352\times10^3\, m^2 \cdot K/W$

Overall heat transfer coefficient based on the outside surface of the inner tube, with fouling:

$$U_f = \left[\frac{1}{U_c} + R_{ft}\right]^{-1}$$

Where

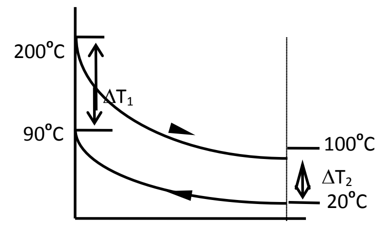
$$R_{ft} = R_{fo} + \frac{d_o}{d_i} \cdot R_{fi} = 0.000352 + \frac{19}{16} \times 0.000176 = 0.000561$$

since the wall resistance is neglected.

Then,

$$U_f = \left[\frac{1}{U_c} + R_{ft} \right]^{-1} = \left[\frac{1}{1500} + 0.000561 \right]^{-1} = 815 \;\; W/m^2.K$$

Log-mean temperature difference:



$$\Delta T_{lm,cf} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} = \frac{110 - 80}{\ln \frac{110}{80}} = 94.2 \;\; ^\circ C$$

$$Q_c = U_c A_C F \Delta T_{lm,cf}$$

$$Q_f = U_f A_f F \Delta T_{lm,cf}$$

Taking the ratio, the decrease in heat transfer rate is obtained:

$$\frac{Q_f}{Q_c} = \frac{U_f A_c F \Delta T_{lm,cf}}{U_c A_C F \Delta T_{lm,cf}} = \frac{U_f}{U_c} = \frac{815}{1500} = 54.33\%$$

Problem 7.1

A counterflow double pipe heat exchanger is used to cool the lubricating oil for a large industrial gas turbine engine. The flow rate of cooling water through the inner tube is m_c=0.2 kg/s, while the flow rate of oil through the outer annulus is m_h=0.4 kg/s. The oil and water enter at temperature of 60 °C and 30 °C, respectively. The heat transfer coefficient in the annulus is calculated to be 15 W/m^2.K. The I.D. of the tube is 25 mm and the I.D. of outer annulus is 45 mm. The outlet temperature of the oil is 40 °C. Take c_p=4178 J/kg.K for water and c_p=2006 J/kg.K for oil. The tube wall resistance and the curvature of the wall are neglected. Calculated the length of the heat exchanger if fouling is neglected.

GIVEN:

-A double pipe heat exchanger, with oil flowing through annulus.

-Oil inlet temperature (T_{h1}) = 60°C

-Oil outlet temperature (T_{h2}) = 40 °C

-Mass flow rate of oil ($\dot{m}_h$) = 0.4 kg/s

-Cooling water inlet temperature (T_{c1}) = 30°C

-Mass flow rate of cooling water ($\dot{m}_c$) = 0.2 kg/s

-Annulus side heat transfer coefficient for oil (h_o) = 15 W/m^2.K

-Inner diameter of tube (d_i) = 25 mm

-Inside diameter of outer annulus (D_i) = 45 mm

-Specific heat of water ($c_{p,c}$) = 4178 J/kg.K

-Specific heat of oil ($c_{p,h}$) = 2006 J/kg.K

FIND:

The length of the heat exchanger (L).

ASSUMPTIONS:

-Fouling, the tube wall resistance, and the curvature of the wall are neglected.

-Variation of thermal properties with temperature is negligible.

-Inner tube wall thickness is assumed to be t = 1 mm.

SOLUTION:

The cooling water outlet temperature can be obtained using the heat balance equation:

$$Q = \dot{m}_c c_{p,c}\left(T_{c2} - T_{c1}\right) = \dot{m}_h c_{p,h}\left(T_{h2} - T_{h1}\right)$$

$$T_{c2} = T_{c1} + \frac{\dot{m}_h c_{p,h}\left(T_{h1} - T_{h2}\right)}{\dot{m}_c c_{p,c}}$$

$$= 30 + \frac{0.4 \times 2006 \times (60 - 40)}{0.2 \times 4178}$$

$$= 49.2\ ^{o}C$$

$$\Delta T_m = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\Delta T_1 / \Delta T_2\right)} = \frac{10.8 - 10}{\ln(10.8/10)} = 10.39\ ^{o}C$$

<u>Tube-side analysis:</u>

Properties of water at $\frac{49.2 + 30}{2} = 39.6\ ^{o}C$ is:

$$\rho = 992\ \text{kg/m}^3 \qquad \mu = 605 \times 10^{-6}\ \text{N.s/m}^2$$

$$\text{Pr} = 4.36 \qquad k = 0.6\ \text{W/m.K}$$

The water velocity can be obtained as:

$$u_m = \frac{\dot{m}_c}{\rho A_c} = \frac{\dot{m}_c}{\rho \dfrac{\pi d_i^2}{4}} = \frac{0.2}{992 \times \dfrac{\pi \times 0.025^2}{4}} = 0.41\ \text{m/s}$$

Therefore, the Reynolds number is:

$$\text{Re} = \frac{\rho u_m d_i}{\mu} = \frac{992 \times 0.41 \times 0.025}{605 \times 10^{-6}} = 15198 \quad \text{(Turbulent flow)}$$

From table 3.3, Genielinski's correlation can be used:

$$f = \left(1.58 \ln \text{Re} - 3.28\right)^{-2} = \left(1.58 \times \ln 16807 - 3.28\right)^{-2} = 7.0 \times 10^{-3}$$

$$Nu = \frac{(f/2)\,RePr}{1.07 + 12.7(f/2)^{\frac{1}{2}}\left(Pr^{\frac{2}{3}} - 1\right)}$$

$$= \frac{\left(\frac{0.007}{2}\right)15198 \times 4.44}{1.07 + 12.7 \times \left(\frac{0.0068}{2}\right)^{\frac{1}{2}}\left(4.44^{\frac{2}{3}} - 1\right)}$$

$$=100.57 \ \ W/m^2.K$$

The inside heat transfer coefficient is:

$$h_i = \frac{Nu \cdot k}{d_i} = \frac{100.57 \times 0.6}{0.025} = 2413.68 \ \ W/m^2.K$$

The outside heat transfer coefficient is given, h_o = 15 $W/m^2.K$.

The fouling, tube wall resistance and the curvature of the wall are neglected, so the overall heat transfer coefficient can be calculated as:

$$U_o = \frac{1}{\frac{1}{h_i} + \frac{1}{h_o}} = \frac{1}{\frac{1}{2413.68} + \frac{1}{15}} = 14.9 \ \ W/m^2.K$$

$$A_o = \frac{Q}{U_o \Delta T_m} = \frac{\dot{m}_h c_{p,h}\left(T_{h1} - T_{h1}\right)}{U_o \Delta T_m} = \frac{0.4 \times 2006 \times (60 - 40)}{14.9 \times 10} = 108 \ \ m^2$$

$$A_o = \pi \times d_o \times 2L = 2\pi \times 0.0334 \times 3 = 0.63 \ m^2$$

$$\therefore L = \frac{A_o}{\pi d_o} = \frac{108}{\pi \times 0.025} = 1375 \ \ m$$

Problem 7.2

In problem 7.1, the heat transfer coefficient in the annulus is given as 15 W/m^2.K. Is this value acceptable? Assume the water used is city water with a fouling resistance of 0.000176 m^2.K/W inside the tube. Oil side deposit is neglected. If the length of the hairpin used is 4 m, calculate the heat transfer area and the number of hairpins. If the space to accommodate this heat exchanger is limited, what are the alternatives?

GIVEN:

-Those given in problem 7.1

-Fouling resistance inside the tube (R_{fi}) = 0.000176 m2.K/W

-Length of hairpins (L) = 4 m

FIND:

a. Heat transfer area (A_o)

b. The number of hairpins (N_{hp})

ASSUMPTIONS:

-City water is used.

-Oil side deposit is neglected.

SOLUTION:

a.

The heat transfer coefficient in the annulus is too low at 15 W/m^2.K.

The overall heat transfer coefficient is(neglect the curvature and wall resistance):

$$U_f = \frac{1}{\frac{1}{h_i} + R_{ft} + R_{fo} + \frac{1}{h_o}} = \frac{1}{\frac{1}{2413.68} + 0.000176 + 0 + \frac{1}{15}} = 14.87 \quad W/m^2.K$$

The heat transfer surface area needed:

$$A_f = \frac{Q}{U_f \Delta T_m} = \frac{\dot{m}_h c_{p,h}\left(T_{h2} - T_{h1}\right)}{U_f \Delta T_m} = \frac{0.4 \times 2006 \times (60 - 40)}{14.87 \times 10} = 107.9 \ \ m^2$$

Heat transfer area per hairpin:

$$A_{hp} = 2\pi d_o L = 2\pi \times 0.025 \times 4 = 0.63 \ \ m^2$$

b.

Therefore, the number of hairpins needed is:

$$N_p = \frac{A_o}{A_{hp}} = \frac{107.9}{0.63} = 171.3 \approx 172 \ \ \text{Hairpins.}$$

If the space is limited the number off hairpins can be reduced by the use of finned tubes.

Problem 7.3

Engine oil (raffinate) with a flow rate of 5 kg/s will be cooled from 60°C to 40°C by sea water at 20°C in a double-pipe heat exchanger. The water flows through the inner tube, whose outlet is heated to 30°C. The inner tube outside and inside diameters are d_o = 1.315 inches (= 0.0334 m) and d_i = 1.049 inches (= 0.02664 m), respectively. For the annulus, D_o= 4.5 inches (= 0.1143 m) and D_i = 4.206 inches (= 0.10226 m). The length of the hairpin is fixed at 3 m. The wall temperature is 35°C. The number of the tubes in the annulus is 3. The thermal conductivity of the tube wall is 43 W/m · K. Calculate:

a. The heat transfer coefficient in the annulus;

b. The overall heat tranfer coefficient;

c. The pressure drop in the annulus and the inner tube (only straight sections will be considered);

d. What is your decision as an engineer?

GIVEN:

-A double pipe heat exchanger, with engine oil flowing through annulus.

-Oil inlet temperature (T_{h1}) = 60°C

-Oil outlet temperature (T_{h2}) = 40 °C

-Mass flow rate of oil ($\dot{m}_h$) = 5 kg/s

-Cooling water inlet temperature (T_{c1}) = 20°C

-Cooling water outlet temperature (T_{c2}) = 30°C

-Inner tube diameters: (d_i) = 1.049 inches = 0.02664 m

(d_o) =1.315 inches = 0.0334 m

-Annulus diameters: (D_i) = 4.206 inches = 0.10226 m

(D_o) = 4.5 inches = 0.1143 m

-Length of the hairpin (L) = 3 m

-Wall temperature (T_w) = 35°C

-Thermal conductivity of the tube wall is 43 W/m.K.

-Number of tubes in the annulus (N_t) = 3

FIND:

a.) The heat transfer coefficient in the annulus (h_o)

b. The overall heat transfer coefficient (U_o)

c. The pressure drop in the annulus and the inner tube (only straight sections will be considered)

d. What is your decision as an engineer.

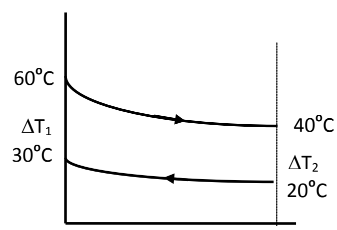
ASSUMPTIONS:

-Fouling, the tube wall resistance, and the curvature of the wall are neglected.

-Variation of thermal properties with temperature is negligible.

SOLUTION:

$$\Delta T_m = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} = \frac{30-20}{\ln\left(\frac{30}{20}\right)} = 24.6\ ^\circ C$$



Properties of water at $\frac{20+30}{2} = 25\ ^\circ C$ is:

$\rho = 997\ kg/m^3$ $\mu = 894 \times 10^{-6}\ N.s/m^2$

$Pr = 6.18$ $k = 0.0606\ W/m.K$

$c_{p,c} = 4180\ J/kg.K$

Properties of engine oil at $\frac{60+40}{2} = 50\ ^\circ C$ is:

$\rho = 870\ kg/m^3$ $\mu = 0.124\ N.s/m^2$

$Pr = 1760$ $k = 0.141\ W/m.K$

$c_{p,c} = 2005\ J/kg.K$

$\mu_w = 0.3\ N.s/m^2$ (at $T_w = 35^\circ C$)

<u>Tube side analysis:</u>

The cooling water mass flow rate can be obtained using the heat balance equation:

$$Q = \dot{m}_c c_{p,c}\left(T_{c2} - T_{c1}\right) = \dot{m}_h c_{p,h}\left(T_{h1} - T_{h2}\right)$$

$$\dot{m}_c = \frac{\dot{m}_h c_{p,h}\left(T_{h1} - T_{h2}\right)}{c_{p,c}\left(T_{c2} - T_{c1}\right)} = \frac{5 \times 2005 \times (60-40)}{4180 \times (30-20)} = 4.8 \text{ kg/s}$$

The water velocity can be obtained as:

$$u_m = \frac{\dot{m}_c}{N_t \rho A_c} = \frac{\dot{m}_c}{N_t \rho \frac{\pi d_i^2}{4}} = \frac{4.8}{3 \times 997 \times \frac{\pi \times 0.02664^2}{4}} = 2.88 \text{ m/s}$$

Therefore, the Renolds number is:

$$Re = \frac{\rho u_m d_i}{\mu} = \frac{997 \times 2.88 \times 0.02664}{894 \times 10^{-6}} = 85{,}562.7 \quad \text{(Turbulent flow)}$$

From table 3.3, Petukhof and Kirillov correlation can be used:

$$f = (1.58 \ln Re - 3.28)^{-2} = (1.58 \times \ln 85562.7 - 3.28)^{-2} = 4.65 \times 10^{-3}$$

$$Nu_b = \frac{(f/2) Re_b Pr_b}{1.07 + 12.7 (f/2)^{1/2}\left(Pr_b^{2/3} - 1\right)}$$

$$Nu_b = \frac{\left(\frac{0.00465}{2}\right) \times 85562.7 \times 6.18}{1.07 + 12.7 \times \left(\frac{0.0037}{2}\right)^{\frac{1}{2}}\left(6.18^{\frac{2}{3}} - 1\right)}$$

$$= 520.2$$

The inside heat transfer coefficient is:

$$h_i = \frac{Nu_b \cdot k}{d_i} = \frac{520.2 \times 0.606}{0.02664} = 2550 \text{ W/m}^2.\text{K}$$

<u>Annulus side analysis:</u>

$$D_e = \left(D_i^2 - N_t d_o^2\right) / N_t d_o = \left(0.10226^2 - 3 \times 0.03340^2\right) / 3 \times 0.03340 = 0.071 \text{ m}$$

$$D_h = D_i - N_t d_o = 0.10226 - 3 \times 0.03340 = 0.035 \text{ m}$$

$$u_m = \frac{\dot{m}_h}{\rho A_c} = \frac{\dot{m}_h}{\rho \frac{\pi\left(D_i^2 - N_t d_o^2\right)}{4}} = \frac{5}{870 \times \frac{\pi \times \left(0.10226^2 - 3 \times 0.0334^2\right)}{4}} = 1.03 \text{ m/s}$$

$$Re = \frac{\rho u_m D_h}{\mu} = \frac{870 \times 1.03 \times 0.035}{0.124} = 252.9 \quad \text{(Laminar flow)}$$

Sieder and Tate correlation is used:

$$Nu_T = \frac{h_o D_e}{k} = 1.86\left(Pr Re \frac{D_h}{L}\right)^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.14}$$

$$\left(\text{PrRe}\frac{D_h}{L}\right)^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.14} = \left(1760\times 252.9\times\frac{0.035}{3}\right)^{1/3}\left(\frac{0.124}{0.3}\right)^{0.14} = 2.32 \geq 2$$

Therefore, Sieder and Tate correlation is valid for the given data, and

$$h_o = \frac{Nu_T k}{D_e} = \frac{1.86\times 2.32\times 0.141}{0.071} = 8.57$$

Fouling factors:

Table 6.10 and 6.11 give:

$$R_{fi} = 8.8\times 10^{-5} \;\; m^2.K/W$$

$$R_{fo} = 0.000176 \;\; m^2.K/W$$

The overall heat transfer coefficient based on the outside surface area is:

$$U_f = \frac{1}{\frac{d_o}{d_i h_i} + \frac{d_o R_{fi}}{d_i} + \frac{r_o \ln(r_o/r_i)}{k} + R_{fo} + \frac{1}{h_o}} = \frac{1}{\frac{0.0334}{0.02664\times 1183.3} + \frac{0.0334\times 8.8\times 10^{-5}}{0.02664} + 0.000176 + \frac{1}{56.6}}$$

$$= 55.6 \;\; W/m^2.K$$

The heat transfer surface area:

$$A_o = \frac{Q}{U_o \Delta T_m} = \frac{\dot{m}_h c_{p,h}\left(T_{h1} - T_{h2}\right)}{U_o \Delta T_m} = \frac{5\times 2005\times(60-40)}{55.6\times 24.6} = 146.7 \;\; m^2$$

Surface area per hairpin:

$$A_o = \pi \times d_o \times 2L = 2\pi \times 0.0334 \times 3 = 0.63 \; m^2$$

The number of hairpins can then be calculated as:

$$N_{hp} = \frac{A_o}{A_{hp} N_t} = \frac{146.7}{0.63\times 3} = 78 \text{ hairpins}$$

The friction factor in the annulus can be calculated as:

$$f = \frac{16}{Re} = \frac{16}{252.9} = 0.0633$$

The pressure drop in the annulus can be calculated as:

$$\Delta P = 4f\frac{L}{D_h}\rho\frac{u_m^2}{2} = 4\times 0.0633\times\frac{3\times 2\times 78}{0.035}\times 870\times\frac{1.03^2}{2} = 1.56 \text{ MPa}$$

Pressure drop in the tubes:

$$f = 0.037$$

$$\Delta P = 4f\frac{L}{d_i}\rho\frac{u_m^2}{2} = 4\times 0.0037\times\frac{3\times 2\times 78}{0.02664}\times 997\times\frac{8.64^2}{2} = 9.67 \text{ MPa}$$

The number of hairpins can be reduced by the use of finned tubes.

Problem 7.4

The objective of this problem is to design an oil cooler with sea water. The decision was made to use a hairpin heat exchanger.

Fluid	Annulus fluid - engine oil	Tube-side fluid - sea water
Flow rate, kg/s	4	-
Inlet temperature, °C	65	20
Outlet temperature, °C	55	30
Density, kg/m^3	885.27	1013.4
Specific heat, kJ/kg.°C	1.902	4.004
viscosity, kg/m.s	0.075	9.64×10^{-4}
Prandtl number (Pr)	1050	6.29
Thermal conductivity, W/m.K	0.1442	0.6389

The length of the hairpin	= 3 m.
Annulus nominal diameter	= 2 inches
Nominal diameter of the inner tube	= 3/4 inches
Fin height, H	= 0.00127 m
Fin thickness, δ	= 0.9 mm
Number of fins	= 18
Material throughout	= carbon steel
Thermal conductivity, k	= 52 W/m.K
Number of tubes inside the annulus	= 3

Select the proper fouling factors. Calculate:

a. The velocity in the tube and in the annulus;

b. The overall heat transfer coefficient for clean and fouled heat exchanger;

c. The total heat transfer area of the heat exchanger with and without fouling; OS design;

d. The surface area of a hairpin and the number of hairpins;

e. Pressure drop inside the tube in the annulus;

f. Pumping powers for both streams.

SOLUTION:

Nominal diameter of the inner tube = 3/4 in. (Schedule 40):

$d_o = 0.02667$ m

$d_i = 0.02093$ m

Annulus nominal diameter = 2 in. (Schedule 40):

$D_i = 0.0525$ m

Heat balance:

$$Q = \dot{m}_c c_{p,c}\left(T_{c2} - T_{c1}\right) = \dot{m}_h c_{p,h}\left(T_{h1} - T_{h2}\right)$$

$$\dot{m}_c = \frac{\dot{m}_h c_{p,h}\left(T_{h1} - T_{h2}\right)}{c_{p,c}\left(T_{c2} - T_{c1}\right)} = \frac{4 \times 1.902 \times 10^3 \times 10}{4.004 \times 10^3 \times 10} = 1.9 \quad kg/s$$

Annulus:

$$A_c = \frac{\pi}{4}\left(D_i^2 - d_o^2 N_t\right)$$

$$= \frac{\pi}{4}\left(0.0525^2 - 0.02667^2 \times 3\right)$$

$$= 4.888 \times 10^{-4} \quad m^2$$

Hydraulic diameter:

$$P_w = \pi\left(D_i + d_o N_t\right)$$

$$= \pi \times \left(0.0525 + 0.02667 \times 3\right)$$

$$= 0.416 \quad m$$

$$D_h = \frac{4A_c}{P_w} = \frac{4 \times 4.888 \times 10^{-4}}{0.416} = 0.0047$$

Equivalent diameter:

$$P_h = \pi d_o N_t$$
$$= \pi \times 0.02667 \times 3$$
$$= 0.251 \ \ m$$

$$D_e = \frac{4A_c}{P_h} = \frac{4 \times 4.888 \times 10^{-4}}{0.251} = 0.0078$$

a.

Inner tube--cold:

$$u_m = \frac{\dot{m}_c}{\rho A_c N_t} = \frac{\dot{m}_c}{\rho\pi \frac{d_i^{\,2}}{4} N_t} = \frac{4 \times 1.9}{1013.4 \times \pi \times 0.02093^2 \times 3} = 1.82 \ \ m/s$$

Annulus--hot:

$$u_m = \frac{\dot{m}_h}{\rho_h A_c} = \frac{4}{885.27 \times 4.888 \times 10^{-4}} = 9.24 \ \ m/s$$

b.

Inner tube--cold:

Reynolds number:

$$Re = \frac{\rho u_m d_i}{\mu} = \frac{1013.4 \times 1.82 \times 0.02093}{9.64 \times 10^{-4}} = 40045 \quad \text{(Turbulent flow)}$$

Petuknov and Krillov correlation:

$$f = (3.64 \log Re - 3.28)^{-2} = (3.64 \times \log 40045 - 3.28)^{-2} = 0.0055$$

$$Nu = \frac{(f/2) Re Pr}{1.07 + 12.7 (f/2)^{\frac{1}{2}} \left(Pr^{\frac{2}{3}} - 1 \right)} = \; = \frac{\left(\frac{0.0055}{2}\right) \times 40045 \times 6.29}{1 + 12.7 \times \left(\frac{0.0055}{2}\right)^{\frac{1}{2}} \left(6.29^{\frac{2}{3}} - 1 \right)} = 259.1$$

$$h_i = \frac{Nu \cdot k}{d_i} = \frac{259.1 \times 0.639}{0.02094} = 7910 \ \ W/m^2.K$$

Annulus -- hot:

Reynolds number:

$$Re = \frac{\rho u_m D_h}{\mu} = \frac{885.27 \times 9.24 \times 0.0047}{0.075} = 512.6 \quad \text{(Laminar)}$$

Sieder and Tate correlation:

$$Nu_a = 1.86 \left(Re\,Pr \frac{D_h}{L} \right)^{1/3} \left(\frac{\mu_b}{\mu_w} \right)^{0/14}$$

$$Re\,Pr \frac{D_h}{L} = 512.6 \times 1050 \times \frac{0.0047}{3} = 843.2 \geq 2$$

$$T_w = \frac{1}{2} \left(\frac{65 + 55}{2} + \frac{20 + 30}{2} \right) = 42.5^o\,C = 315\ K$$

$$\mu_w = 0.197 \ Pa.s$$

$$Nu_a = 1.86(843.2)^{1/3}\left(\frac{0.075}{0.197}\right)^{0/14} = 15.35$$

$$h_o = \frac{Nu \cdot k}{D_e} = \frac{15.35 \times 0.1442}{0.0078} = 284\ W/m^2.K$$

Hausen Correlation:

$$Nu = 3.66 + \frac{0.19\left(Pe\frac{D_h}{L}\right)^{0.8}}{1 + 0.117\left(Pe\frac{D_h}{L}\right)^{0.467}} = 3.66 + \frac{0.19 \times 39.9^{0.8}}{1 + 0.117 \times 39.9^{0.467}} = 5.85$$

$$h_o = \frac{Nu \cdot k}{D_e} = \frac{5.85 \times 0.1442}{1.053 \times 10^{-3}} = 801.1\ W/m^2.K$$

Compare with h_o = 760 W/m^2.K for Sieder and Tate correlation.

Finned area:

$$A_f = 2N_tN_fL(2H_f + \delta) = 2 \times 3 \times 18 \times 3 \times \left(2 \times 0.0127 + 0.9 \times 10^{-3}\right) = 8.521\ m^2$$

Unfinned area:

$$A_u = 2N_tL(\pi d_o - N_f\delta) = 2 \times 3 \times (\pi \times 0.02667 \times 3) - 18 \times 3 \times 0.9 \times 10^{-3} = 1.217\ m^2$$

Total area:

$$A_t = A_u + A_f = 8.521 + 1.217 = 9.738\ m^2$$

Inner tube surface area:

$$A_i = 2N_tL\pi d_i = 2 \times 3 \times 3 \times \pi \times 0.02093 = 1.184\ m^2$$

Fin efficiency:

$$m = \sqrt{\frac{2h_o}{\delta k_f}} = \sqrt{\frac{2 \times 760}{0.9 \times 10^{-3} \times 52}} = 180.22$$

$$\eta_f = \frac{\tanh(mH_f)}{mH_f} = \frac{\tanh(180.22 \times 0.0127)}{180.22 \times 0.0127} = 0.428$$

Overall surface efficiency:

$$\eta_o = \left[1 - (1 - \eta_f)\frac{A_f}{A_t}\right] = \left[1 - (1 - 0.428)\frac{8.521}{9.738}\right] = 0.5$$

Overall heat transfer coefficient (fouled):

Fouling resistance:

$R_{fo} = 0.176\times10^{-3}$ m^2.K/W (engine oil, Table 4.6/p123)

$R_{fi} = 0.088\times10^{-3}$ m^2.K/W (sea water, Table 4.10/p127)

$$U_{of} = \frac{1}{\frac{A_o}{A_ih_i} + \frac{A_oR_{fi}}{A_i} + A_oR_w + R_{fo} + \frac{1}{h_o}}$$

$$U_{of} = \frac{1}{\frac{d_o}{d_i h_i} + \frac{d_o R_{fi}}{d_i} + d_o \frac{\ln(d_o/d_i)}{2k} + R_{fo} + \frac{1}{h_o}}$$

$$= \left[\frac{0.02667}{0.02094 \times 7910} + \frac{0.02667}{0.02094} \times 0.088 \times 10^{-3} + \frac{0.02667 \ln\left(\frac{0.02667}{0.02093}\right)}{2 \times 52} + 0.176 \times 10^{-3} + \frac{1}{284} \right]^{-1}$$

$$= 248 \ \ W/m^2.K$$

Clean conditions:

$$U_{oc} = \frac{1}{\frac{d_o}{d_i h_i} + d_o \frac{\ln(d_o/d_i)}{2k} + \frac{1}{h_o}}$$

$$= \left[\frac{0.02667}{0.02093 \times 7910} + \frac{0.02667 \ln\left(\frac{0.02667}{0.02093}\right)}{2 \times 52} + \frac{1}{284} \right]^{-1}$$

$$= 279 \ \ W/m^2.K$$

Cleanliness factor:

$$CF = \frac{U_{of}}{U_{oc}} = \frac{248}{279} = 0.89$$

Total fouling resistance:

$$R_{ft} = \frac{1 - CF}{U_{oc} \cdot CF} = \frac{1 - 0.89}{279 \times 0.89} = 0.000443$$

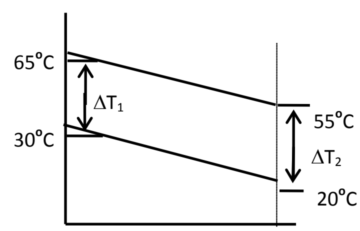
c.

Percentage over surface:

$$OS = 100 \cdot U_{oc} R_{ft} = 100 \times 279 \times 0.000443 = 12.4\%$$

Log-mean temperature difference:

$$\Delta T_{lm,cf} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)}$$



$$\Delta T_m = \Delta T_1 = \Delta T_2 = 35^o \text{ C}$$

Total heat transfer area (without fouling):

$$A_o = \frac{Q}{U_{oc} \Delta T_m} = \frac{\dot{m}_h c_{p,h} (T_{h1} - T_{h2})}{U_{oc} \Delta T_m} = \frac{4 \times 1902 \times (65 - 55)}{279 \times 35} = 7.791 \ \ m^2$$

Total heat transfer area (with fouling):

$$A_o = \frac{Q}{U_{of} \Delta T_m} = \frac{\dot{m}_h c_{p,h} (T_{h1} - T_{h2})}{U_{of} \Delta T_m} = \frac{4 \times 1902 \times (65 - 55)}{248 \times 35} = 8.765 \ \ m^2$$

d.

Hairpin surface area:

$$A_{hp} = \pi d_o (2L) \cdot N_{hp} = \pi \times 0.02093 \times (2 \times 3) \times 3 = 1.508 \ \ m^2$$

Therefore, the number of hairpins (without fouling):

$$N_{hp} = \frac{A_{of}}{A_{hp}} = \frac{7.791}{1.508} \approx 6 \text{ Hairpins.}$$

The number of hairpins (with fouling):

$$N_{hp} = \frac{A_{oc}}{A_{hp}} = \frac{8.765}{1.508} \approx 6 \text{ Hairpins.}$$

e.

Pressure drop:

Inner tube:

$$\Delta P_t = 4f \frac{L}{d_i} \rho \frac{u_m^{\ 2}}{2} \cdot N_{hp} = 4 \times 0.0055 \times \frac{3 \times 2}{0.02093} \times 1013.4 \times \frac{1.82^2}{2} \times 6 = 63.48 \ \ kPa$$

Annulus:

$$f_{cp} = \frac{16}{Re} = \frac{16}{512.6} = 0.0312$$

$$f = f_{cp} \left(\frac{\mu_b}{\mu_w} \right)^{-0.58} = 0.0312 \left(\frac{0.075}{0.197} \right)^{-0.58} = 0.055$$

$$\Delta P_a = 4f \frac{L}{D_h} \rho \frac{u_m^{\ 2}}{2} \cdot N_{hp} = 4 \times 0.055 \times \frac{3 \times 2}{0.0047} \times 885.27 \times \frac{9.24^2}{2} \times 6 = 63.7 \ \ MPa$$

f.

Pumping power:

Inner tube:

$$P_t = \frac{\Delta P_t \dot{m}}{\eta_p \rho_t} = \frac{63.48 \times 10^3 \times 1.9}{0.80 \times 1013.4} = 148.8 \ \ W$$

Annulus:

$$P_a = \frac{\Delta P_a \dot{m}_a}{\eta_p \rho_a} = \frac{63.7 \times 10^6 \times 4}{0.80 \times 885.27} = 360 \ \ kW$$

Problem 7.5

Repeat Problem 7.4 with unfinned inner tubes. Write your comments.

SOLUTION:

Nominal diameter of the inner tube = 3/4 in. (Schedule 40):

$d_o = 0.02667$ m

$d_i = 0.02093$ m

Annulus nominal diameter = 2 in. (Schedule 40):

$D_i = 0.0525$ m

Heat balance:

$$Q = \dot{m}_c c_{p,c}\left(T_{c2} - T_{c1}\right) = \dot{m}_h c_{p,h}\left(T_{h1} - T_{h2}\right)$$

$$\dot{m}_c = \frac{\dot{m}_h c_{p,h}\left(T_{h1} - T_{h2}\right)}{c_{p,c}\left(T_{c2} - T_{c1}\right)} = \frac{4 \times 1.902 \times 10^3 \times 10}{4.004 \times 10^3 \times 10} = 1.9 \ \text{kg/s}$$

Annulus:

$$A_c = \frac{\pi}{4}\left(D_i^{\,2} - d_o^{\,2} N_t\right)$$

$$= \frac{\pi}{4}\left(0.0525^2 - 0.02667^2 \times 3\right)$$

$$= 4.888 \times 10^{-4} \ \text{m}^2$$

Hydraulic diameter:

$$P_w = \pi\left(D_i + d_o N_t\right)$$

$$= \pi \times \left(0.0525 + 0.02667 \times 3\right)$$

$$= 0.416 \ \text{m}$$

$$D_h = \frac{4A_c}{P_w} = \frac{4 \times 4.888 \times 10^{-4}}{0.416} = 0.0047$$

Equivalent diameter:

$$P_h = \pi d_o N_t$$

$$= \pi \times 0.02667 \times 3$$

$$= 0.251 \ \text{m}$$

$$D_e = \frac{4A_c}{P_h} = \frac{4 \times 4.888 \times 10^{-4}}{0.251} = 0.0078$$

a.

<u>Inner tube--cold:</u>

$$u_m = \frac{\dot{m}_c}{\rho A_c N_t} = \frac{\dot{m}_c}{\rho\pi \frac{d_i^{\,2}}{4} N_t} = \frac{4 \times 1.9}{1013.4 \times \pi \times 0.02093^2 \times 3} = 1.82 \ \ m/s$$

<u>Annulus--hot:</u>

$$u_m = \frac{\dot{m}_h}{\rho_h A_c} = \frac{4}{885.27 \times 4.888 \times 10^{-4}} = 9.24 \ \ m/s$$

b.

<u>Inner tube--cold:</u>

Reynolds number:

$$Re = \frac{\rho u_m d_i}{\mu} = \frac{1013.4 \times 1.82 \times 0.02093}{9.64 \times 10^{-4}} = 40045 \quad \text{(Turbulent flow)}$$

Petuknov and Krillov correlation:

$$f = (3.64 \log Re - 3.28)^{-2} = (3.64 \times \log 40045 - 3.28)^{-2} = 0.0055$$

$$Nu = \frac{(f/2) Re Pr}{1.07 + 12.7 (f/2)^{\frac{1}{2}} \left(Pr^{\frac{2}{3}} - 1 \right)} = \ = \frac{\left(\frac{0.0055}{2} \right) \times 40045 \times 6.29}{1 + 12.7 \times \left(\frac{0.0055}{2} \right)^{\frac{1}{2}} \left(6.29^{\frac{2}{3}} - 1 \right)} = 259.1$$

$$h_i = \frac{Nu \cdot k}{d_i} = \frac{259.1 \times 0.639}{0.02094} = 7910 \ \ W/m^2.K$$

<u>Annulus -- hot:</u>

Reynolds number:

$$Re = \frac{\rho u_m D_h}{\mu} = \frac{885.27 \times 9.24 \times 0.0047}{0.075} = 512.6 \quad \text{(Laminar)}$$

Sieder and Tate correlation:

$$Nu_a = 1.86 \left(Re\, Pr \frac{D_h}{L} \right)^{1/3} \left(\frac{\mu_b}{\mu_w} \right)^{0/14}$$

$$Re\, Pr \frac{D_h}{L} = 512.6 \times 1050 \times \frac{0.0047}{3} = 843.2 \geq 2$$

$$T_w = \frac{1}{2} \left(\frac{65 + 55}{2} + \frac{20 + 30}{2} \right) = 42.5^o\, C = 315\ K$$

$$\mu_w = 0.197 \ Pa.s$$

$$Nu_a = 1.86 (843.2)^{1/3} \left(\frac{0.075}{0.197} \right)^{0/14} = 15.35$$

$$h_o = \frac{Nu \cdot k}{D_e} = \frac{15.35 \times 0.1442}{0.0078} = 284 \ W/m^2.K$$

Overall heat transfer coefficient (fouled):

Fouling resistance:

$R_{fo} = 0.176 \times 10^{-3}$ $m^2.K/W$ (engine oil, Table 4.6/p123)

$R_{fi} = 0.088\times10^{-3}$ $m^2.K/W$ (sea water, Table 4.10/p127)

$$U_{of} = \frac{1}{\dfrac{A_o}{A_i h_i} + \dfrac{A_o R_{fi}}{A_i} + A_o R_w + R_{fo} + \dfrac{1}{h_o}}$$

$$U_{of} = \frac{1}{\dfrac{d_o}{d_i h_i} + \dfrac{d_o R_{fi}}{d_i} + d_o \dfrac{\ln(d_o/d_i)}{2k} + R_{fo} + \dfrac{1}{h_o}}$$

$$= \left[\frac{0.02667}{0.02094\times7910} + \frac{0.02667}{0.02094}\times0.088\times10^{-3} + \frac{0.02667\ln\left(\dfrac{0.02667}{0.02093}\right)}{2\times52} + 0.176\times10^{-3} + \frac{1}{284}\right]^{-1}$$

$$= 248 \ \ W/m^2.K$$

Clean conditions:

$$U_{oc} = \frac{1}{\dfrac{d_o}{d_i h_i} + d_o \dfrac{\ln(d_o/d_i)}{2k} + \dfrac{1}{h_o}}$$

$$= \left[\frac{0.02667}{0.02093\times7910} + \frac{0.02667\ln\left(\dfrac{0.02667}{0.02093}\right)}{2\times52} + \frac{1}{284}\right]^{-1}$$

$$= 279 \ \ W/m^2.K$$

Cleanliness factor:

$$CF = \frac{U_{of}}{U_{oc}} = \frac{248}{279} = 0.89$$

Total fouling resistance:

$$R_{ft} = \frac{1-CF}{U_{oc}\cdot CF} = \frac{1-0.89}{279\times0.89} = 0.000443$$

c.

Percentage over surface:

$$OS = 100\cdot U_{oc}R_{ft} = 100\times279\times0.000443 = 12.4\%$$

Log-mean temperature difference:

$$\Delta T_{lm,cf} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\dfrac{\Delta T_1}{\Delta T_2}\right)}$$

$$\Delta T_m = \Delta T_1 = \Delta T_2 = 35^o\,C$$

Total heat transfer area (without fouling):

$$A_o = \frac{Q}{U_{oc}\Delta T_m} = \frac{\dot m_h c_{p,h}(T_{h1}-T_{h2})}{U_{oc}\Delta T_m} = \frac{4\times1902\times(65-55)}{279\times35} = 7.791 \ \ m^2$$

Total heat transfer area (with fouling):

$$A_o = \frac{Q}{U_{of}\,\Delta T_m} = \frac{\dot{m}_h c_{p,h}\left(T_{h1} - T_{h2}\right)}{U_{of}\,\Delta T_m} = \frac{4 \times 1902 \times (65 - 55)}{248 \times 35} = 8.765 \ \ m^2$$

d.

Hairpin surface area:

$$A_{hp} = \pi d_o (2L) \cdot N_{hp} = \pi \times 0.02093 \times (2 \times 3) \times 3 = 1.508 \ \ m^2$$

Therefore, the number of hairpins (without fouling):

$$N_{hp} = \frac{A_{of}}{A_{hp}} = \frac{7.791}{1.508} \approx 6 \text{ Hairpins.}$$

The number of hairpins (with fouling):

$$N_{hp} = \frac{A_{oc}}{A_{hp}} = \frac{8.765}{1.508} \approx 6 \text{ Hairpins.}$$

e.

Pressure drop:

Inner tube:

$$\Delta P_t = 4f \frac{L}{d_i} \rho \frac{{u_m}^2}{2} \cdot N_{hp} = 4 \times 0.0055 \times \frac{3 \times 2}{0.02093} \times 1013.4 \times \frac{1.82^2}{2} \times 6 = 63.48 \ \ kPa$$

Annulus:

$$f_{cp} = \frac{16}{Re} = \frac{16}{512.6} = 0.0312$$

$$f = f_{cp}\left(\frac{\mu_b}{\mu_w}\right)^{-0.58} = 0.0312\left(\frac{0.075}{0.197}\right)^{-0.58} = 0.055$$

$$\Delta P_a = 4f \frac{L}{D_h} \rho \frac{{u_m}^2}{2} \cdot N_{hp} = 4 \times 0.055 \times \frac{3 \times 2}{0.0047} \times 885.27 \times \frac{9.24^2}{2} \times 6 = 63.7 \ \ MPa$$

f.

Pumping power:

Inner tube:

$$P_t = \frac{\Delta P_t \dot{m}}{\eta_p \rho_t} = \frac{63.48 \times 10^3 \times 1.9}{0.80 \times 1013.4} = 148.8 \ \ W$$

Annulus:

$$P_a = \frac{\Delta P_a \dot{m}_a}{\eta_p \rho_a} = \frac{63.7 \times 10^6 \times 4}{0.80 \times 885.27} = 360 \ \ kW$$

Problem 7.6

A double pipe heat exchanger is designed as an engine oil cooler. The flow rate of oil is 5 kg/s and it will be cooled from 60°C to 40°C through annulus (ID = 0.10226 m, OD = 0.1143 m). Sea water flows through the tubes (ID = 0.02664 m, OD = 0.03340) and is heated from 10°C to 30°C. The number of bare tubes in the annulus is 3 and the length of the hairpin is 3 m. Assume that the tube wall temperature is 35°C. Design calculations give the number of hairpins as 85. Allowable pressure drop in the heat exchanger for both stream is 20 psi. Is this design acceptable? Outline your comments.

GIVEN:

Tube side (water):

-Inlet water temperature (T_{c1}) = 10°C

-Outlet water temperature (T_{c2}) = 30°C

-Number of bare tubes in the annulus (#) = 3

-Diameters: d_o = 1.315 in. = 0.03340 m

d_i = 1.049 in. = 0.02664 m

- wall temperature (T_w) = 35 °C

-Length of the hairpin (L) = 3 m

Annulus (engine oil):

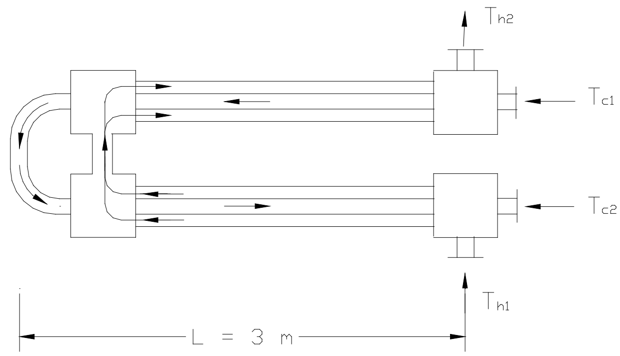
-Inlet temperature (T_{h1}) = 60°C

-Outlet temperature (T_{h2}) = 40°C

-Mass flow rate ($\dot{m}_h$) = 5 kg/s

-Diameters: D_o = 4.5 in. = 0.1143 m

D_i = 4.026 in. = 0.10226 m



SOLUTION:

Properties of water at $T = \frac{10+30}{2} = 20^o\,C$:

$\rho = 998 \;\; kg/m^3 \qquad c_p = 4182 \;\; J/kg.K \qquad \mu = 10.07\times10^{-4} \;\; N.s/m^2$

$k = 0.599 \; W/m.K \qquad Pr = 7.05 \qquad \mu_w = 7.20\times10^{-4} \;\; N.s/m^2$

Properties of oil at $T = \frac{40+60}{2} = 50^o\,C$:

$\rho = 870 \;\; kg/m^3 \qquad c_p = 2006 \;\; J/kg.K \qquad \mu = 0.123 \;\; N.s/m^2$

$k = 0.141 \; W/m.K \qquad Pr = 1749 \qquad \mu_w = 0.30 \;\; N.s/m^2$

a.

Calculation of heat transfer coefficient in annulus:

Equivalent diameter for heat transfer:

$$A_c = \frac{\pi}{4}\left(D_i^{\,2} - 3d_o^{\,2}\right)$$

$$= \frac{\pi}{4}\left(0.1022604^2 - 3\times0.0334^2\right)$$

$$= 5.585\times10^{-3} \;\; m^2$$

$$D_e = \frac{4A_c}{3\pi d_o} = \frac{4\times5.585\times10^{-3}}{3\pi\times0.0334} = 0.071 \text{ m}$$

Hydraulic diameter:

$$D_h = \frac{4A_c}{\pi D_i + 3\pi d_o} = \frac{4\times5.585\times10^{-3}}{\pi\times0.10226 + 3\pi\times0.0334} = 0.035 \text{ m}$$

Reynolds number:

$$u_m = \frac{\dot{m}_h}{\rho_h A_c} = \frac{5}{870\times5.585\times10^{-3}} = 1.029 \;\; m/s$$

$$Re = \frac{\rho u_m D_h}{\mu} = \frac{870 \times 1.029 \times 0.035}{0.123} = 254.7 \quad \text{(Laminar flow)}$$

Sieder and Tate correlation:

$$Nu_a = 1.86\left(Re\,Pr\frac{D_h}{L}\right)^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0/14}$$

$$\left(Re\,Pr\frac{D_h}{L}\right)^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.14} = \left(254.7 \times 1749 \times \frac{0.035}{3}\right)^{1/3}\left(\frac{0.123}{0.3}\right)^{0.14} = 15.29 \geq 2\ ,$$

so this correlation is valid for the given data.

$$Nu_T = 1.86\left(Re\,Pr\frac{D_h}{L}\right)^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.14} = 1.86 \times 15.29 = 28.44$$

$$h_o = \frac{Nu \cdot k}{D_e} = \frac{28.44 \times 0.141}{0.071} = 56.48\ W/m^2.K$$

b.

Calculation of heat transfer coefficient in tubes:

Heat balance:

$$Q = \dot{m}_c c_{p,c}\left(T_{c2} - T_{c1}\right) = \dot{m}_h c_{p,h}\left(T_{h2} - T_{h1}\right)$$

$$\dot{m}_c = \frac{\dot{m}_h c_{p,h}\left(T_{h2} - T_{h1}\right)}{c_{p,c}\left(T_{c2} - T_{c1}\right)} = \frac{5 \times 2006 \times (60-40)}{4182(30-10)} = 2.4\ \ kg/s$$

$$A_t = \frac{\pi}{4} d_i^2 = \frac{\pi}{4} \times 0.02664^2 = 5.57 \times 10^{-4}\ \ m^2$$

$$\bar{u} = \frac{\dot{m}_c}{\rho A_t n} = \frac{2.4}{998 \times 5.57 \times 10^{-4} \times 3} = 1.44\ \ m/s$$

$$Re = \frac{\rho \bar{u} d_i}{\mu} = \frac{998 \times 1.44 \times 0.02664}{10.07 \times 10^{-4}} = 38019\ \ \text{(Turbulent flow)}$$

Petuknov and Krillov correlation:

$$f = \left(3.64 \log Re - 3.28\right)^{-2} = \left(3.64 \times \log 38019 - 3.28\right)^{-2} = 0.0056$$

$$Nu = \frac{\left(f/2\right) Re Pr}{1.07 + 12.7\left(f/2\right)^{\frac{1}{2}}\left(Pr^{\frac{2}{3}} - 1\right)} = \ = \frac{\left(\frac{0.0056}{2}\right) \times 38019 \times 7.05}{1.07 + 12.7 \times \left(\frac{0.0056}{2}\right)^{\frac{1}{2}}\left(7.05^{\frac{2}{3}} - 1\right)} = 261.6$$

$$h_i = \frac{Nu \cdot k}{d_i} = \frac{261.6 \times 0.599}{0.02664} = 5882\ \ W/m^2.K$$

Overall heat transfer coefficient:

From 1988 TEMA standards, fouling resistance:

$R_{fo} = 0.176 \times 10^{-3}\ \ m^2.K/W$

$R_{fi} = 0.088 \times 10^{-3}\ \ m^2.K/W$

$k = 43$ W/m.K

$$U_o = \frac{1}{\dfrac{r_o}{r_i h_i} + \dfrac{r_o R_{fi}}{r_i} + r_o \dfrac{\ln(r_o/r_i)}{k} + R_{fo} + \dfrac{1}{h_o}}$$

$$= \left[\frac{0.0334}{0.02664 \times 5882} + \frac{0.0334}{0.02664} \times 0.088 \times 10^{-3} + \frac{0.0167 \ln\left(\dfrac{0.0334}{0.02664}\right)}{43} + 0.176 \times 10^{-3} + \frac{1}{56.48} \right]^{-1}$$

$$= 55 \ \text{W/m}^2.\text{K}$$

c.

Calculation of heat transfer area:

$$Q = U_o A_o \Delta T_{lm}$$

$$Q = \dot{m}_h c_{p,h} (T_{h2} - T_{h1})$$

$$= 5 \times 2006 \times (60 - 40)$$

$$= 200.5 \ \text{kW}$$

Log-mean temperature difference:

$$\Delta T_{lm,cf} = \Delta T_1 = \Delta T_2 = 30 \ ^oC$$

$$A_o = \frac{Q}{U_o \Delta T_m} = \frac{200.5 \times 10^3}{55 \times 30} = 121.5 \ \text{m}^2$$

Surface area per hairpin:

$$A = 2\pi d_o L_{hp} = 2\pi \times 0.0334 \times 3 = 0.630 \ \text{m}^2$$

d.

Number of hairpins:

$$N_{hp} = \frac{A_o}{A_{hp} \cdot n} \quad (\text{ n = number of tubes in annulus })$$

$$N_{hp} = \frac{121.5}{0.63 \times 3} = 64.28 \ \text{hairpins} \quad \Rightarrow n = 65 \ \text{hairpins}$$

e.

Total pressure drop in annulus:

$$f = \frac{16}{Re} = \frac{16}{254.7} = 0.0628$$

$$\Delta P = 4f \frac{L N_{hp}}{D_h} \rho \frac{u^2}{2} = 4 \times 0.0628 \times \frac{3 \times 2 \times 65}{0.035} \times 870 \times \frac{1.029^2}{2} = 1.29 \ \text{MPa} = 187.09 \ \text{psi}$$

Total pressure drop in tubes:

$$f = 0.0056$$

$$\Delta P = 4f \frac{L}{D_h} \rho \frac{u^2}{2} = 4 \times 0.0056 \times \frac{3 \times 2 \times 65}{0.02664} \times 998 \times \frac{1.44^2}{2} = 0.34 \ \text{MPa} = 49 \ \text{psi}$$

Problem 7.7

Assume that the mass flow rate of oil in Problem 7.6 is doubled (10 kg/s) which may result in an unacceptable pressure drop. It is decided that two-hairpins will be used in which the hot fluid flows in parallel between two units and the cold fluid in series flow. Calculate:

a. the pressure drop in the annulus for series arrangements for both streams;

b. the pressure drop in the annulus if the hot fluid is split equally and flows in parallel between these two units.

GIVEN:

Tube side (water):

-Inlet water temperature (T_{c1}) = 10^{o}C

-Outlet water temperature (T_{c2}) = 30^{o}C

-Number of bare tubes in the annulus (#) = 3

-Diameters: d_o = 1.315 in. = 0.03340 m

d_i = 1.049 in. = 0.02664 m

- wall temperature (T_w) = 35 ^{o}C

Annulus (engine oil):

-Inlet temperature (T_{h1}) = 60^{o}C

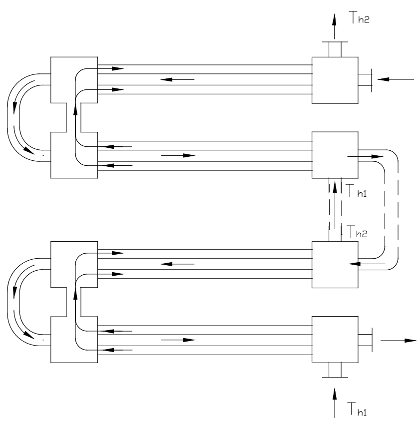
-Outlet temperature (T_{h2}) = 40^{o}C

-Mass flow rate ($\dot{m}_h$) = 10 kg/s

-Diameters: D_o = 4.5 in. = 0.1143 m

D_i = 4.026 in. = 0.10226 m

-Number of hairpins (N_{hp}) = 2



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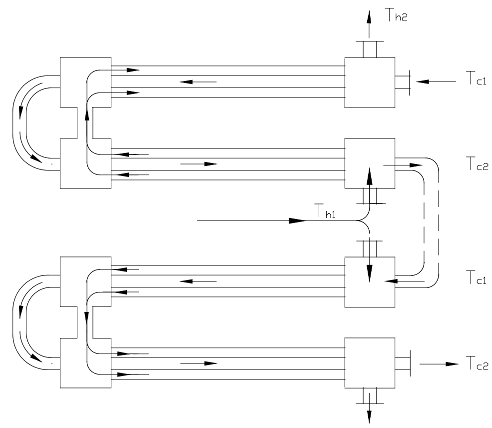

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SOLUTION:

Properties of water at $T = \frac{10+30}{2} = 20^{o}C$:

$\rho = 998 \ \ kg/m^3 \qquad c_p = 4182 \ \ J/kg.K \qquad \mu = 10.07\times10^{-4} \ \ N.s/m^2$

$k = 0.599 \ W/m.K \qquad Pr = 7.05 \qquad \mu_w = 7.20\times10^{-4} \ \ N.s/m^2$

Properties of oil at $T = \frac{40+60}{2} = 50^{o}C$:

$\rho = 870 \ \ kg/m^3 \qquad c_p = 2006 \ \ J/kg.K \qquad \mu = 0.123 \ \ N.s/m^2$

$k = 0.141 \ W/m.K \qquad Pr = 1749 \qquad \mu_w = 0.30 \ \ N.s/m^2$

a.

Annulus oil velocity:

$$u_m = \frac{\dot{m}_h}{\rho_h A_c} = \frac{0.4}{870\times1099.6\times10^{-6}} = 0.42 \ \ m/s$$

$$Re = \frac{\rho u_m D_h}{\mu} = \frac{870\times2.058\times0.035}{0.123} = 509.5 \quad \text{(Laminar flow)}$$

$$f = \frac{16}{Re} = \frac{16}{509.5} = 0.0314$$

$$D_h = 0.035 \text{ m}$$

$$N_{hp} = 2$$

$$\Delta P = 4f\frac{L\cdot N_{hp}}{D_h}\rho\frac{u^2}{2} = 4\times 0.0314\times\frac{3\times 2}{0.035}\times 870\times\frac{2.058^2}{2} = 39.7\ \ \text{kPa}$$

b.

$$u_m = \frac{\dot{m}_h}{\rho A_c} = \frac{0.4}{996\times\frac{\pi}{4}\times 0.025^2} = 0.818\ \ \text{m/s}$$

$$Re = \frac{\rho u_m D_h}{\mu} = \frac{870\times 1.029\times 0.035}{0.123} = 254.7\quad \text{(Laminar flow)}$$

$$f = \frac{16}{Re} = \frac{16}{254.7} = 0.0628$$

$$\Delta P = 4f\frac{L\cdot N_{hp}}{D_h}\rho\frac{u^2}{2} = 4\times 0.0628\times\frac{3\times 2}{0.035}\times 870\times\frac{1.029^2}{2} = 19.8\ \ \text{kPa}$$

Calculation of heat transfer coefficient in annulus:

Sieder and Tate correlation is used for Re = 509.5 in annular side:

$$Nu_a = 1.86\left(Re\,Pr\frac{D_h}{L}\right)^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0/14}$$

$$\left(Re\,Pr\frac{D_h}{L}\right)^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.14} = \left(509.5\times 1749\times\frac{0.035}{3}\right)^{1/3}\left(\frac{0.123}{0.3}\right)^{0.14} = 15.29 \ge 2\,,$$

so this correlation is valid for the given data.

$$Nu_T = 1.86\left(Re\,Pr\frac{D_h}{L}\right)^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.14} = 1.86\times 15.29 = 28.44$$

$$h_o = \frac{Nu\cdot k}{D_e} = \frac{28.44\times 0.141}{0.071} = 56.48\ \text{W/m}^2.\text{K}$$

Calculation of heat transfer coefficient in tubes:

Heat balance:

$$Q = \dot{m}_c c_{p,c}\left(T_{c2}-T_{c1}\right) = \dot{m}_h c_{p,h}\left(T_{h2}-T_{h1}\right)$$

$$\dot{m}_c = \frac{\dot{m}_h c_{p,h}\left(T_{h2}-T_{h1}\right)}{c_{p,c}\left(T_{c2}-T_{c1}\right)} = \frac{5\times 2006\times(60-40)}{4182(30-10)} = 2.4\ \ \text{kg/s}$$

$$A_t = \frac{\pi}{4}d_i^{\,2} = \frac{\pi}{4}\times 0.02664^2 = 5.57\times 10^{-4}\quad \text{m}^2$$

$$\bar{u} = \frac{\dot{m}_c}{\rho A_t n} = \frac{2.4}{998\times 5.57\times 10^{-4}\times 3} = 1.44\ \ \text{m/s}$$

$$Re = \frac{\rho\bar{u}d_i}{\mu} = \frac{998\times 1.44\times 0.02664}{10.07\times 10^{-4}} = 38019\ \ \text{(Turbulent flow)}$$

Petuknov and Krillov correlation:

$$f = \left(3.64\log Re - 3.28\right)^{-2} = \left(3.64\times\log 38019 - 3.28\right)^{-2} = 0.0056$$

$$Nu = \frac{(f/2)RePr}{1.07 + 12.7(f/2)^{\frac{1}{2}}\left(Pr^{\frac{2}{3}} - 1\right)} = \; = \frac{\left(\frac{0.0056}{2}\right) \times 38019 \times 7.05}{1.07 + 12.7 \times \left(\frac{0.0056}{2}\right)^{\frac{1}{2}}\left(7.05^{\frac{2}{3}} - 1\right)} = 261.6$$

$$h_i = \frac{Nu \cdot k}{d_i} = \frac{261.6 \times 0.599}{0.02664} = 5882 \;\; W/m^2.K$$

Overall heat transfer coefficient:

From 1988 TEMA standards, fouling resistance:

$R_{fo} = 0.176\times10^{-3}$ $m^2.K/W$

$R_{fi} = 0.088\times10^{-3}$ $m^2.K/W$

k = 43 W/m.K

$$U_o = \frac{1}{\frac{r_o}{r_i h_i} + \frac{r_o R_{fi}}{r_i} + r_o \frac{\ln(r_o/r_i)}{k} + R_{fo} + \frac{1}{h_o}}$$

$$= \left[\frac{0.0334}{0.02664 \times 5882} + \frac{0.0334}{0.02664} \times 0.088 \times 10^{-3} + \frac{0.0167 \ln\left(\frac{0.0334}{0.02664}\right)}{43} + 0.176 \times 10^{-3} + \frac{1}{56.48}\right]^{-1}$$

$$= 55 \;\; W/m^2.K$$

Calculation of heat transfer area:

$$Q = U_o A_o \Delta T_{lm}$$

$$Q = \dot{m}_h c_{p,h}\left(T_{h2} - T_{h1}\right)$$

$$= 5 \times 2006 \times (60 - 40)$$

$$= 200.5 \;\; kW$$

Log-mean temperature difference:

$$\Delta T_{lm,cf} = \Delta T_1 = \Delta T_2 = 30 \; ^oC$$

$$A_o = \frac{Q}{U_o \Delta T_m} = \frac{200.5 \times 10^3}{55 \times 30} = 121.5 \;\; m^2$$

Surface area per hairpin:

$$A = 2\pi d_o L_{hp} = 2\pi \times 0.0334 \times 3 = 0.630 \;\; m^2$$

Number of hairpins:

$$N_{hp} = \frac{A_o}{A_{hp} \cdot n}$$ (n = number of tubes in annulus)

$$N_{hp} = \frac{121.5}{0.63 \times 3} = 64.28 \;\; \text{hairpins} \quad \Rightarrow n = 65 \text{ hairpins}$$

Total pressure drop in annulus:

$$f = \frac{16}{Re} = \frac{16}{254.7} = 0.0628$$

$$\Delta P = 4f\,\frac{LN_{hp}}{D_h}\rho\frac{u^2}{2} = 4\times 0.0628\times\frac{3\times 2\times 65}{0.035}\times 870\times\frac{1.029^2}{2} = 1.29\ \ \text{MPa} = 187.09\ \text{psi}$$

Total pressure drop in tubes:

$$f = 0.0056$$

$$\Delta P = 4f\,\frac{L}{D_h}\rho\frac{u^2}{2} = 4\times 0.0056\times\frac{3\times 2\times 65}{0.02664}\times 998\times\frac{1.44^2}{2} = 0.34\ \ \text{MPa} = 49\ \text{psi}$$

Problem 7.8

Seawater at 30°C flows on the inside of a 25-mm I.D. steel tube with a 0.8-mm wall thickness at a flow rate of 0.4 kg/s. The tube forms the inside of a double -pipe heat exchanger. Engine oil (refined lubricating oil) at 100°C flows in the annular space with a flow rate of 0.1 kg/s. The outlet temperature of the oil is 60°C. The material is carbon steel. The I.D. of the outer tube is 45 m. Calculate:

a. Heat transfer coefficient in the annulus

b. Heat transfer coefficient inside the tube

c. Overall heat transfer coefficient with fouling

d. Area of the heat exchanger; and by assuming the length of a hairpin to be 4 m, the number of hairpins

e. Pressure drop and pumping powers for both streams

GIVEN:

Tube side (water):

-Inlet water temperature (T_{c1}) = 30 °C

-Mass flow rate of inlet water ($\dot{m}_c$) = 0.4 kg/s

-Number of bare tubes in the annulus (#) = 3

-Diameters: d_o = 26.6 mm

d_i = 25 mm

-Length of the hairpin (L) = 4 m

Annulus (engine oil):

-Lubricating oil inlet temperature (T_{h1}) = 100°C

-outlet temperature (T_{h2}) = 60°C

-Mass flow rate ($\dot{m}_h$) = 0.1 kg/s

-Diameters: D_i = 45 mm

FIND:

a. Heat transfer coefficient in annulus (h_o)

b. Heat transfer coefficient inside the tube (h_i)

c. Overall heat transfer coefficient with fouling (U_o)

d. Area of the heat exchanger (A_o), number of hairpin (N_{hp})

e. Pressure drop (Δp_a , Δp_t) and pumping powers (P_a , P_t)

SOLUTION:

Properties of oil at $\frac{100+60}{2} = 80\ ^oC$ = 353.15 K:

$$\rho = 852\ kg/m^3 \qquad \mu = 0.032\ \ N.s/m^2$$
$$Pr = 501 \qquad k = 0.138\ \ W/m.K$$
$$c_p = 2133\ \ J/kg.K$$

Properties of water at 30 ^oC :

$$\rho = 996\ kg/m^3 \qquad \mu = 7.98\times10^{-4}\ \ N.s/m^2$$
$$Pr = 5.44 \qquad k = 0.614\ \ W/m.K$$
$$c_p = 4179\ \ J/kg.K$$

a. Annular side heat transfer coefficient:

$$A_c = \frac{\pi}{4}\left(D_i^2 - d_o^2\right) = \frac{\pi}{4}\left(0.045^2 - 0.0266^2\right) = 0.001035\ \ m^2$$

$$D_h = \frac{4A_c}{P_w} = \frac{\left(D_i^2 - d_o^2\right)}{D_i + d_o} = D_i - d_o = 0.045 - 0.0266 = 0.0184\ m$$

$$D_e = \frac{4A_c}{P_h} = \frac{\left(D_i^2 - d_o^2\right)}{d_o} = \frac{0.045^2 - 0.0266^2}{0.0266} = 0.0495\ \ m$$

Annulus oil velocity:

$$u_m = \frac{\dot{m}_h}{\rho_h A_c} = \frac{0.4}{870\times1099.6\times10^{-6}} = 0.42\ \ m/s$$

$$Re == \frac{\rho u_m D_h}{\mu} = \frac{852\times0.1134\times0.0184}{0.032} = 55.6$$

Assume the wall temperature to be 50 ^oC, μ_w = 0.124 Pa.s

Sieder and Tate correlation is used:

$$Nu_T = \frac{h_o D_e}{k} = 1.86\left(PrRe\frac{D_h}{L}\right)^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.14}$$

$$\left(PrRe\frac{D_h}{L}\right)^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.14} = \left(501\times 55.6\times\frac{0.0184}{4}\right)^{1/3}\left(\frac{0.032}{0.124}\right)^{0.14} = 4.17 \geq 2$$

Therefore, Sieder and Tate correlation is valid for the given data, and

$$h_o = \frac{Nu_T k}{D_e} = \frac{1.86\times 4.17\times 0.138}{0.0495} = 21.6$$

b. Tube side heat transfer coefficient:

$$u_m = \frac{\dot{m}_h}{\rho A_c} = \frac{0.4}{996\times\frac{\pi}{4}\times 0.025^2} = 0.818 \ \ m/s$$

$$Re == \frac{\rho u_m d_i}{\mu} = \frac{996\times 0.818\times 0.025}{7.98\times 10^{-4}} = 25529$$

$$f = (3.64\ln Re - 3.28)^{-2} = (3.64\times\ln 25529 - 3.28)^{-2} = 0.0061$$

$$Nu_b = \frac{(f/2)Re_b Pr_b}{1.07 + 12.7(f/2)^{1/2}\left(Pr_b^{2/3} - 1\right)}$$

$$Nu_b = \frac{\left(\frac{0.0061}{2}\right)\times 25529\times 5.44}{1.07 + 12.7\times\left(\frac{0.0061}{2}\right)^{\frac{1}{2}}\left(5.44^{\frac{2}{3}} - 1\right)}$$

$$= 166.9$$

The inside heat transfer coefficient is:

$$h_i = \frac{Nu_b\cdot k}{d_i} = \frac{166.9\times 0.614}{0.025} = 4099 \ W/m^2.K$$

c. Overall heat transfer coefficient with fouling:

Assuming fouling: $R_{fi} = 0.000088\ m^2/(K.W)$

$R_{fo} = 0.000176\ m^2/(K.W)$

and $k_w = 52$ W/(m.K):

The overall heat transfer coefficient based on the outside surface area is:

$$U_f = \frac{1}{\frac{d_o}{d_i h_i} + \frac{d_o R_{fi}}{d_i} + \frac{r_o \ln(r_o/r_i)}{k_w} + R_{fo} + \frac{1}{h_o}}$$

$$= \frac{1}{\frac{0.0266}{0.025\times 4099} + \frac{0.0266\times 8.8\times 10^{-5}}{0.025} + \frac{0.0266\ln(0.0266/0.025)}{2\times 52} + 0.000176 + \frac{1}{21.6}}$$

$$= 21.3 \ \ W/m^2.K$$

d. Heat exchanger area and number of hairpins:

$$Q = \dot{m}_c c_{p,c}\left(T_{c2} - T_{c1}\right) = \dot{m}_h c_{p,h}\left(T_{h1} - T_{h2}\right)$$

$$Q = \dot{m}_h c_{p,h}\left(T_{h1} - T_{h2}\right) = 0.1 \times 2133 \times (100 - 60) = 8532 \text{ W}$$

$$T_{c2} = -T_{c1} + \frac{\dot{m}_h c_{p,h}\left(T_{h1} - T_{h2}\right)}{\dot{m}_c c_{p,c}} = 30 + \frac{8532}{0.4 \times 4179} = 35.1 \ ^{o}\text{C}$$

$$T_m = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\Delta T_1 / \Delta T_2\right)} = \frac{(100 - 35.1) - (60 - 30)}{\ln\left(\frac{100 - 35.1}{60 - 30}\right)} = 45.23 \ ^{o}\text{C}$$

$$Q = U \Delta T_m A_o$$

$$A_o = \frac{Q}{U \Delta T_m} = \frac{8531}{21.3 \times 45.23} = 8.86 \text{ m}^2$$

$$A_o = N_{hp}(2L)\pi d_o$$

$$N_{hp} = \frac{A_o}{(2L)\pi d_o} = \frac{8.86}{2 \times 4 \times \pi \times 0.0266} = 13.25 \approx 14 \text{ hairpins}$$

e. pressure drops and pumping powers:

$$f = \frac{16}{Re} = \frac{16}{55.6} = 0.288$$

$$\Delta P_a = 4f \frac{2LN_{hp}}{D_h} \rho \frac{u^2}{2} = 4 \times 0.288 \frac{2 \times 4 \times 14}{0.0184} \times 852 \times \frac{0.1134^2}{2} = 38.4 \text{ kPa}$$

$$\Delta P_t = 4f \frac{(2L)N_{hp}}{d_i} \rho \frac{u^2}{2} = 4 \times 0.0061 \times \frac{2 \times 4 \times 14}{0.025} \times 996 \times \frac{0.818^2}{2} = 36.4 \text{ kPa}$$

$$P_a = \frac{\dot{m}_h \Delta p_a}{\rho \eta_p} = \frac{0.1 \times 38.4 \times 10^3}{852 \times 0.8} = 5.63 \text{ W}$$

$$P_t = \frac{\dot{m}_c \Delta p_t}{\rho \eta_p} = \frac{0.4 \times 36.4 \times 10^3}{996 \times 0.8} = 18.3 \text{ W}$$

Problem 7.9

A counterflow double pipe heat exchanger is used to cool the lubricating oil for a large industrial gas turbine engine. The flow rate of oil through the annulus is m_h = 0.4 kg/s. The oil and water enter at temperature of 60 °C and 30 °C, respectively. The heat transfer coefficient in the inner tube is calculated to be 3000 W/m^2.K. The inner tube diameter is 25 mm and the I.D. of outer tube is 45 mm. The outlet temperature of oil and water are 40°C and 50°C respectively. Take c_p = 4178 J/kg.K for water and c_p = 2000 J/kg.K for oil. The tube wall resistance and the curvature of the wall are neglected. Assume the length of the double pipe heat exchanger as 4 m. Calculate:

a. the heat transfer coefficient in the annulus.

b. the heat transfer area and the number of hairpins.

Because of the space limitation, assume that maximum number of hairpins should not be more than 10. Is this condition satisfied? If not, suggest a new design for a solution for the use of double pipe heat exchanger.

GIVEN:

Tube side (water):

-Inlet water temperature (T_{c1}) = 30°C

-Outlet water temperature (T_{c2}) = 50°C

-Diameters: d_o = 25 mm

-Heat transfer coefficient (h_i) = 3000 W/m^2.K

-Length of the hairpin (L) = 4 m

-Specific heat of water ($c_{p,c}$) = 4178 J/kg.K

-Limitation on number of hairpins = 10.

Annulus (oil):

-Inlet temperature (T_{h1}) = 60°C

-Outlet temperature (T_{h2}) = 40^{o}C

-Mass flow rate ($\dot{m}_h$) = 0.4 kg/s

-Specific heat of oil ($c_{p,h}$) = 2000 J/kg.K

-Diameters: D_i = 45 mm

FIND:

a. Heat transfer coefficient in annulus (h_o)

b. Heat transfer area (A_o) and the number of hairpins (N_{hp})

Length of hairpins (L), and suggest a new design if the length limitation doesn't satisfied.

SOLUTION:

a.

Properties of oil at $\frac{60+40}{2} = 50\ ^oC$:

$$\rho = 870\ kg/m^3 \qquad \mu = 0.124\ N.s/m^2$$
$$k = 0.14\ \ W/m^2K \qquad \mu_w = 0.22\ \ N.s/m^2 \qquad Pr = 1761$$

Log-mean temperature:

$$\Delta T_1 = \Delta T_2 = \Delta T_{lm} = 10^o C$$

Equivalent diameter for heat transfer:

$$D_e = \frac{4A_e}{\pi d_o} = \frac{D_i{}^2 - d_o{}^2}{d_o} = \frac{45^2 - 25^2}{25} = 56\ \ mm = 0.056\ m$$

Hydraulic diameter for pressure drop analysis:

$$D_h = D_i - d_o = 45 - 25 = 20\ \ mm = 0.02\ m$$

Flow velocity:

$$A_c = \frac{\pi}{4}\left(D_i{}^2 - d_o{}^2\right) = 1099.6\ mm^2 = 1099.6 \times 10^{-6}\ \ m^2$$

$$u_m = \frac{\dot{m}_h}{\rho_h A_c} = \frac{0.4}{870 \times 1099.6 \times 10^{-6}} = 0.42\ \ m/s$$

$$Re = \frac{\rho u_m D_h}{\mu} = \frac{870 \times 0.42 \times 0.02}{0.124} = 58.9\ \ \text{(Laminar flow)}$$

One can use Sieder-Tate correlation:

$$Nu_a = 1.86\left(Re\,Pr\frac{D_h}{L}\right)^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0/14}$$

Check validity:

$$\left(Re\,Pr\frac{D_h}{L}\right)^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.14} = \left(58.7\times1761\times\frac{0.02}{4}\right)^{1/3}\left(\frac{0.124}{0.22}\right)^{0.14} = 7.4 \geq 2,$$

so this correlation is valid for the given data.

$$Nu_T = 1.86\left(Re\,Pr\frac{D_h}{L}\right)^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0/14} = 1.86\times7.4 = 13.76$$

$$h_o = \frac{Nu\cdot k}{D_e} = \frac{13.76\times0.14}{0.056} = 34.4\ W/m^2.K$$

b.

Overall heat transfer coefficient:

$$U_o = \frac{1}{\dfrac{r_o}{r_i h_i} + r_o\dfrac{\ln(r_p/r_i)}{k} + \dfrac{1}{h_o}} \approx \frac{1}{\dfrac{1}{h_i}+\dfrac{1}{h_o}} = \frac{1}{\dfrac{1}{3000}+\dfrac{1}{34.4}} = 34\ W/m^2.K$$

$$A_o = \frac{Q}{U_o \Delta T_{lm}}$$

where $A_o = 2\pi d_o L N_{hp}$

Surface area of a hairpin:

$$A_{hp} = 2\pi d_o L$$

Heat balance:

$$Q = \dot{m}_c c_{p,c}(T_{c2}-T_{c1}) = \dot{m}_h c_{p,h}(T_{h2}-T_{h1}) = 0.4\times2000\times(60-40) = 16,000\ W$$

$$N_{hp} = \frac{A_o}{A_{hp}} = \frac{\dfrac{Q}{U_o\Delta T_{lm}}}{2\pi d_o L} = \frac{\dfrac{16000}{34\times10}}{2\pi\times0.025\times4} = 74.89$$

More than 10 hairpins are needed to satisfy the heat duty specified, so finned-tube can be applied to increase the heat transfer area in annulus side, where the heat transfer coefficient is too low.

Problem 7.10

In example 7.1, assume that the mass flow rate of cold fluid is increased to 20,000 kg/h. The remaining geometric parameter and the process specifications remain the same. In this case pressure drop will increase. Assume that the allowable pressure drop in the annulus is 10 kPa. The velocity of cold water through the annulus will be around 3 m/s, which would require a large pressure drop to drive the cold fluid through the inner tube. The pressure drop could be reduced by using several units with cold water flowing in four parallel units and hot water flowing through the inner tube in series. Calculate:

a. Number of hairpins

b. Pressure drop in annulus

c. The appropriate mean temperature difference

GIVEN:

-Several double pipe heat exchangers, with cold water flowing in parallel units and hot water flowing through the inner tube in series.

-Cold water inlet temperature (T_{c1}) = 20 °C

-Cold water outlet temperature (T_{c2}) = 35 °C

-Mass flow rate of oil ($\dot{m}_c$) = 20,000 kg/h = 5.556 kg/s

-Hot water inlet temperature (T_{h1}) = 140 °C

-Hot water temperature drop (ΔT_h) = 15 °C

-Allowable pressure drop in annulus (Δp_a) = 10 kPa

-I.D. of inner tube (d_i) = 0.0525 m

-O.D. of inner tube (d_o) = 0.0603 m

-Inside diameter of outer annulus (D_i) = 0.0799 m

-Length of the hairpins (L) = 3.5 m

-Fouling factors: R_{fi} = 0.000176 m^2.K/W

R_{fo} = 0.000352 m^2.K/W

FIND:

a. Number of hairpins (N_{hp})

b. Pressure drop in annulus(Δp_a)

c. The appropriate mean temperature difference

SOLUTION:

Properties of water at $\frac{140+125}{2} = 132.5\ ^oC$ is:

$$\rho = 932.53\ kg/m^3 \qquad \mu = 0.207\times10^{-3}\ N.s/m^2$$
$$Pr = 1.28 \qquad k = 0.687\ W/m.K$$
$$c_p = 4268\ J/kg.K$$

Properties of water at $\frac{35+20}{2} = 27.5\ ^oC$ is:

$$\rho = 996.4\ kg/m^3 \qquad \mu = 0.841\times10^{-6}\ N.s/m^2$$
$$Pr = 5.77 \qquad k = 0.609\ W/m.K$$
$$c_p = 4179\ J/kg.K$$

a.

Calculate hot water mass flow rate by:

$$Q = \dot{m}_c c_{p,c}\left(T_{c2} - T_{c1}\right) = \dot{m}_h c_{p,h}\left(T_{h2} - T_{h1}\right)$$

$$\dot{m}_h = \frac{\dot{m}_c c_{p,c}\left(T_{c2} - T_{c1}\right)}{c_{p,h}\left(T_{h2} - T_{h1}\right)} = \frac{(20{,}000/3600)\times4179\times(35-20)}{4268\times15} = 5.44\ m/s$$

$$u_m = \frac{\dot{m}_h}{\rho_h A_c} = \frac{\dot{m}_h}{\rho\frac{\pi d_i^2}{4}} = \frac{5.44}{932.53\times\frac{\pi\times0.0525^2}{4}} = 2.695\ m/s$$

$$Re = \frac{\rho u_m d_i}{\mu} = \frac{932.53\times2.695\times0.0525}{0.207\times10^{-3}} = 637{,}398 \quad \text{(Turbulent flow)}$$

Prandtl's correlation is used here with constant properties:

$$f = \left(1.58\ln Re - 3.28\right)^{-2} = \left(1.58\times\ln 637398 - 3.28\right)^{-2} = 0.00314$$

$$Nu = \frac{(f/2)RePr}{1 + 8.7(f/2)^{\frac{1}{2}}(Pr - 1)}$$

$$= \frac{\left(\frac{0.00314}{2}\right) \times 637398 \times 4.44}{1 + 8.7 \times \left(\frac{0.00314}{2}\right)^{\frac{1}{2}}(1.28 - 1)}$$

$$= 1168 \ \ W/m^2.K$$

The inside heat transfer coefficient is:

$$h_i = \frac{Nu \cdot k}{d_i} = \frac{1168 \times 0.687}{0.0525} = 15284 \ \ W/m^2.K$$

Heat transfer coefficient in annulus:

$$u_m = \frac{\dot{m}_c}{\rho_c (nA_c)} = \frac{\dot{m}_c}{\rho\left(4 \times \frac{\pi d_i^{\,2}}{4}\right)} = \frac{(20{,}000/3600)}{996.4 \times 4 \times \frac{\pi \times (0.0779^2 - 0.0603^2)}{4}} = 0.73 \ m/s$$

where n is the number of units in parallel connection in annulus side.

$$D_h = \frac{4A_c}{P_w} = D_i - d_o = 0.0779 - 0.0603 = 0.0176 \ \ m$$

$$Re = \frac{\rho_c u_m D_h}{\mu} = \frac{996.4 \times 0.73 \times 0.0176}{0.841 \times 10^{-3}} = 15{,}222 \quad \text{(Turbulent flow)}$$

Prandtl’s correlation is used again for the annulus:

$$f = (3.64 \log Re - 3.28)^{-2} = (3.64 \times \log 15222 - 3.28)^{-2} = 0.007$$

$$Nu = \frac{(f/2)RePr}{1 + 8.7(f/2)^{\frac{1}{2}}(Pr - 1)}$$

$$= \frac{\left(\frac{0.007}{2}\right) \times 15222 \times 5.77}{1 + 8.7 \times \left(\frac{0.007}{2}\right)^{\frac{1}{2}}(5.77 - 1)}$$

$$= 88.97 \ \ W/m^2.K$$

The hydraulic diameter for heat transfer is:

$$D_e = \frac{D_i^{\,2} - d_o^{\,2}}{d_o} = \frac{0.0779^2 - 0.0603^2}{0.0603} = 0.0403 \ \ m$$

The annulus heat transfer coefficient is:

$$h_o = \frac{Nu \cdot k}{D_e} = \frac{88.97 \times 0.609}{0.0403} = 1344 \ \ W/m^2.K$$

The overall heat transfer coefficient based on the outside area of the inner tube:

$$U_o = \frac{1}{\frac{d_o}{d_i h_i} + \frac{d_o R_{fi}}{d_i} + \frac{d_o \ln(d_o/d_i)}{2k} + R_{fo} + \frac{1}{h_o}}$$

$$= \frac{1}{\frac{0.0603}{0.0525 \times 15284} + \frac{0.0603 \times 0.000176}{0.0525} + \frac{0.0603 \ln(603/525)}{2 \times 54} + 0.000352 + \frac{1}{1344}}$$

$$= 702 \text{ W/m}^2.\text{K}$$

The heat transfer surface is:

$$A_o = \frac{Q}{U_o \Delta T_m} = \frac{\dot{m}_h c_{p,h}(T_{h2} - T_{h1})}{U_o \Delta T_m} = \frac{5.556 \times 4179 \times (35 - 20)}{702 \times 105} = 4.72 \text{ m}^2$$

where $\Delta T_m = \Delta T_1 = \Delta T_2 = 105\ ^oC$

The heat transfer area per hairpin is:

$$A_{hp} = \pi \times d_o \times 2L = 2\pi \times 0.0603 \times 3.5 = 1.326 \text{ m}^2$$

$$\frac{A_o}{A_{hp}} = \frac{4.72}{1.326} = 3.56$$

Therefore, the number of hairpins, $N_{hp} = 4$.

b.

Pressure drop in the annulus side is calculated by:

$$\Delta p_a = 4f \frac{2L}{D_h} \rho \frac{u_m^2}{2} N_{hp}$$

$$= 4 \times 0.007 \times \frac{2 \times 3.5}{0.0176} \times 996.4 \times \frac{0.73^2}{2} \times 4 = 11826 \text{ Pa}$$

c.

The appropriate mean temperature difference

$\Delta T_m = \Delta T_1 = \Delta T_2 = 105\ ^oC$

Problem 7.11

A sugar solution (ρ = 1080 kg/m^3, c_p = 3601 J/kg.K, k_f = 0.5764 W/m.K, μ = 1.3×10^{-3} N.s/m^2) flows at rate of 2 kg/s and is to be heated from 25 °C to 50 °C. Water at 95 °C is available at a flow rate of 1.5 kg/s (c_p = 4004 J/kg.K). Sugar solution flows through annulus. It is proposed to design a double pipe heat exchanger for this process.

The following geometrical parameters are proposed:

The length of the hairpin = 3 m

D_o(annulus) = 6 cm

D_i (annulus) = 5.25 cm

d_o (inner tubes) = 2.6 cm

d_i (inner tubes) = 2.09 cm

# of tubes inside the annulus = 1

of the fins = 20

Fin height = 0.0127 m

Fin thickness = 0.0009 m

Thermal conductivity of the material, k = 52 W/m.K.

a. Calculate the hydraulic diameter of the annulus of the pressure drop analysis.

b. Calculate the equivalent diameter for heat transfer analysis.

c. Calculate the heat transfer surface area of this heat exchanger and number of hairpins.

GIVEN:

-A double pipe heat exchanger, with sugar solution flowing through annulus.

-Organic vapor temperature (T_{h1}) = 95°C

-Mass flow rate of hot water ($\dot{m}_h$) = 1.5 kg/s

-Cold fluid inlet temperature (T_{c1}) = 25°C

-Cold fluid outlet temperature (T_{c2}) = 50°C

-Mass flow rate of sugar solution ($\dot{m}_c$) = 2 kg/s

-Overall surface efficiency (η_o) = 0.90

-Thermal conductivity of the material (k) = 52 W/m.K

-Geometrical parameters listed in the statement of the problem.

FIND:

a. Hydraulic diameter of the annulus for pressure drop analysis (D_h).

b. Equivalent diameter for heat transfer analysis (D_e)

c. Heat transfer surface area (A_o) and # of hairpins (N_{hp})

SOLUTION:

a.

Annulus:

The cross-sectional area in the annulus with longitudinal finned tubes is:

$$A_c = \frac{\pi}{4}\left(D_i^2 - d_o^2 N_t\right) - \left(\delta H_f N_f N_t\right)$$
$$= \frac{\pi}{4}\left(0.0525^2 - 0.026^2 \times 1\right) - \left(0.0009 \times 0.0127 \times 20 \times 1\right)$$
$$= 1.405 \times 10^{-3} \ \ \text{m}^2$$

Hydraulic diameter:

$$D_h = \frac{4A_c}{P_w}$$

$$P_w = \pi\left(D_i + d_o N_t\right) + 2H_f N_f N_t$$
$$= \pi \times \left(0.0525 + 0.026 \times 1\right) + 2 \times 0.0127 \times 20 \times 1$$
$$= 0.755 \ \ \text{m}$$

$$\therefore D_h = \frac{4 \times 1.405 \times 10^{-3}}{0.755} = 6.04 \times 10^{-3} \ \text{m}$$

b.

$$P_h = \pi d_o N_t + 2H_f N_f N_t$$
$$= \pi \times 0.026 \times 1 + 2 \times 0.0127 \times 20 \times 1$$
$$= 0.59 \ \ \text{m}$$

$$D_e = \frac{4A_c}{P_h} = \frac{4 \times 1.405 \times 10^{-3}}{0.59} = 9.525 \times 10^{-3}$$

Sugar solutions flows through the annulus:

$$u_m = \frac{\dot{m}_c}{\rho A_c} = \frac{2}{1080 \times 1.405 \times 10^{-3}} = 1.318 \ \text{m/s}$$

$$\Pr = \frac{\mu c_p}{k} = \frac{1.3 \times 10^{-3} \times 3601}{0.5764} = 8.12$$

$$\text{Re} = \frac{\rho u_m D_h}{\mu} = \frac{1080 \times 1.318 \times 7.44 \times 10^{-3}}{1.3 \times 10^{-3}} = 8146.46 \quad \text{(Turbulent flow)}$$

Genielinski's correlation:

$$f = (1.58 \ln \text{Re} - 3.28)^{-2} = (1.58 \times \ln 8146.46 - 3.28)^{-2} = 8.34 \times 10^{-3}$$

$$\text{Nu} = \frac{(f/2)(\text{Re} - 1000)\Pr}{1 + 12.7(f/2)^{\frac{1}{2}}\left(\Pr^{\frac{2}{3}} - 1\right)}$$

$$= \frac{\left(\frac{0.00834}{2}\right)(8146.46 - 1000) \times 8.12}{1 + 12.7 \times \left(\frac{0.00834}{2}\right)^{\frac{1}{2}}\left(8.12^{\frac{2}{3}} - 1\right)}$$

$$= 69.28 \ \text{W/m}^2.\text{K}$$

$$h_o = \frac{\text{Nu} \cdot k}{D_e} = \frac{69.28 \times 0.5764}{9.525 \times 10^{-3}} = 4192.44 \ \text{W/m}^2.\text{K}$$

Finned and unfinned heat transfer area:

$$A_f = 2N_f N_t (2H_f + \delta) \cdot L$$

$$A_u = 2N_t L (d_o - N_f \delta)$$

$$A_t = A_f + A_u = 2N_f N_t (2H_f) \cdot L + 2N_t L \pi d_o$$

$$= 2 \times 1 \times 20 \times 2 \times 0.0127 \times 3 + 2 \times 1 \times 3 \times \pi \times 0.026$$

$$= 3.538 \ \text{m}^2$$

Overall heat transfer coefficient

$$U_o = \frac{1}{\frac{A_t}{A_i h_i} + \frac{A_t \ln(d_o/d_i)}{2k\pi(2L)} + \frac{1}{\eta_o h_o}}$$

$$= \left[\frac{3.538}{\pi \times 2 \times 3 \times 0.0209} \cdot \frac{1}{4000} + \frac{3.538 \times \ln\left(\frac{0.026}{0.0209}\right)}{2 \times 52 \times \pi \times 6} + \frac{1}{0.9 \times 4192.44}\right]^{-1}$$

$$= 344.32 \ \text{W/m}^2.\text{K}$$

c.

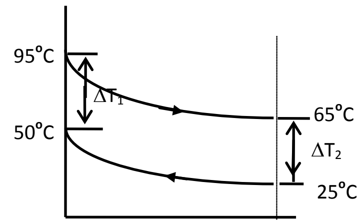
Heat balance:

$$Q = \dot{m}_c c_{p,c}\left(T_{c2} - T_{c1}\right) = \dot{m}_h c_{p,h}\left(T_{h1} - T_{h2}\right)$$

$$T_{h2} = T_{h1} - \frac{\dot{m}_c c_{p,c}\left(T_{c2} - T_{c1}\right)}{\dot{m}_h c_{p,h}}$$

$$= 95 - \frac{2 \times 3601 \times (50 - 25)}{1.5 \times 4004}$$

$$= 65\ ^oC$$



Log-mean temperature difference:

$$\Delta T_{lm,cf} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} = \frac{45 - 40}{\ln \frac{45}{40}} = 42.45\ ^oC$$

$$Q = \dot{m}_c c_{p,c}\left(T_{c2} - T_{c1}\right) = 2 \times 3601 \times (50 - 25) = 180050\ \ W$$

$$A_o = \frac{Q}{U_o \Delta T_m} = \frac{180050}{344.32 \times 42.45} = 12.318\ \ m^2$$

Number of hairpins:

$$N_{hp} = \frac{A_o}{A_t} = \frac{12.318}{3.538} = 3.48 \approx 4$$

Problem 8.1

Repeat example 8.1 for the refrigerant-22 and compare with R-134A.

GIVEN:

-Quiescent refrigerant 22 vapor at saturation temperature of 47 °C

-Wall temperature: 40 °C

-O.D. of tube: d_o = 19 mm

SOLUTION:

Properties of saturated R-22 at T_{sat} = 47 °C are (Table B.8, Appendix B):

$$\rho_l = 997.7 \ \text{kg/m}^3 \qquad c_{p_g} = 4.181 \ \text{kJ/kg.K}$$

$$k_l = 0.602 \ \text{W/mK} \qquad \mu_l = 959 \times 10^{-6} \ \text{Pa.s}$$

$$\text{Pr} = 6.68$$

Using Eg. (8.2):

$$h_l = \frac{k_l}{d_o}(0.728)\left[\frac{\rho_l\left(\rho_l - \rho_g\right)g \cdot i_{lg} d_o{}^3}{\mu_l\left(T_{sat} - T_w\right)k_l}\right]^{1/4}$$

$$h_l = \frac{0.077}{0.019} \times 0.728 \times \left[\frac{1098.6 \times \left(1098.6 - 79.05\right) \times 9.81 \times 158.1 \times 10^3 \times 0.019^3}{1.76 \times 10^{-4} \times \left(47 - 40\right) \times 0.077}\right]^{1/4}$$

$$= 1738.6 \ \text{W/m}^2\text{.K}$$

Comparing with R-134A, the average heat transfer coefficient is a little bit higher with R-22.

Problem 8.2

Repeat Example 8.2 for the refrigerant-22 and compare with R-134A.

GIVEN:

-R-22 moving downward over the tube at a velocity of 10 m/s at saturation temperature of 47 °C.

-Wall temperature: 40 °C

-O.D. of tube: d_o = 19 mm

SOLUTION:

Properties of R-22 at T_{sat} = 47 °C:

$$\rho_l = 1098.6 \text{ kg/m}^3 \qquad \rho_g = 79.05 \text{ kg/m}^3$$
$$k_l = 0.077 \text{ W/mK} \qquad \mu_l = 1.76 \times 10^{-4} \text{ Pa.s}$$
$$i_{lg} = 158.1 \text{ kJ/kg}$$

Using Eg. (8.7):

$$h_m = \frac{k_l}{d_o}(0.416)\left(1+\left(1+9.47F\right)^{1/2}\right)^{1/2} \tilde{R}e^{1/2}$$

Where F is given by Equation (8.5):

$$F = \frac{gd\mu_l i_{lg}}{u_g{}^2 k_l \Delta T}$$

$$F = \frac{9.81 \times 0.019 \times 1.76 \times 10^{-4} \times 158.1 \times 10^3}{10^2 \times 0.068 \times (47-40)} = 0.109$$

The two-phase Reynolds number is

$$\tilde{R}e = \frac{\rho_l u_g d}{\mu_l} = \frac{1098.6 \times 10 \times 0.019}{1.76 \times 10^{-4}} = 1{,}185{,}989$$

Then the average heat transfer coefficient is obtained as:

$$h_m = \frac{0.077}{0.019} \times (0.416) \times \left(1+\left(1+9.47 \times 0.109\right)^{1/2}\right)^{1/2} (1{,}185{,}989)^{1/2}$$
$$= 2859 \text{ W/(m}^2\text{K)}$$

Comparing with R-134A under the same condition, the average heat transfer coefficient is a little higher with R-22.

Problem 8.3

Under the conditions given in Example 8.4, calculate the local heat transfer coefficient for the 20th row of tubes using the method given by Butteworth.

GIVEN:

-Saturation temperature of steam: T_{sat} = 373.15 K = 100 °C

-N = 320 tubes

-The tube are arranged in a square, in-line pitch (p = 35.0 mm)

-d_o = 0.03 m

-T_w = 93 °C

- Mass flow rate $\dot{m}_g$ = 14.0 kg/s

SOLUTION:

Properties of steam at T_{sat} = 100 °C:

$$\rho_l = 957.9 \text{ kg/m}^3 \qquad \rho_g = 0.598 \text{ kg/m}^3$$

$$k_l = 0.681 \text{ W/mK} \qquad \mu_l = 2.79\times10^{-4} \text{ Pa.s}$$

$$\mu_g = 1.2\times10^{-5} \text{ kg/m.s} \qquad i_{lg} = 2257 \text{ kJ/kg}$$

$$c_{p,l} = 4.219 \text{ kJ/kg.K} \qquad \Pr_l = 1.73$$

Mean flow area:

$$A_m = wN_tL$$

where w is given by Equation (8.14):

$$w = \frac{p_L p_t - \pi d^2/4}{p_L} = \frac{0.035^2 - \pi\times0.03^2/4}{0.035} = 0.0148 \text{ m}$$

$$A_m = 0.0148\times16\times4 = 0.947 \text{ m}^2$$

The local steam velocity is:

$$u_g = \frac{14}{0.598\times0.947} = 24.7 \text{ m/s}$$

The local heat transfer coefficient is given by Equation (8.16):

$$h_N = \left[\frac{1}{2}h_{sh}^{\ 2} + \left(\frac{1}{4}h_{sh}^{\ 4} + h_l^{\ 4}\right)^{1/2}\right]^{1/2} \times \left[N^{5/6} - (N-1)^{5/6}\right]$$

where

$$h_{sh} = 0.59\frac{k_l}{d}\widetilde{Re}^{1/2}$$

and

$$h_l = \frac{k_l}{d_o}(0.728)\left[\frac{\rho_l(\rho_l - \rho_g)g \cdot i_{lg} d_o^{\ 3}}{\mu_l(T_{sat} - T_w)k_l}\right]^{1/4}$$

The two-phase Reynolds number is:

$$\widetilde{Re} = \frac{\rho_l u_g d}{\mu_l} = \frac{957.9 \times 24.7 \times 0.03}{2.79 \times 10^{-4}} = 2.54 \times 10^6$$

So

$$h_{sh} = 0.59\frac{0.681}{0.03} \times \left(2.54 \times 10^6\right)^{1/2} = 21{,}345 \ \ W/(m^2K)$$

$$h_l = \frac{0.681}{0.03} \times (0.728) \times \left[\frac{957.9 \times (957.9 - 0.598) \times 9.81 \times 2.257 \times 10^6 \times 0.03^3}{2.79 \times 10^{-4} \times (100 - 93) \times 0.681}\right]^{1/4}$$

$$= 13{,}241 \ \ W/(m^2K)$$

Therefore, with N=20, the local heat transfer coefficient is:

$$h_N = \left[\frac{1}{2} \times 21345^2 + \left(\frac{1}{4} \times 21345^4 + 13241^4\right)^{1/2}\right]^{1/2} \times \left[20^{5/6} - (20-1)^{5/6}\right]$$

$$= 11530 \ \ W/(m^2K)$$

Problem 8.4

Calculate the average heat transfer coefficient for film-type condensation of water at pressure of 10 kPa for:

a. An outside surface of 19-mm O.D. horizontal tubes 2-m long

b. A 12-tube vertical bank of 19-mm horizontal tubes 2-m long

It is assumed that the vapor velocity is negligible and the surface temperatures are constant at 10 °C below saturation temperature.

GIVEN:

-P_{sat} = 0.1 bar, T_{sat} = 333.23 K = 60.08 °C

-d = 0.019 m

-L = 2 m

-T_w = 323.23 K = 50.08 °C

SOLUTION:

Properties of steam at $P_{sat} = 0.1$ bar:

$$\rho_l = 997.7 \ \ \mathrm{kg/m^3} \qquad c_{p_g} = 4.181 \ \ \mathrm{kJ/kg.K}$$

$$k_l = 0.602 \ \ \mathrm{W/mK} \qquad \mu_l = 959 \times 10^{-6} \ \ \mathrm{Pa.s}$$

$$\mathrm{Pr} = 6.68$$

a.

Using Eq. (8.2) for condensation on a single horizontal tube:

$$h_m = \frac{k_l}{d_o}(0.728)\left[\frac{\rho_l(\rho_l - \rho_g)g \cdot i_{lg} d_o^{\ 3}}{\mu_l(T_{sat} - T_w)k_l}\right]^{1/4}$$

$$= \frac{0.635}{0.019} \times 0.728 \times \left[\frac{989.8 \times (989.8 - 0.0682) \times 9.81 \times 2.392 \times 10^3 \times 0.019^3}{5.88 \times 10^{-4} \times (10) \times 0.635}\right]^{1/4}$$

$$= 1961.3 \ \ \mathrm{W/m^2.K}$$

b.

Using Kern's correlation to account for the effect of Condensate Inundation in a tube bundle:

$$\frac{h_{m,N}}{h_1} = N^{-1/6}$$

$$\therefore h_{m,N} = h_1 N^{-1/6} = 1961.3 \times 12^{-1/6} = 1336.2 \ \ \mathrm{W/(m^2K)}$$

Problem 8.5

A horizontal 2-cm O.D. tube is maintained at a temperature of 27 °C on its outer surface. Calculate the average heat transfer coefficient if saturated steam at 6.22 kPa is condensing on this tube.

GIVEN:

-Steam saturation pressure P_{sat} = 6.22 kPa

-Tube wall temperature T_w = 27 °C

-d_o = 0.02 m

SOLUTION:

Properties of steam at 6.22 kPa are:

$$\rho_l = 997.7 \ \text{kg/m}^3 \qquad c_{p_g} = 4.181 \ \text{kJ/kg.K}$$

$$k_l = 0.602 \ \text{W/mK} \qquad \mu_l = 959 \times 10^{-6} \ \text{Pa.s}$$

$$\text{Pr} = 6.68$$

Using Eg. (8.2):

$$h_l = \frac{k_l}{d_o}(0.728)\left[\frac{\rho_l(\rho_l - \rho_g)g \cdot i_{lg} d_o^{\ 3}}{\mu_l(T_{sat} - T_w)k_l}\right]^{1/4}$$

$$h_l = \frac{0.673}{0.02} \times 0.728 \times \left[\frac{967.4 \times (967.4 - 0.376) \times 9.81 \times 2.291 \times 10^3 \times 0.02^3}{3.27 \times 10^{-4} \times (359.94 - 300.15) \times 0.673}\right]^{1/4}$$

$$= 1465 \ \text{W/m}^2.\text{K}$$

Problem 8.6

In a shell-and-tube type steam condenser, assume that there are 81 tubes arranged in a square pitch with 9 tubes per column. The tubes are made of copper with an outside diameter of 1 in. The length of the condenser is 1.5 m. The shell-side condensation occurs at saturation pressure. Water flows inside the tubes with a mass flow rate of 4 kg/s. The tube outside wall temperature is 90°C. With reasonable assumptions, perform thermal and hydraulic analysis of the condenser.

<u>GIVEN:</u>

The fluid conditions are as follows:

-Refrigerant flow rate 19.3 kg/hr (per tube path)

-Refrigerant inlet temperature $-5^{o}C$

-Refrigerant inlet quality 0.1

-Refrigerant exit condition $0^{o}C$ superheat

-Air velocity 2 m/s

-Air inlet temperature $15^{o}C$

The geometrical parameters are as follows:

-Tube inside diameter 0.00826 m

-Tube outside diameter 0.00952 m

-Width of heat exchanger 1.25 m

-Number of rows 4

-Number of independent circuits 4

-Horizontal tube spacing 0.0222 m

-Vertical tube spacing	0.025 m
-Number of tubes vertical to air flow direction	12
-Number of tubes along air flow direction	4
-Fin thickness	1.6154×10^{-4} m
-Fin pitch	551.18 m
-Fin surface area/total outside surface area	0.948
-Free air-flow area/frontal area	0.5466

SOLUTION:

Properties of R-134a at -5°C are:

$$\rho_1 = 997.7 \ \text{kg/m}^3 \qquad c_{p_g} = 4.181 \ \text{kJ/kg.K}$$

$$k_1 = 0.602 \ \text{W/mK} \qquad \mu_1 = 959\times10^{-6} \ \text{Pa.s}$$

$$\text{Pr} = 6.68$$

The evaporation heat transfer coefficient using kandlikar's correlation Eq.(8.89) is:

$$h_{TP} = C_1(\text{Co})^{C_2}(25\text{Fr}_l)^{C_5} h_l + C_3(\text{Bo})^{C_r} F_{fl} h_l$$

The convection number, Co, is given by Eq.(8.67):

$$\text{Co} = \left(\frac{1-x}{x}\right)^{0.8}\left(\frac{\rho_g}{\rho_l}\right)^{0.5} = \left(\frac{1-0.1}{0.1}\right)^{0.8}\left(\frac{12.2}{1308}\right)^{0.5} = 0.56$$

Boiling number, Bo, is given by Eq. (8.68):

$$\text{Bo} = \frac{q''}{G\cdot h_{lg}} = \frac{7.4\times10^3}{100\times202.3\times10^3} = 3.66\times10^{-4}$$

Froude number, Fr, is given by Eq. (8.65):

$$\text{Fr}_l = \frac{G^2}{\rho_l^2 g d_i} = \frac{100^2}{1308^2\times9.81\times0.00826} = 0.0721$$

Liquid Reynolds number, Eq. (8.84):

$$\text{Re}_l = \frac{G(1-x)d_i}{\mu_l} = \frac{100\times(1-0.1)\times0.00826}{301\times10^{-6}} = 2470$$

The liquid-only heat transfer coefficient can be calculated using the Dittus-correlation, Eq. (8.56):

$$h_l = 0.023\text{Re}^{0.8}\text{Pr}^{0.4}\frac{k_e}{d_i} = 0.023\times 2470^{0.8}\times 3.98^{0.4}\times\frac{0.0981}{0.00826} = 245.77\ \ \text{W/m}^2.\text{K}$$

The constants C_1 through C_5 are given in Table 8.3 for Co< 0.65 as:

C_1 = 1.136, C_2 = -0.9, C_3 = 667.2, C_4 = 0.7, C_5 = 0.4

$$h_{TP} = \left[1.136\times 0.56^{-0.9}\times(25\times 0.0722)^{0.4} + 667.2\times\left(3.64\times 10^{-4}\right)^{0.7}\times 1.5\right]\times 245.76$$

$$= 1558.82\ \ \text{W/m}^2.\text{K}$$

F_{fl} =1.5 for R-134a.

Problem 8.7

A water -cooled, shell-and-tube freon condenser with in-tube condensation will be designed to satisfy the following specifications:

Cooling load of the condenser:	**125 kW**
Refrigerant:	**R-22**
Condensing temperature:	**37°C**
Coolant water:	**City water**
inlet temperature:	**18°C**
Outlet temperature:	**26°C**
Mean pressure:	**0.4 Mpa**
Heat transfer matrix:	**3/4 inch OD, 20 BWG**
	Brass tubes

If it is proposed that the following heat exchanger parameters are fixed: one-tube pass with shell diameter of 15 1/4 in., pitch size is 1 in. with baffle spacing of 35 cm. The number of tubes is 137.

a. Calculate the shell-and-tube size heat transfer coefficients.

b. By assuming proper fouling factors, calculate the length of the condenser.

c. Space available is 6 m. Is the design acceptable?

SOLUTION:

For preliminary design:

Assumed values for the shell-and-tube condenser:

N_p = 1 tube pass

D_s = 15.25″ = 0.387 m

N_T = 137 tubes

p = 1″ square pitch

B = 0.35 m baffle spacing

Properties of shell-side coolant:

Fluid: City water

T_{c1} = 18°C (inlet temperature)

T_{c2} = 26°C (outlet temperature)

d_o = 0.75 ″= 0.01905 m

Bulk temperature:

$T_b = (T_{c1} + T_{c2})/2 = (18+26)/2 = 22\ ^oC$

Properties at bulk temperature from Appendix:

$$\rho_l = 997.7\ \text{kg/m}^3 \qquad c_{p_g} = 4.181\ \text{kJ/kg.K}$$

$$k_l = 0.602\ \text{W/mK} \qquad \mu_l = 959 \times 10^{-6}\ \text{Pa.s}$$

$$\text{Pr} = 6.68$$

Cooling water flow rate (shell-side):

$$Q = \dot{m}_c c_{p,c}\left(T_{c2} - T_{c1}\right)$$

$$\dot{m}_c = \frac{Q}{c_{p,c}\left(T_{c2} - T_{c1}\right)} = \frac{125000}{4.181 \times (26 - 18)} = 3.74\ \text{kg/s}$$

Properties of tube side - condensing:

Fluid: R-22

$T_H = T_{sat}$ = 37 °C (saturation temperature)

d_i = 0.68″ = 0.01727 m

A_c = 0.3632 in^2 = 2.34×10^{-4} m^2

Properties at saturation temperature from Appendix:

$$P_{sat} = 14.17\ \text{bar} \qquad c_{pl} = 1.305\ \text{kJ/kg.K}$$

$$\rho_l = 1145\ \text{kg/m}^3 \qquad \rho_g = 60.9\ \text{m}^3/\text{kg}$$

$$\mu_l = 0.000186\ \text{Pa.s} \qquad \mu_g = 0.0000141\ \text{Pa.s}$$

$$k_l = 0.082\ \text{W/m.K} \qquad \text{Pr} = 2.96$$

$$i_{fg} = 169.6\ \text{kJ/kg}$$

Refrigerant flow rate (in-tube):

$$\dot{m}_R = \frac{Q}{i_{fg}} = \frac{125}{169.6} = 0.737\ \text{kg/s}$$

a.

Shell-side calculations:

Calculate A_s = shell-side area by calculating fictitious cross-sectional area:

$$A_s = \frac{D_s CB}{P_T} = \frac{0.387 \times (0.0254 - 0.01905) \times 0.35}{0.0254} = 0.0339\ \text{m}^2$$

Mass flux:

$$G_s = \frac{\dot{m}_c}{A_s} = \frac{3.74}{0.0339} = 110.3\ \text{kg/m}^2.\text{s}$$

Equivalent diameter, D_e:

Assuming square pitch:

$$D_e = \frac{4\left(P_T^2 - \frac{\pi d_o^2}{4}\right)}{\pi d_o} = \frac{4\left(0.0254^2 - \frac{\pi \times 0.01905^2}{4}\right)}{\pi \times 0.01905} = 0.0241 \text{ m}$$

Reynolds number, shell-side:

$$Re_s = \frac{G_s d_o}{\mu} = \frac{110.3 \times 0.0241}{959 \times 10^{-6}} = 2772 \text{ (Turbulent flow on shell side)}$$

Shell-side heat transfer coefficient:

For 2000 < Re < 10^5

$$\frac{h_o D_e}{k} = 0.35 Re_s^{0.55} Pr_s^{1/3} \left(\frac{\mu_b}{\mu_w}\right)^{0.14}$$

where μ_b/μ_w is assumed to be 1 because of the small temperature difference.

$$h_o = 0.35 \times 2772^{0.55} \times 6.68^{1/3} \left(\frac{0.602}{0.0241}\right) = 1288 \text{ W/m}^2.\text{K}$$

Tube-side calculations:

Mass flux (refrigerant):

$$A_R = \frac{A_c N_T}{N_p} = \frac{0.000234 \times 137}{1} = 0.0321 \text{ m}^2$$

$$G_R = \frac{\dot{m}_R}{A_R} = \frac{0.737}{0.0321} = 22.96 \text{ kg/m}^2.\text{s}$$

where A_c = cross-sectional tube area.

Reynolds number:

$$Re_L = \frac{G_R (1-x) d_i}{\mu_l}$$

For a quality of x = 0.5:

$$Re_L = \frac{22.96 \times (1-0.5) \times 0.01727}{186 \times 10^{-6}} = 1066$$

As first assumption, the length of the tube is 6m:

$$Bo = \frac{q''}{G \cdot i_{fg}} = \frac{Q}{A_w \cdot G \cdot i_{fg}} = \frac{Q}{(\pi d_i L N_t) \cdot G \cdot i_{fg}}$$

$$= \frac{125}{(\pi \times 0.01727 \times 6 \times 137) \times 22.96 \times 169.6} = 0.000718$$

$$Co = \left(\frac{1-x}{x}\right)^{0.8} \left(\frac{\rho_g}{\rho_l}\right)^{0.5} = \left(\frac{1-0.5}{0.5}\right)^{0.8} \left(\frac{60.9}{1145}\right)^{0.5} = 0.231$$

$\because Re_{lo} < 2300$, Sieder-Tate correlation: (otherwise, Petukhov correlation or Gnielinski's are used)

$$Nu_a = 1.86 \left(Re\, Pr \frac{d_i}{L}\right)^{1/3} \left(\frac{\mu_b}{\mu_w}\right)^{0/14}$$

Check validity (ignore the temperature difference between fluid and wall):

$$\left(\text{Re}\,\text{Pr}\frac{d_i}{L}\right)^{1/3} = \left(1066 \times 2.96 \times \frac{0.01727}{6}\right)^{1/3} = 2.1 \geq 2,$$

so this correlation is valid for the given data.

$$\text{Nu}_T = 1.86\left(\text{Re}\,\text{Pr}\frac{d_i}{L}\right)^{1/3} = 1.86 \times 2.1 = 3.91$$

$$h_{Lo} = \frac{\text{Nu}\cdot k}{d_i} = \frac{3.91 \times 0.082}{0.01727} = 18.6 \text{ W/m}^2.\text{K}$$

Froude number, Fr, is given by Eq. (8.65):

$$\text{Fr}_l = \frac{G^2}{\rho_l{}^2 g d_i} = \frac{22.96^2}{1145^2 \times 9.81 \times 0.01727} = 0.0024 > 0.04$$

then there is no stratified effects need to be taken care of.

F_{fl} is found to be 2.20 for R-22 in Table (8.1).

$$h_{TP,CBD} = \left(0.6683Co^{-0.2} + 1058.0Bo^{0.7}F_{fl}\right)(1-x)^{0.8}h_{lo}$$
$$= \left(0.6683 \times 0.231^{-0.2} + 1.058 \times 0.000718^{0.7} \times 2.20\right) \times (1-0.5)^{0.8} \times 18.6$$
$$= 1.05 \quad \text{W/m}^2K$$

$$h_{TP,NBD} = \left(1.136Co^{-0.9} + 667.2Bo^{0.7}F_{fl}\right)(1-x)^{0.8}h_{lo}$$
$$= \left(1.136 \times 0.231^{-0.9} + 667.2 \times 0.000718^{0.7} \times 2.2\right) \times (1-0.5)^{0.8} \times 18.6$$
$$= 103.02 \quad \text{W/m}^2K$$

$\because h_{TP,NBD} > h_{TP,CBD}$

$\therefore h_{TP} = h_{TP,NBD} = 103.02$ W/m^{2}K

b.

Assuming the fouling:

R_{fi} = 0.000088 m^{2}K/W

R_{fo} = 0.000176 m^{2}K/W

Thermal conductivity of brass is approximated as: k_w = 104 W/(m.K)

The overall heat transfer coefficient is :

Overall heat transfer coefficient:

$$U_o = \frac{1}{\dfrac{r_o}{r_i h_i} + \dfrac{r_o Rfi}{r_i} + r_o\dfrac{\ln(r_o/r_i)}{k} + R_{fo} + \dfrac{1}{h_o}}$$

$$U_o = \frac{1}{\dfrac{0.01905}{0.01727 \times 93.25} + \dfrac{0.01905 \times 0.000088}{0.01727} + \dfrac{0.01905}{2} \times \dfrac{\ln(0.01905/0.01727)}{104} + 0.000176 + \dfrac{1}{1288}}$$
$$= 77.59 \text{ W/(m}^2\text{K)}$$

$$\Delta T_m = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = \frac{(37-18)-(37-26)}{\ln((37-18)/(37-26))} = 14.64\ ^o\text{C}$$

$$Q = A_o U_o \Delta T_m$$

$$A_o = \frac{Q}{U_o \Delta T_m} = \frac{125000}{77.59 \times 14.64} = 110.04 \ \ m^2$$

$$L = \frac{A_o}{\pi d_o N_t N_p} = \frac{110.04}{\pi \times 0.01905 \times 137} = 13.42 \ \ m > 6 \text{ m}$$

c.

Since we assume in the earlier calculation that L = 6 m, we have to repeat the former procedure with the length finally obtained (13.42 m), until the assumed and the obtained match each other.

Problem 8.8

R-134a flows in a horizontal 8 mm diameter circular tube. The mass flux is 400 kg/m^2.s, entering quality is 0.0 and the exiting quality is 0.8. The tube length is 3m. Assuming a constant saturation temperature of 20^{o}C, plot the variation of h_{TP} as a function of x.

GIVEN:

G = 400 kg/m^2.s;

L = 3 m;

d = 0.008 m;

x_i = 0.0;

x_o = 0.8;

T^s = 20^{o}C.

SOLUTION:

Properties of R-134a at 20^{o}C (from Applendix) are:

$$\rho_l = 1225.9 \ \ \text{kg/m}^3$$
$$\rho_g = 27.9 \ \text{kg/m}^3$$
$$\mu = 2.25 \times 10^{-4} \ \ \text{kg/m.s}$$
$$c_p = 1029.5 \ \ \text{J/kg.K}$$
$$\text{Pr}_l = 3.761$$
$$k = 0.0843 \ \ \text{W/mK}$$

From Table 8.1, F_{fl} = 1.63 for R-134a.

$$\text{Fr}_{lo} = \frac{G^2}{\rho_l^{\,2} g d} = \frac{400^2}{1225.9^2 \times 9.81 \times 0.008} = 1.356507 \geq 0.04$$

$\therefore$ No correction is needed.

To calculate the variation of h_{Tp} with x, we assume the starting x to be 0.01. Calculation of h_{TP} corresponding to x = 0.01 is given below, then, a computer program is developed to calculate variation of h_{TP} with x in a same way when x = 0.01. The final results are given in both table and figure.

x = 0.01:

$$\text{Bo} = \frac{q''}{G \cdot i_{fg}} = \frac{G \cdot A_c \cdot (x_o - x_i) \cdot i_{fg}}{A_w \cdot G \cdot i_{fg}} = \frac{(x_o - x_i) d_i}{4L} = \frac{(0.8 - 0.0) \times 0.008}{4 \times 3} = 0.000533$$

$$Co = \left(\frac{1-x}{x}\right)^{0.8}\left(\frac{\rho_g}{\rho_l}\right)^{0.5} = \left(\frac{1-0.01}{0.01}\right)^{0.8}\left(\frac{27.9}{1225.9}\right)^{0.5} = 5.958$$

$$Re_{lo} = \frac{G(1-x)d}{\mu_l} = \frac{400\times(1-0.01)\times 0.008}{2.25\times 10^{-4}} = 14080$$

$$f = (1.58\ln Re_{lo} - 3.28)^{-2} = 0.0072$$

$\because Re_{lo} > 10^4$, Petukhov correlation is used: (otherwise, Genielinski correlation is used)

$$Nu_{lo} = \frac{Re_{lo}\,Pr_l\,(f/2)}{1.07+12.7(f/2)^{0.5}\left(Pr_l^{2/3}-1\right)} = \frac{14080\times 3.761\times 0.0072/2}{1.07+12.7\times(0.0072/2)^{0.5}\times\left(3.761^{2/3}-1\right)} = 91.295$$

$$h_{lo} = Nu\frac{k}{d} = \frac{91.295\times 0.0843}{0.008} = 962.02 \;\; W/m^2K$$

$$h_{TP,NBD} = \left(0.6683Co^{-0.2} + 1058.0Bo^{0.7}F_{fl}\right)(1-x)^{0.8}h_{lo}$$

$$= \left(0.6683\times 5.958^{-0.2} + 1.058\times 0.000533^{0.7}\times 1.63\right)\times(1-0.01)^{0.8}\times 962.02$$

$$= 8865.45 \quad W/m^2K$$

$$h_{TP,NBD} = \left(1.136Co^{-0.9} + 667.2Bo^{0.7}F_{fl}\right)(1-x)^{0.8}h_{lo}$$

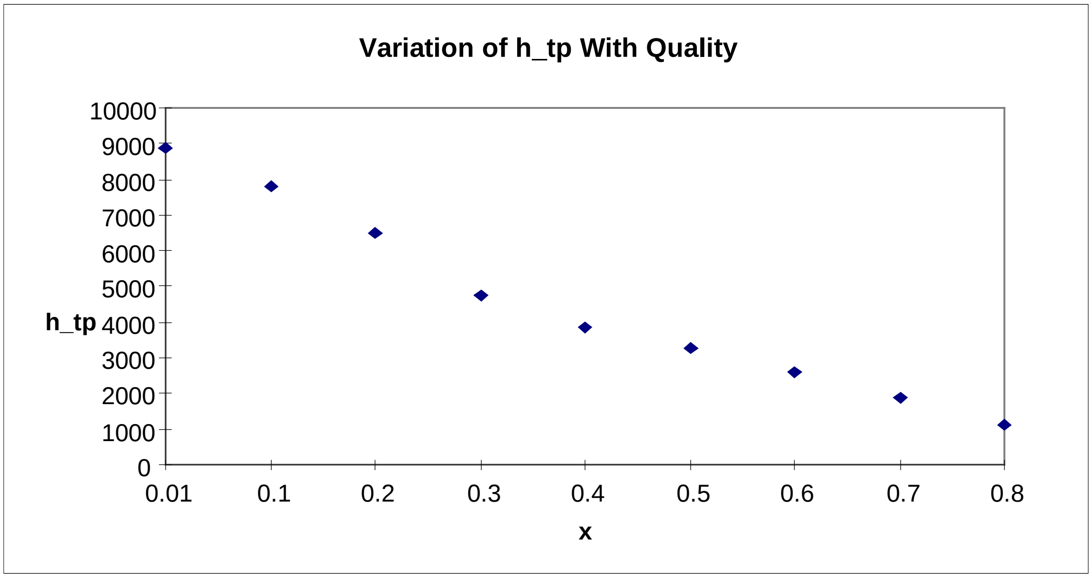
$$= \left(1.136\times 5.958^{-0.9} + 667.2\times 0.000533^{0.7}\times 1.63\right)\times(1-0.01)^{0.8}\times 962.02$$

$$= 5526.82 \quad W/m^2K$$

$\because h_{TP,NBD} > h_{TP,CBD}$

$\therefore h_{TP} = h_{TP,NBD} = 8865.45\ W/m^2K$

x	Bo	Co	Re_{lo}	f	Nu_{lo}	h_{lo}	$h_{TP,NBD}$	$h_{TP,CBD}$	h_{TP}
0.01	0.000533	5.958	14080	0.0072	91.295	962.025	8865.45	5526.82	8865.45
0.1	0.000533	0.875	12800	0.0074	84.586	891.329	7790.12	5607.64	7790.12
0.2	0.000533	0.457	11378	0.0076	76.994	811.328	6517.82	5334.82	6517.82
0.3	0.000533	0.297	9956	0.0079	62.281	656.282	4772.81	4415.46	4772.81
0.4	0.000533	0.209	8533	0.0082	54.097	570.042	3688.3	3870.72	3870.72
0.5	0.000533	0.151	7111	0.0087	45.614	480.661	2704.82	3256.47	3256.47
0.6	0.000533	0.109	5689	0.0093	36.739	387.139	1834.54	2587.05	2587.05
0.7	0.000533	0.077	4267	0.0101	27.304	287.72	1091.5	1870.67	1870.67
0.8	0.000533	0.05	2844	0.0116	16.96	178.713	495.12	1108.28	1108.28


Variation of h_tp With Quality
h_tp
10000
9000
8000
7000
6000
5000
4000
3000
2000
1000
0
0.01
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
x

Problem 9.1

Crude oil at a flow rate of 63.77 kg/s enters the exchanger at 102°C and leaves at 65°C. The heat will be transferred to 45 kg/s of tube water (city water) coming from a supply at 21°C. The exchanger data is given below: 3/4″ O.D., 18 BWG tubes on 1 inch square pitch 2-tube passes and 4-tube passes will be considered. Tube material is carbon steel. The heat exchanger has one shell. Two different shell diameter of I.D. 35 and 37 inches should be studied. Baffle spacing is 275 mm. Calculate the length of the heat exchanger for clean and fouled surfaces. Calculate:

a. Tube side velocity for 1-2 and 1-4 arrangements.
b. Overall heat transfer coefficients for clean and fouled surfaces.
c. Pressure-drops.
d. Pumping powers.

The allowable shell-side and tube-side pressure drops are 60 kPa and 45 kPa respectively. The following properties are given:

	Shell-side	Tube-side
Specific heat, J/kg.K	2177	4186.8
Dynamic viscosity, N.s/m2	0.00189	0.00072
Thermal conductivity, W/m.K	0.122	0.605
Density, kg/m3	786.4	995
Prandtl number	33.73	6.29
Maximum pressure loss, Pa	60,000	45,000

GIVEN:

- Mass flow rate of crude oil ($\dot{m}_h$) = 63.77 kg/s
- Inlet temperature of crude oil (T_{h1}) = 102 °C
- Outlet temperature of crude oil (T_{h2}) = 65 °C
- Mass flow rate of city water ($\dot{m}_c$) = 45 kg/s
- Inlet temperature of city water (T_{c1}) = 21 °C
- Heat exchanger data: 3/4″ OD, 18 BWG tubes on 1 inch square pitch
- One shell pass (with 35 inches ID or 37 inches ID) and two tube passes (or 4 tube passes)
- Baffle spacing (B) = 275 mm

FIND:

a. Tube side velocity for 1-2 and 1-4 arrangements (u).
b. Overall heat transfer coefficients for clean and fouled surfaces (U_c, U_f).
c. Pressure-drops (Δp).
d. Pumping powers (P_{pump}).

SOLUTION:

Design calculations for D_s = 35″ and N_p = 2 passes and 4 passes.

a. Tube side velocity:

i. From Table 9.3 for 3/4″ OD tubes in 1- in. square pitch and D_s = 35″:

Number of tubes N_t = 824 for 2-P, and N_t = 780 for 4-P.

Tube flow area:

N_p = 2 passes:

$$A_t = \frac{N_t}{N_p} \cdot \frac{\pi d_i^2}{4} = \frac{824}{2} \cdot \frac{\pi \times 0.01656^2}{4} = 0.089 \text{ m}^2$$

N_p = 4 passes:

$$A_t = \frac{N_t}{N_p} \cdot \frac{\pi d_i^2}{4} = \frac{780}{4} \cdot \frac{\pi \times 0.01656^2}{4} = 0.042 \text{ m}^2$$

Tube side velocity:

N_p = 2 passes:

$$u_m = \frac{\dot{m}_c}{\rho A_t} = \frac{45}{995 \times 0.089} = 0.51 \text{ m/s}$$

N_p = 4 passes:

$$u_m = \frac{\dot{m}_c}{\rho A_t} = \frac{45}{995 \times 0.042} = 1.08 \text{ m/s}$$

ii. From Table 9.3 for 3/4″ OD tubes in 1- in. square pitch and D_s = 37″:

Number of tubes N_t = 914 for 2-P, and N_t = 886 for 4-P.

Tube flow area:

N_p = 2 passes:

A_t = 0.098 m^2

N_p = 4 passes:

A_t = 0.048 m^2

Tube side velocity:

N_p = 2 passes:

u_m = 0.46 m/s

N_p = 4 passes:

u_m = 0.94 m/s

b. Overall heat transfer coefficient for clean and fouled surfaces:

i. For D_s = 35″:

Tube side heat transfer coefficient:

Reynolds number:

N_p = 2 passes:

$$Re = \frac{\rho u_m d_i}{\mu} = \frac{995 \times 0.51 \times 0.01656}{0.00072} = 11{,}671 \text{ (Turbulent flow)}$$

N_p = 4 passes:

$$Re = \frac{\rho u_m d_i}{\mu} = \frac{995 \times 1.08 \times 0.01656}{0.00072} = 24716 \text{ (Turbulent flow)}$$

Nusselt number:

Using the simplified Gnielinski correlation valid for 1.5 < Pr < 500 and 3×10^3 < Re < 10^6, we have:

N_p = 2 passes:

$$Nu = 0.012\left(Re^{0.87} - 280\right)Pr^{0.4} = 0.012 \times \left(11671^{0.87} - 280\right) \times 6.29^{0.4} = 79.5$$

N_p = 4 passes:

$$Nu = 0.012\left(Re^{0.87} - 280\right)Pr^{0.4} = 0.012 \times \left(24716^{0.87} - 280\right) \times 6.29^{0.4} = 159.2$$

Tube side heat transfer coefficient:

N_p = 2 passes:

$$h_i = \frac{Nu \cdot k}{d_i} = \frac{79.5 \times 0.605}{0.01656} = 2904.44\ W/m^2.K$$

N_p = 4 passes:

$$h_i = \frac{Nu \cdot k}{d_i} = \frac{159.2 \times 0.605}{0.01656} = 5816.18\ W/m^2.K$$

ii. For D_s = 37″:

Tube side heat transfer coefficient:

Reynolds number:

N_p = 2 passes:

Re = 10527 (Turbulent flow)

N_p = 4 passes:

Re = 21512 (Turbulent flow)

Nusselt number:

Using the simplified Gnielinski correlation valid for 1.5 < Pr < 500 and $3\times10^3 < Re < 10^6$, we have:

N_p = 2 passes:

Nu = 72.1

N_p = 4 passes:

Nu = 140.2

Tube side heat transfer coefficient:

N_p = 2 passes:

h_i = 2634.09 W/(m^2 K)

N_p = 4 passes:

h_i = 5122.04 W/(m^2 K)

Shell side heat transfer coefficient:

D_s = 35″:

$$A_s = \frac{D_s \cdot C \cdot B}{P_T} = \frac{D_s \cdot (P_T - d_o) \cdot B}{P_T} = \frac{0.889 \times (0.0254 - 0.01905) \times 0.275}{0.0254} = 0.061\,m^2$$

Ds = 37 ″:

$$A_s = \frac{D_s \cdot C \cdot B}{P_T} = \frac{D_s \cdot (P_T - d_o) \cdot B}{P_T} = \frac{0.9398 \times (0.0254 - 0.01905) \times 0.275}{0.0254} = 0.065\,m^2$$

Equivalent diameter:

$$D_e = \frac{4\cdot\left(P_T^{\,2} - \pi\, d_o^{\,2}/4\right)\cdot}{\pi d_o} = \frac{4\times\left[0.0254^2 - \left(\pi\times 0.01905^2\right)/4\right]}{\pi\times 0.01905} = 0.024\ \text{m}$$

Reynolds number:

$D_s = 35''$:

$$Re = \frac{\rho u_m D_e}{\mu} = \frac{\rho D_e}{\mu}\cdot\frac{\dot{m}_h}{\rho A_s} = \frac{D_e \dot{m}_h}{\mu A_s} = \frac{0.024\times 63.77}{0.00189\times 0.061} = 13275 \text{ (Turbulent flow)}$$

$D_s = 37''$:

$$Re = \frac{\rho u_m D_e}{\mu} = \frac{\rho D_e}{\mu}\cdot\frac{\dot{m}_h}{\rho A_s} = \frac{D_e \dot{m}_h}{\mu A_s} = \frac{0.024\times 63.77}{0.00189\times 0.065} = 12458 \text{ (Turbulent flow)}$$

Nusselt number:

For shell with baffles:

$D_s = 35''$:

$$Nu = 0.36 Re^{0.55} Pr^{1/3} = 0.36\times 13275^{0.55}\times 33.73^{1/3} = 215.2$$

$D_s = 37''$:

$$Nu = 0.36 Re^{0.55} Pr^{1/3} = 0.36\times 12458^{0.55}\times 33.73^{1/3} = 207.8$$

Shell side heat transfer coefficient:

$D_s = 35''$:

$$h_o = \frac{Nu\cdot k}{D_e} = \frac{215.2\times 0.122}{0.024} = 1093.9\ \text{W/m}^2\text{.K}$$

$D_s = 37''$:

$$h_o = \frac{Nu\cdot k}{D_e} = \frac{207.8\times 0.122}{0.024} = 1056.3\ \text{W/m}^2\text{.K}$$

Overall heat transfer coefficient:

For $D_s = 35''$ and $N_p = 2$ passes:

Cleaned conditions:

$$U_o = \left[\frac{d_o}{d_i h_i} + \frac{d_o \ln(d_o/d_i)}{2k} + \frac{1}{h_o}\right]^{-1}$$

$$= \left[\frac{0.01905}{0.01656\times 2904.44} + \frac{0.01905\ln\left(\frac{0.01905}{0.01656}\right)}{2\times 50} + \frac{1}{1093.9}\right]^{-1}$$

$$= 747.99\ \text{W/m}^2\text{.K}$$

Fouled conditions:

Fouling resistance for city water is R_{fi} = 0.000176 m^2.K/W, and for crude oil is R_{fo} = 0.000352. Thermal conductivity for carbon steel is k = 50 W/m.K

$$U_f = \left[\frac{d_o}{d_i h_i} + \frac{d_o}{d_i} R_{fi} + \frac{d_o \ln(d_o/d_i)}{2k} + R_{fo} + \frac{1}{h_o}\right]^{-1}$$

$$= \left[\frac{0.01905}{0.01656 \times 2904.44} + \frac{0.01905 \times 0.000176}{0.01656} + \frac{0.01905 \times \ln\left(\frac{0.01905}{0.01656}\right)}{2 \times 50} + 0.000352 + \frac{1}{1093.9}\right]^{-1}$$

$$= 528.78 \ \text{W/m}^2\text{.K}$$

c.

Shell-side pressure drop:

$$\Delta p_s = \frac{f G_s^{\,2} (N_b + 1) D_s}{2 \rho D_e \phi_s}$$

Number of baffles:

$$N_b = \frac{L}{B} - 1$$

Heat transferred:

$$\dot{Q} = \dot{m}_h c_{p,h} (T_{h1} - T_{h2}) = 63.77 \times 2177 \times (102 - 65) = 5136.6 \text{ kW}$$

Cold fluid outlet temperature:

$$\dot{Q} = \dot{m}_h c_{p,h} (T_{h1} - T_{h2}) = \dot{m}_c c_{p,c} (T_{c2} - T_{c1})$$

$$T_{c2} = T_{c1} + \frac{\dot{Q}}{\dot{m}_c c_{p,c}} = 21 + \frac{5136.6 \times 10^3}{45 \times 4186.8} = 48^o \text{C}$$

Log-mean temperature difference:

$$\Delta T_{lm,cf} = \frac{(102 - 48) - (65 - 21)}{\ln \frac{102 - 48}{65 - 21}} = 48.83^o \text{C}$$

Heat transfer area (F = 0.9 for preliminary design for any even number of tube-side passes):

$$A_c = \frac{Q}{U_o F \Delta T_{lm,cf}} = \frac{5136.6 \times 10^3}{747.99 \times 0.9 \times 48.83} = 156.26 \ \text{m}^2$$

$$A_f = \frac{Q}{U_f F \Delta T_{lm,cf}} = \frac{5136.6 \times 10^3}{528.78 \times 0.9 \times 48.83} = 221.04 \ \text{m}^2$$

Heat exchanger length:

$$L = \frac{A_f}{\pi d_o N_t} = \frac{221.04}{\pi \times 0.01905 \times 824} = 4.48 \ \text{m}$$

Number of baffles:

$$N_b = \frac{L}{B} - 1 = \frac{4.48}{0.275} - 1 = 15.29 \approx 16$$

Mass velocity:

$$G_s = \frac{\dot{m}_s}{A_s} = \frac{63.77}{0.061} = 1045.4 \ \text{kg/m}^2\text{.s}$$

Shell-side friction factor:

$$f = \exp(0.576 - 0.19\ln Re_s) = \exp[0.576 - 0.19\ln(13275)] = 0.293$$

Shell-side pressure drop:

$$\Delta p_s = \frac{fG_s^{\,2}(N_b + 1)D_s}{2\rho D_e \phi_s} = \frac{0.293 \times 1045.4^2 \times (16+1) \times 0.889}{2 \times 786.4 \times 0.024 \times 1} = 128203 \quad \text{Pa}$$

$\Delta p_s > \Delta p_a$ = 60,000 Pa. (Unacceptable)

Tube side pressure drop:

$$f = (1.58\ln Re - 3.28)^{-2} = [1.58\ln(11671) - 3.28]^{-2} = 0.0075$$

$$\Delta p_t = \left(4f\,\frac{LN_p}{d_i} + 4N_p\right)\frac{\rho u_m^{\,2}}{2}$$

$$= \left[4 \times 0.0075 \times \frac{4.48 \times 2}{0.01656} + 4 \times 2\right] \times \frac{995 \times 0.51^2}{2}$$

$$= 4154.6 \quad \text{Pa}$$

$\Delta p_t < \Delta p_a$ = 45,000 Pa (Acceptable)

d. Pumping powers:

Shell side:

$$P_t = \frac{\dot{m}_h \Delta p_s}{\eta_t \rho} = \frac{63.77 \times 128203}{0.8 \times 786.4} = 13.0 \quad \text{kW}$$

Tube side:

$$P_t = \frac{\dot{m}_c \Delta p_s}{\eta_t \rho} = \frac{45 \times 4154.6}{0.8 \times 995} = 234.9 \quad \text{W}$$

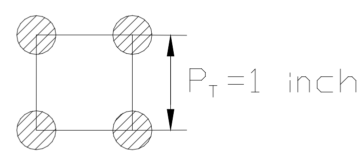
Problem 9.2

Water at a flow rate of 60 kg/s enters a baffled shell-and-tube heat exchanger at 35 °C and leaves at 25 °C. The heat will be transferred to 150 kg/s of raw water coming from a supply at 15 °C. You are requested to design the heat exchanger for this purpose. A single shell and single tube pass is preferable. The tube diameter is 3/4″ (19 mm O.D. with 16 mm I.D.) and tubes are laid out on 1 in. square pitch, Maximum length of the heat exchanger 8 m is required because of space limitations. The tube material is 0.5 Cr-alloy. Assume a total fouling resistance of 0.000176 m^2.K/W. Note that surface over design should not exceed 30%. Reasonable design assumption can be made along with the calculation if it is needed.
Calculate:

Shell diameter, number of tubes, fluid velocities, shell-side heat transfer coefficients, overall heat transfer coefficient, mean temperature difference, total area and pressure-drops. Is the final design acceptable? Discuss your findings.

GIVEN:

-Mass flow rate of distilled water ($\dot{m}_h$) = 60 kg/s
-Inlet temperature of distilled water (T_{h1}) = 35 °C
-Outlet temperature of distilled water (T_{h2}) = 25 °C
-Mass flow rate of raw water ($\dot{m}_c$) = 150 kg/s
-Inlet temperature of raw water (T_{c1}) = 15 °C
-Heat exchanger configuration:
- Single shell pass, Single tube pass.
- OD (d_o) = 19 mm, ID (d_i) =16 mm.
- PT = 1 in, Square pitch 90°.
- L = 8 m.



SOLUTION:

Properties of raw water in tube side at 15 °C:

$$c_{p,c} = 4186 \ \text{J/kg.K} \qquad \text{Pr} = 8.21$$

$$\mu = 11.54\times10^{-4} \ \text{N.s/m}^2 \qquad k = 0.591 \ \text{W/m}^2\text{.K}$$

$$\rho = 999 \ \text{kg/m}^3$$

Properties of distilled water in shell side at 30 °C:

$$c_{p,h} = 4179 \ \text{J/kg.K} \qquad \text{Pr} = 5.44$$

$$\mu = 7.98\times10^{-4} \ \text{N.s/m}^2 \qquad k = 0.614 \ \text{W/m}^2\text{.K}$$

$$\rho = 996 \ \text{kg/m}^3$$

Cold fluid outlet temperature:

$$\dot{Q} = \dot{m}_h c_{p,h}\left(T_{h1}-T_{h2}\right) = \dot{m}_c c_{p,c}\left(T_{c2}-T_{c1}\right) = 60\times4179\times(35-25) = 2507400 \quad \text{W}$$

$$T_{c2} = T_{c1} + \frac{\dot{m}_h c_{p,h}\left(T_{h1}-T_{h2}\right)}{\dot{m}_c c_{p,c}} = 15+\frac{60\times4179\times(35-25)}{150\times4186} = 19^o\,\text{C}$$

Log-mean temperature difference:

$$\Delta T_{lm,cf} = \frac{\left(T_{h1} - T_{c2}\right) - \left(T_{h2} - T_{c1}\right)}{\ln \dfrac{T_{h1} - T_{c2}}{T_{h2} - T_{c1}}}$$

$$= \frac{\left(35\text{-}19\right) - \left(25\text{-}15\right)}{\ln \dfrac{35\text{-}19}{25\text{-}15}} = 12.76\ ^{o}\text{C}$$

One tube pass: CTP = 0.93

The tube layout: CL = 1.0 for 90^{o}

Assume velocity: u_t = 2 m/s for tube side.

$$\dot{m}_t = \rho \frac{\pi d_i^{\,2}}{4} N_t u_t$$

$$N_t = \frac{4\dot{m}_t}{\rho \pi d_i^{\,2} u_t} = \frac{4 \times 150}{999 \times \pi \times 0.016^2 \times 2} = 373.39 \approx 374$$

Flow area:

$$A_t = \frac{\pi d_i^{\,2}}{4} N_t = \frac{\pi \times 0.016^2}{4} \times 374 = 0.075\ \text{m}^2$$

Shell diameter:

$$A_1 = (CL)P_T^{\,2} = 1 \times \left(2.54 \times 10^{-2}\right)^2 = 645.16 \times 10^{-6}\ \text{m}^2 \quad \text{(tube pitch } P_T = 2.54 \text{ cm)}$$

$$D_s^{\,2} = \frac{4N_t A_1}{(CTP)\pi} = \frac{4 \times 374 \times 645.16 \times 10^{-6}}{0.93 \times \pi} = 0.330344\ \text{m}^2$$

$$D_s = 574\ \text{mm} = 22.6\ \text{in.}$$

Round to the standards in Table 9.3, D_s = 23.25 in. and the number of tubes N_t = 341 for 1-P.

Design and results evaluation

Shell diameter	D_s = 23.25 in.
Number of tubes	N_t = 341
Tube length	L = 8 m
Tube diameter	d_o = 19 mm, d_i = 16 mm
Shell baffle spacing	B = 500 mm, baffle cut 25%

Tube side heat transfer:

$$u_t = \frac{4\dot{m}_{ti}}{\rho \pi d_i^{\,2} N_t} = \frac{4 \times 150}{999 \times \pi \times 0.016^2 \times 341} = 2.19\ \text{m/s}$$

$$Re_t = \frac{\rho u_t d_i}{\mu} = \frac{999 \times 2.19 \times 0.016}{11.54 \times 10^{-4}} = 30334$$

$$f = (1.58 \ln Re - 3.28)^{-2} = (1.58 \ln(30334) - 3.28)^{-2} = 0.0059$$

$$Nu = \frac{\left(f/2\right) Re\, Pr}{1.07 + 12.7\left(f/2\right)^{0.5}(Pr^{2/3} - 1)} = \frac{\left(0.0059/2\right) \times 30334 \times 8.20}{1.07 + 12.7 \times \left(0.0059/2\right)^{0.5}(8.20^{2/3} - 1)} = 230.22$$

$$h_t = \frac{k}{d_i} Nu = \frac{0.595}{0.016} \times 230.22 = 8504 \quad W/m^2.K$$

Pressure drop:

$$\Delta p_{ft} = 4f \frac{\rho u_t^2}{2} \frac{L}{d_i} = 4 \times 0.059 \times \frac{999 \times 2.19^2}{2} \times \frac{8}{0.016} = 113.08 \ \ kPa$$

Shell side heat transfer:

The equivalent diameter is:

$$D_e = \frac{4A_c}{P_w} = \frac{4\left(P_T^2 - \frac{\pi d_o^2}{4}\right)}{\pi d_o} = \frac{4\left[\left(2.54 \times 10^{-2}\right)^2 - \frac{\pi \times 0.019^2}{4}\right]}{\pi \times 0.019} = 0.0242 \ \ m$$

$$A_s = \frac{D_s CB}{P_T} = \frac{(23.25 \times 0.0254) \times (0.0254 - 0.019) \times 0.5}{0.0254} = 0.0744 \ \ m^2$$

$$G_s = \frac{\dot{m}}{A_s} = \frac{60}{0.0744} = 806.45 \ \ kg/(m^2.s)$$

$$Re = \frac{G_s D_e}{\mu} = \frac{806.45 \times 0.0242}{7.98 \times 10^{-4}} = 24456$$

Assuming constant properties:

$$Nu = 0.36\, Re^{0.55}\, Pr^{1/3} \left(\frac{\mu_b}{\mu_w}\right)^{0.14} = 0.36 \times 24456^{0.55} \times 5.44^{0.4} = 164.1$$

$$h_s = \frac{Nu_s k}{D_e} = \frac{164.1 \times 0.614}{0.0242} = 4164 \ \ W/m^2.K$$

Shell side pressure drop:

$$f = \exp\left(0.576 - 0.19 \ln Re_s\right) = \exp\left[0.576 - 0.19 \ln\left(24456\right)\right] = 0.259$$

$$N_b + 1 = \frac{L}{B} = \frac{8}{0.5} = 16 \ \text{(15 baffles)}$$

$$\Delta p_s = \frac{f G_s^2 \left(N_b + 1\right) D_s}{2 \rho D_e \phi_s} = \frac{0.259 \times 806.45^2 \times \left(16\right) \times (23.25 \times 0.0254)}{2 \times 996 \times 0.0242 \times 1} = 33.02 \ \ kPa$$

Overall heat transfer:

Assuming for the material: 0.18% C, 0.65% Cr, 0.23% Mo, 0.6% Si$\Rightarrow$ k = 42.3 W/m.K

For clean surface:

$$U_c = \left[\frac{1}{h_s} + \frac{1}{h_t}\frac{d_o}{d_i} + \frac{r_o \ln\left(d_o/d_i\right)}{k}\right]^{-1}$$

$$= \left[\frac{1}{4164} + \frac{1}{8504}\frac{0.019}{0.016} + \frac{9.5 \times 10^{-3} \ln\left(0.019/0.016\right)}{42.3}\right]^{-1}$$

$$= 2390 \quad W/m^2.K$$

For fouled surface:

$$U_f = \left[\frac{1}{U_c} + R_{ft}\right]^{-1} = \left[\frac{1}{2390} + 0.000176\right]^{-1} = 1682 \quad W/m^2.K$$

$$F_f = \frac{U_c}{U_f} = \frac{2390}{1682} = 1.42 \qquad \text{fouling factor.}$$

The overall surface design should not be more than 30 %, then clean schedule must be arranged accordingly.

Assume $\frac{U_c}{U_f} = 1.2$:

$$U_f = \frac{U_c}{1.20} = \frac{2390}{1.2} = 1992 \qquad W/(m^2.K)$$

$$R_f = \frac{1}{U_f} - \frac{1}{U_c} = 0.0000834 \qquad (m^2.K)/W$$

Heat transfer area:

$$A_f = \frac{Q}{U_c \Delta T_{lm,cf}} = \frac{2507400}{1992 \times 12.76} = 98.65 \quad m^2$$

$$L = \frac{98.65}{\pi \times 0.019 \times 341} = 4.84 < 8m$$

Problem 9.3

Distilled water at a flow rate of 80,000 kg/h enters an exchanger at 35 °C and leaves at 25 °C. The heat will be transferred to 140,000 kg/h of raw water coming from a supply at 20 °C. The baffles will be spaced 12 inches apart. Write a computer program to determine the effects of varying tube size and configuration as well as shell diameter on thermal characteristics, size and fluid flow characteristics of the shell and tube type of heat exchanger. 1-P, 2-P, and 4-P configurations will be considered. The exchanger data is given below:

3/4 ″ OD, 18 BWG tubes on 1 in. square pitch.

3/4 ″ OD, 18 BWG tubes on 1 in. triangular pitch

Study three different shell diameter of $15\frac{1}{4}$, $17\frac{1}{4}$, and $19\frac{1}{4}$ inch. Calculate the length of this heat exchanger for clean and fouled surfaces. A 120,000 N/m^2 pressure drop may be expended on both streams. Will this heat exchanger be suitable? For the pumping power assume a pump efficiency of 80%.

Square pitch   triangular pitch

SOLUTION:

Calculations for square pitch, shell diameter is 15.25 in. and 1 pass. All tubes are 18 BWG tubes 3/4 inch outer diameter.

Tube side:

Outer diameter, d_o = 3/4 in = 0.01905 m

Inner diameter, d_i = 0.652 in = 0.01656 m

Flow area, A_f = 0.3339 in^2 = 0.0002154 m^2

Wall thickness, t_w = 0.049 in = 0.001245 m

Mass flow rate, $\dot{m}_t = \dfrac{140000 \text{ kg/h}}{3600 \text{ s/h}}$ =38.89 kg/s

Shell side:

Pitch size, P_T = 1.0 in = 0.0254 m

Clearance, C = 0.25 in = 0.00635 m

Baffle spacing, B = 12 in = 0.3048 m

Shell diameter, D_s = 15.25 = 0.3874 m

Mass flow rate, $\dot{m}_s = \dfrac{80000 \text{ kg/h}}{3600 \text{ s/h}}$ =22.22 kg/s

For a single pass shell and tube heat exchanger with a shell diameter of 15.25 in., the number of tubes, N_t in a 1.0 in. square pitch with an outer tube diameter of 3/4 in. is 137 tubes.

Shell side:

$T_{in,\,h} = 35\ ^oC$

$T_{out,\,h} = 25\ ^oC$

$T_{bulk,\,h} = \dfrac{35+25}{2} = 30\ ^oC$

Water properties:

$c_{p,h} = 4.179\ \ kJ/kg.K$

$Pr = 5.44$

$\mu_b = 0.000798\ N.s/m^2$

$k = 0.614\ \ W/m^2.K$

$\rho = 996\ kg/m^3$

Tube side:

$T_{in,c} = 20\ ^oC$

$T_{out,\,c} = 25\ ^oC$

assume: $T_{bulk,\,c} = \dfrac{20+25}{2} = 22.5\ ^oC$

Water properties:

$c_{p,c} = 4.181\ \ kJ/kg.K$

$Pr = 6.58$

$\mu_b = 0.000947\ N.s/m^2$

$k = 0.602\ \ W/m^2.K$

$\rho = 998\ kg/m^3$

Wall temperature, $T_{wall} = \dfrac{T_{bulk,h} + T_{bulk,c}}{2} = \dfrac{30+22.5}{2} = 26.25^o C$, $\mu_w = 0.000867$ Pa.s.

Heat load:

$$Q = \dot{m}_c c_{pc}(T_{c2} - T_{c1}) = \dot{m}_h c_{ph}(T_{h1} - T_{h2})$$

$$T_{c2} = T_{c1} + \frac{Q}{\dot{m}_c c_{pc}} = 20 + \frac{80{,}000 \times 4.179 \times (35-25)}{140{,}000 \times 4.181} = 25.7\ ^oC$$

Log-mean temperature difference:

$$\Delta T_{lm,cf} = \frac{(T_{in,h} - T_{out,c}) - (T_{out,h} - T_{in,c})}{\ln\dfrac{T_{in,h} - T_{out,c}}{T_{out,h} - T_{in,c}}} = \frac{(35-25.7)-(25-20)}{\ln\dfrac{35-25.7}{25-20}} = 6.93\ ^oC$$

F, for a single pass shell and tube heat exchanger is one.

F = 1.

Shell side:

$$A_s = \frac{(D_{s'} \cdot C \cdot B)}{P_T} = \frac{(0.3874 \times 0.00635 \times 0.3048)}{0.0254} = 0.02952\ \ m^2$$

$$G_s = \frac{\dot{m}_s}{A_s} = \frac{22.22}{0.02952} = 752.71\ \ kg/m^2.s$$

$$D_e = \frac{4\left(P_T^{\,2} - \pi d_o^{\,2}/4\right)}{\pi d_o} = \frac{4 \times \left(0.0254^2 - \pi \times 0.01905^2/4\right)}{\pi \times 0.01905} = 0.0241\ \ m$$

$$Re_s = \frac{D_e \cdot G_s}{\mu} = \frac{0.0241 \times 752.41}{0.000798} = 22732$$

Therefore, the flow of the fluid on shell side is turbulent. Using the McAdam's Correlation,

$$Nu = 0.36 \cdot \left(\frac{D_e \cdot G_s}{\mu_b}\right)^{0.55} \cdot \left(\frac{c_{p,h} \cdot \mu_b}{k}\right)^{0.33} \cdot \left(\frac{\mu_b}{\mu_w}\right)^{0.14}$$

$$= 0.36 \times 22732^{0.55} \times 5.44^{0.33} \times \left(\frac{0.000798}{0.000867}\right)^{0.14}$$

$$= 154.94$$

The shell side heat transfer coefficient, h_o, is then found by the equation,

$$h_o = \frac{Nu \cdot k}{D_e} = \frac{154.94 \times 0.614}{0.0241} = 3947.4 \quad W/(m^2.K)$$

Tube side:

$$A_t = \frac{\pi d_i^2}{4} = \frac{\pi \times 0.01656^2}{4} = 0.0002154 \ \ m^2$$

$$A_{tp} = \frac{N_t A_t}{\#\ \text{of passes}} = \frac{137 \times 0.0002154}{1} = 0.02951 \ \ m^2$$

$$G_t = \frac{\dot{m}_t}{A_{tp}} = \frac{38.889}{0.02951} = 1317.94 \ \ kg/m^2.s$$

$$u_t = \frac{G_t}{\rho} = \frac{1317.94}{998} = 1.32 \ \ m/s$$

$$Re_t = \frac{u_t \cdot \rho \cdot d_i}{\mu} = \frac{1.32 \times 998 \times 0.01656}{0.000947} = 23036$$

Therefore, the flow of the fluid on tube side is turbulent. Since it is the variable property condition, the equation (3) in Table 3.6 is used:

$$Nu = \frac{(f/8) Re_b Pr_b}{1.07 + 12.7 (f/8)^{1/2} \left(Pr^{2/3} - 1\right)} \left(\frac{\mu_b}{\mu_w}\right)^{0.11}$$

where $f = (1.82 \log Re_b - 1.64)^{-2} = [1.28 \log(23{,}036) - 1.64]^{-2} = 0.0252$

$$Nu = 169.23$$

The tube side heat transfer coefficient, h_i, is then found by the equation,

$$h_i = \frac{Nu \cdot k}{d_i} = \frac{169.23 \times 0.602}{0.01656} = 6152.0 W/(m^2 \cdot K)$$

Now, finding the overall heat transfer coefficient, U_o, is done by the following equation,

$$U_o = \left[\frac{d_o}{d_i h_i} + \frac{d_o \ln(d_o/d_i)}{2k} + \frac{1}{h_o}\right]^{-1}$$

$$= \left[\frac{0.01905}{0.01656 \times 6152.0} + \frac{0.01905 \times \ln(0.01905/0.01656)}{2 \times 54} + \frac{1}{3947.4}\right]^{-1}$$

$$= 2150.9 \ \ W/m^2K$$

Assume the total fouling resistance to be $R_{ft} = 0.000176\ (m^2.K)/W$

$$U_f = \left[\frac{1}{U_c} + R_{ft}\right]^{-1} = \left[\frac{1}{2150.9} + 0.000176\right]^{-1} = 1560.25$$

To find the area and consequently, the length of the heat exchanger, the required heat transfer must first be determined. This heat transfer is determined by,

$$Q = \dot{m}_h c_{p,h}\left(T_{in,h} - T_{out,h}\right) = 22.22 \times 4179 \times (35 - 25) = 928574 \quad W$$

Now, the heat transfer is also defined as:

$$Q = U_o \cdot A \cdot \Delta T_m$$

Therefore, the area can be determined,

$$A_c = \frac{Q}{U_c \Delta T_m} = \frac{928574}{2150.9 \times 6.93} = 62.30 \ \ m^2$$

$$A_f = \frac{Q}{U_f \Delta T_m} = \frac{928574}{1560.25 \times 6.93} = 85.88 \ \ m^2$$

and the length,

$$L = \frac{A_f}{N_t \pi d_o} = \frac{85.88}{137 \times \pi \times 0.01905} \approx 11 \ \ m$$

Pressure drop in tube side:

$$\Delta p_t = 4f\frac{LN_p}{d_i} \cdot \frac{\rho u_m^2}{2} = 4 \times 0.0252 \times \frac{11 \times 1}{0.01656} \times \frac{998 \times 1.32^2}{2} = 58215.9 \ \ Pa < 120{,}000 \ Pa$$

Pressure drop in shell side:

Shell-side friction factor:

$$f = \exp\left(0.576 - 0.19 \ln Re_s\right) = \exp\left[0.576 - 0.19 \ln(22732)\right] = 0.2645$$

Shell-side pressure drop:

$$\phi_s = \left(\frac{\mu_b}{\mu_w}\right)^{0.14} = \left(\frac{0.000798}{0.000867}\right)^{0.14} = 0.988$$

$$N_B + 1 = \frac{L}{B} = \frac{11}{0.3048} = 36$$

$$\Delta p_s = \frac{f G_s^2 (N_b + 1) D_s}{2\rho D_e \phi_s} = \frac{0.2645 \times 752.71^2 \times 36 \times 0.3874}{2 \times 996 \times 0.0241 \times 0.988} = 44063 \quad Pa < 120{,}000 \ Pa$$

Following the same procedure to calculate the other size and configuration choice.

Problem 9.4

A heat exchanger is available to heat raw water by the use of condensed water at 67 ^oC which flows in the shell-side with a mass flow rate of 50,000 kg/h. Shell side dimensions are:

$$D_s = 19\frac{1}{4} \text{ in.}, \quad P_T = 1.25 \text{ in. (square)}, \quad \text{Baffle spacing } B = 0.3 \text{ m}$$

The raw water enters the tubes at 17 ^oC with mass flow rate of 30,000 kg/h. Tubes dimensions are:

$$d_o = 1 \text{ in.} = 0.0254 \text{ m (18 BWG tubes)}, \qquad d_i = 0.902 \text{ in.}$$

The length of the heat exchanger is 6 m with two passes. The permissible maximum pressure drop on the shell side is 1.5 psi. Water outlet temperature should not be less than 40 ^{o}.

Calculate:

a. outlet temperature;

b. heat load of the heat exchanger;

Is the heat exchanger appropriate to be used for this purpose?

SOLUTION:

Shell side (Condensed water):

$T_{h1} = 67\ ^oC$

$\dot{m}_h = 50{,}000$ kg/hr

$P_T = 1.25$ in $= 0.03175$ m (square)

$B = 0.3$ m

$D_s = 19.25 = 0.489$ m

Tube side (Raw water):

$T_{c1} = 17\ ^oC$

$\dot{m}_c = 30{,}000$ kg/hr

$d_o = 1'' = 0.0254$ m (18 BWG)

$d_i = 0.902'' = 0.02291$ m

$L = 6$ m

$N_p = 2$ passes

Water properties taken at inlet temperatures:

At $T_{h1} = 340$ K:

$c_{p,h} = 4.188$ kJ / kg.K

$Pr = 2.69$

$\mu_b = 0.000422$ N.s / m^2

$k = 0.625$ W / m^2.K

$\rho = 979$ kg / m^3

At $T_{c1} = 290$ K:

$c_{p,h} = 4.184$ kJ / kg.K

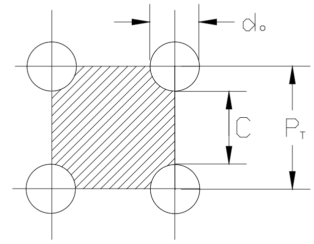
$Pr = 7.69$

$\mu_b = 0.00109$ N.s / m^2

$k = 0.594$ W / m^2.K

$\rho = 999$ kg / m^3

$C = P_T - d_o = 0.03175 - 0.0254 = 0.00635$ m



Shell side heat transfer coefficient:

Shell side flow area:

$$A_s = \frac{(D_s \cdot C \cdot B)}{P_T}$$

$$= \frac{(0.489 \times 0.00635 \times 0.3)}{0.03175} = 0.02934 \ \ m^2$$

Equivalent diameter:

$$D_e = \frac{4\left(P_T{}^2 - \pi d_o{}^2 / 4\right)}{\pi d_o} = \frac{4 \times \left(0.03175^2 - \pi \times 0.0254^2 / 4\right)}{\pi \times 0.0254} = 0.02513 \ \ m$$

Reynolds number:

$$u_m = \frac{\dot{m}_s}{\rho A_s} = \frac{\left(\frac{50{,}000}{3600}\right)}{979 \times 0.02934} = 0.48 \ \ m/s$$

$$Re = \frac{\rho u_m D_e}{\mu} = \frac{979 \times 0.48 \times 0.02513}{0.000422} = 27984 \ \ \text{(Turbulent)}$$

Using the McAdam's Correlation (assume $\mu_b = \mu_w$)

$$Nu = 0.36 \cdot Re^{0.55} \cdot Pr^{0.33} \cdot \left(\frac{\mu_b}{\mu_w}\right)^{0.14}$$

$$= 0.36 \times 27984^{0.55} \times 2.69^{0.33} \times 1^{0.14}$$

$$= 139.3$$

Shell side heat transfer coefficient:

$$h_o = \frac{Nu \cdot k}{D_e} = \frac{139.3 \times 0.625}{0.02513} = 3464 \ \ W/m^2.K$$

Tube side heat transfer coefficient:

For a two pass shell and tube heat exchanger with a shell diameter of 19.25 in., the number of tubes, N_t in a 1 in. square pitch with an outer tube diameter of 1 in. is 132 tubes.

Tube area:

$$A_t = \frac{N_t}{N_P} \cdot \frac{\pi d_i{}^2}{4} = \frac{132}{2} \times \frac{\pi \times 0.02291^2}{4} = 0.02721 \ \ m^2$$

$$u_m = \frac{\dot{m}_t}{\rho A_t} = \frac{\left(\frac{30{,}000}{3600}\right)}{999 \times 0.02721} = 0.3066 \ \ m/s$$

$$Re = \frac{u_m \cdot \rho \cdot d_i}{\mu} = \frac{0.3066 \times 999 \times 0.022911}{0.00109} = 6438$$

Using the Petukhov and Kirillov Correlation,

$$Nu = \frac{(f/2)(Re - 1000)Pr}{1.0 + 12.7(f/2)^{1/2}\left(Pr^{2/3} - 1\right)}$$

where $f = (1.58 \ln Re - 3.28)^{-2} = [1.58\ln(6438) - 3.28]^{-2} = 0.00894$

$$Nu = \frac{(0.00894/2) \times (6437 - 1000) \times 7.69}{1.07 + 12.7(0.00894/2)^{1/2}\left(7.69^{2/3} - 1\right)} = 52.97$$

$$h_i = \frac{Nu \cdot k}{d_i} = \frac{63.97 \times 0.594}{0.022911} = 1373 \ \ W/m^2.K$$

a. Fluid outlet temperature:

$$C_h = \dot{m}_h c_{p,h} = \left(\frac{50{,}000}{3600}\right) \times 4188 = 58167 \ \ W/K$$

$$C_c = \dot{m}_c c_{p,c} = \left(\frac{30{,}000}{3600}\right) \times 4184 = 34867 \ \ W/K$$

$$C_{min} = C_c = 34867 \ \ W/K$$

$$C^* = C_{min}/C_{max} = \frac{34867}{58167} = 0.6$$

Assume copper tube, k = 386 W/m.K, $R_{fi} = 3.52\times10^{-5}$ m^2.K/W, $R_{fo} = 8.8\times10^{-5}$ m^2.K/W.

$$U_o = \left[\frac{d_o}{d_i h_i} + \frac{d_o R_{fi}}{d_i} + \frac{d_o \ln(d_o/d_i)}{2k} + R_{fo} + \frac{1}{h_o}\right]^{-1}$$

$$= \left[\frac{0.0254}{0.02291 \times 1373} + \frac{0.0254 \times 0.0000352}{0.02291} + \frac{0.0254 \times \ln(0.0254/0.02291)}{2 \times 386} + 0.000088 + \frac{1}{3464}\right]^{-1}$$

$$= 815.26 \ \ W/m^2 K$$

Heat exchanger surface area:

$$A_o = N_t \pi d_o L = 132 \times \pi \times 0.0254 \times 6 = 63.2 \ \ m^2$$

$$NTU = \frac{A_o U_o}{C_{min}} = \frac{63.2 \times 815.26}{34867} = 1.5$$

From Fig. (2.15),

$$\varepsilon = \frac{(T_{c2} - T_{c1})}{(T_{h1} - T_{c1})} = 0.6$$

Then,

$$T_{c2} = \varepsilon(T_{h1} - T_{c1}) + T_{c1} = 0.6 \times (67 - 17) + 17 = 47^o C$$

From

$$Q = \dot{m}_c c_{p,c}(T_{c2} - T_{c1}) = \dot{m}_h c_{p,h}(T_{h1} - T_{h2})$$

$$T_{h2} = T_{h1} - \frac{\dot{m}_c c_{p,c}(T_{c2} - T_{c1})}{\dot{m}_h c_{p,h}} = 67 - \frac{34867 \times (47 - 17)}{58167} = 49 \ ^o C$$

b. heat load of the heat exchanger

$$\text{P} = \frac{T_{c2} - T_{c1}}{T_{h1} - T_{c1}} = \frac{47 - 17}{67 - 18} = 0.612$$

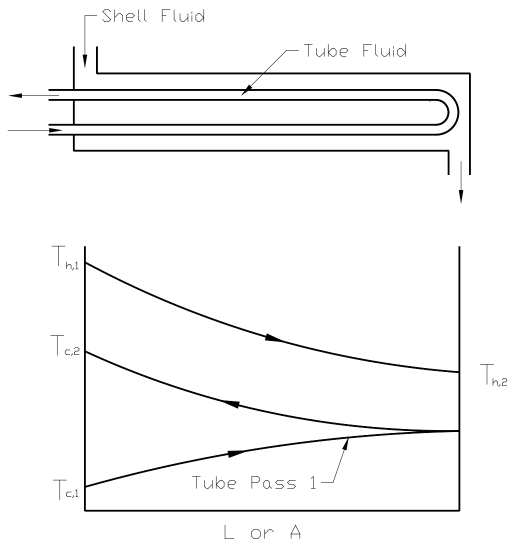
$$\text{R} = \frac{T_{h1} - T_{h2}}{T_{c2} - T_{c1}} = \frac{67 - 49}{47 - 17} = 0.6$$

From Figure 2.7; F = 0.83

$$\Delta \text{T}_{lm} = \frac{(T_{h2} - T_{c1}) - (T_{h1} - T_{c2})}{\ln\left[(T_{h2} - T_{c1})/(T_{h1} - T_{c2})\right]} = \frac{(49 - 17) - (67 - 47)}{\ln\left[(49 - 17)/(67 - 47)\right]} = 25.532$$

From Eq (2.36)

$$\text{Q} = F\Delta T_m \text{A}_\text{o}\text{U}_\text{o} = 0.83 \times 25.532 \times 63.2 \times 815.26 = 1092 \times 10^3 \; W$$



Shell side pressure drop:

$$\Delta \text{p}_\text{s} = \frac{\text{f} \cdot \text{D}_\text{s}(\text{N}_\text{B} + 1) \cdot \text{G}_\text{s}^{\;2}}{2\rho \text{D}_\text{e} \phi_s}$$

$$\begin{aligned} \text{f} &= \exp\left[0.576 - 0.19\ln(\text{Re})\right] \\ &= \exp\left[0.576 - 0.19\ln(27984)\right] \\ &= 0.254 \end{aligned}$$

$$\text{N}_\text{B} + 1 = \frac{\text{L}}{\text{B}} = \frac{6}{0.3} = 20$$, therefore 19 baffles

$$\text{G}_\text{s} = \frac{\dot{\text{m}}_\text{s}}{\text{A}_\text{s}} = \frac{(50000/3600)}{0.02934} = 473.28$$

$\phi_\text{s} = 1$ (Since assume constant properties.)

$$\Delta p_s = \frac{f \cdot D_s (N_B + 1) \cdot G_s^{\,2}}{2\rho D_e \phi_s}$$

$$= \frac{0.254 \times 0.489 \times (19 + 1) \times 473.38^2}{2 \times 979 \times 0.02513}$$

$$= 11313 \quad N/m^2$$

$$= 1.64 \quad Psi$$

Heat exchanger is inappropriate since $\Delta p_s > 1.5$ Psi. To make improvement, baffle space can be increased, for example, to 0.5 m, to decrease the pressure drop in shell side.

Problem 9.5

Distilled water at a flow rate of 20 kg/s enters an exchanger at 35°C and leaves at 25 °C. The heat will be transferred to 40 kg/s of raw water coming from supply at 20 °C. Available for this service is a 17 1/4-in. I.D. exchanger having 166, 3/4-in. O.D. Tubes (18 BWG) and laid out on 1-in. square pitch. The bundle is arranged for two passes and baffles are spaced 12 in. apart. Calculate the length of this heat exchanger if all the surfaces are clean. Will this heat exchanger be suitable? Assume fouling factors. A 12×10^4 N/m^2 pressure drop may be expended on both streams.

GIVEN:

- -Distilled water flowing in shell side, and raw water in tube side, of a shell and tube H.E.
- -Hot water inlet temperature ($T_{h,1}$) = 35 °C
- -Hot water outlet temperature ($T_{h,2}$) = 25 °C
- -Mass flow rate of hot water ($\dot{m}_h$) = 20 kg/s
- -Raw water inlet temperature ($T_{c,1}$) = 20 °C
- -Mass flow rate of raw water ($\dot{m}_c$) = 40 kg/s
- -I.D. of shell (D_s) = 17.25 in.= 0.4382 m
- -Number of tubes (N_t) = 166 tubes.
- -Tube arrangement: 1 in. square pitch
- -Tube diameter: d_o = 3/4 in. = 0.01905 m
  d_i = 0.652 in. = 0.01656 m
  (for 3/4 O.D., 18 BWG tubes given)
- -Baffle space (L_B) = 12 in. = 0.3048 m
- -Number of tube passes (N_p) = 2

SOLUTION:

a.

Properties of distilled water in shell side at 30 °C:

$$c_{p,h} = 4179 \ \text{J/kg.K} \qquad \text{Pr} = 5.44$$

$$\mu = 7.98\times10^{-4} \ \text{N.s/m}^2 \qquad k = 0.614 \ \text{W/m}^2\text{.K}$$

$$\rho = 996 \ \text{kg/m}^3$$

Properties of raw water in tube side at 22.5 °C (Assume bulk temperature = (20+25)/2):

$$c_{p,c} = 4.181 \ \text{kJ/kg.K} \qquad \text{Pr} = 6.58$$

$$\mu_b = 0.000947 \ \text{N.s/m}^2 \qquad k = 0.602 \ \text{W/m}^2\text{.K}$$

$$\rho = 998 \ \text{kg/m}^3$$

<u>Tube side heat transfer coefficient:</u>

$$u_m = \frac{\dot{m}_c}{\rho\left(\pi d_i^{\,2}/4\right)\left(N_t/N_p\right)} = \frac{40}{998\times\left(\pi\times0.01656^2/4\right)\left(166/2\right)} = 2.24 \ \text{m/s}$$

$$\text{Re} = \frac{\rho u_m d_i}{\mu} = \frac{998\times2.24\times0.01656}{9.47\times10^{-4}} = 39092 > 2300$$

Using Prandtle's Correlation:

$$f = (1.58\ln\text{Re} - 3.28)^{-2} = \left[1.58\ln(39092) - 3.28\right]^{-2} = 0.0055$$

$$\mathrm{Nu} = \frac{(f/2)\mathrm{RePr}}{1.0 + 8.7(f/2)^{1/2}(\mathrm{Pr}-1)} = \frac{(0.0055/2)\times 39092 \times 6.58}{1 + 8.7\times(0.0055/2)^{1/2}\times(6.58-1)} = 199.5$$

$$h_i = \frac{\mathrm{Nu}\cdot k}{d_i} = \frac{115.05\times 0.602}{0.01656} = 7252 \;\; \mathrm{W/m^2K}$$

Shell side heat transfer coefficient:

$$C = P_T - d_o = 0.0254 - 0.01905 = 0.00635\,\mathrm{m}$$

$$A_s = \frac{(D_s \cdot C \cdot B)}{P_T} = \frac{(0.4382\times 0.00635\times 0.3048)}{0.0254} = 0.03339 \;\; \mathrm{m^2}$$

$$G_s = \frac{\dot{m}_s}{A_s} = \frac{20}{0.03339} = 599 \;\; \mathrm{kg/m^2.s}$$

$$D_e = \frac{4\left(P_T^{\,2} - \pi d_o^{\,2}/4\right)}{\pi d_o} = \frac{4\times\left(0.0254^2 - \pi\times 0.01905^2/4\right)}{\pi\times 0.01905} = 0.0241 \;\; \mathrm{m}$$

$$\mathrm{Re}_s = \frac{D_e \cdot G_s}{\mu} = \frac{0.0241\times 599}{0.000798} = 18089$$

Therefore, the flow of the fluid on shell side is turbulent. Using the McAdam’s Correlation(neglect the temperature difference between the tube wall and bulk fluid),

$$\mathrm{Nu} = 0.36\cdot\left(\frac{D_e\cdot G_s}{\mu_b}\right)^{0.55}\cdot\left(\frac{c_{p,h}\cdot\mu_b}{k}\right)^{0.33}$$

$$= 0.36\times 18089^{0.55}\times 5.44^{0.33}$$

$$= 138.24$$

The shell side heat transfer coefficient, h_o, is then found by the equation,

$$h_o = \frac{\mathrm{Nu}\cdot k}{D_e} = \frac{138.24\times 0.614}{0.0241} = 3522 \quad \mathrm{W/(m^2.K)}$$

Overall heat transfer coefficient (Clean surface):

$$U_o = \left[\frac{d_o}{d_i h_i} + \frac{d_o \ln(d_o/d_i)}{2k_w} + \frac{1}{h_o}\right]^{-1}$$

$$= \left[\frac{0.01905}{0.01656\times 7252} + \frac{0.01905\ln(0.01905/0.01656)}{2\times 52} + \frac{1}{3522}\right]^{-1}$$

$$= 2136 \;\; \mathrm{W/m^2K}$$

assume the tube wall thermal conductivity to be 52 W/(m.K).

LMTD:

Number of tubes:

Heat balance:

$$\dot{Q} = (\dot{m}c_p)_c(T_{c2} - T_{c1}) = (\dot{m}c_p)_h(T_{h1} - T_{h2})$$

$$\dot{Q} = (\dot{m}c_p)_h(T_{h1} - T_{h2}) = 20\times 4179\times(35-25) = 835800 \;\; \mathrm{W}$$

$$T_{c2} = T_{c1} + \frac{Q}{(\dot{m}c_p)_c} = 20 + \frac{835800}{40\times 4181} = 25 \;\; ^{o}\mathrm{C}$$

$$R = \frac{T_{h1} - T_{h2}}{T_{c2} - T_{c1}} = \frac{35-25}{25-20} = 2$$

$$P = \frac{T_{c2} - T_{c1}}{T_{h1} - T_{c1}} = \frac{25-20}{35-20} = 0.33$$

Then, F = 0.77

$$\Delta T_{lm,cf} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{10-5}{\ln(10/5)} = 7.21 \ \ {}^{o}C$$

$$A_o = \frac{\dot{Q}}{U_o \cdot \Delta T_{lm,cf} \cdot F} = \frac{835800}{2136 \times 7.21 \times 0.77} = 70.48 \ \ m^2$$

$$A_o = \pi d_o N_T L$$

$$\Rightarrow L = \frac{A_o}{\pi d_o N_T} = \frac{70.48}{\pi \times 0.01905 \times 166} = 7.09 \approx 8 \ \ m$$

b.

Assume the fouling factor to be R_{ft} = 0.000079 (m^2.K)/W, then the overall heat transfer coefficient U_f after fouling is:

$$U_f = \left[\frac{1}{U_c} + R_{ft}\right]^{-1} = \left[\frac{1}{2136} + 0.000079\right]^{-1} = 1828 \quad W/(m^2.K)$$

$$A_o = \frac{\dot{Q}}{U_o \cdot \Delta T_{lm,cf} \cdot F} = \frac{835800}{1828 \times 7.21 \times 0.77} = 82.36 \ \ m^2$$

$$A_o = \pi d_o N_T L$$

$$\Rightarrow L = \frac{A_o}{\pi d_o N_T} = \frac{82.36}{\pi \times 0.01905 \times 166} = 8.28 \approx 9 \ \ m$$

<u>Pressure drop in tube side:</u>

$$\Delta p_t = 4f \frac{LN_p}{d_i} \cdot \frac{\rho u_m^{\ 2}}{2} = 4 \times 0.0055 \times \frac{9 \times 2}{0.01656} \times \frac{998 \times 2.24^2}{2} = 59873 \ \ \text{Pa} < 120{,}000 \ \text{Pa}$$

<u>Pressure drop in shell side:</u>

Shell-side friction factor:

$$f = \exp(0.576 - 0.19 \ln Re_s) = \exp[0.576 - 0.19 \ln(18089)] = 0.276$$

Shell-side pressure drop:

$$\phi_s = 1$$

$$N_B + 1 = \frac{L}{B} = \frac{9}{0.3048} = 29.52 \approx 30$$

$$\Delta p_s = \frac{f G_s^{\ 2} (N_b + 1) D_s}{2 \rho D_e \phi_s} = \frac{0.276 \times 599^2 \times 30 \times 0.4382}{2 \times 996 \times 0.0241 \times 1} = 27118 \quad \text{Pa} < 120{,}000 \ \text{Pa}$$

The heat exchanger is suitable for the designated heat load and pressure drop requirement.

Problem 9.6

A 1 to 2 baffled shell-and-tube type heat exchanger is used as an engine oil cooler. Cooling water flows through tubes at 25 °C at a rate of 8.16 kg/s and exits at 35 °C. The inlet and outlet temperatures of the engine oil are 65 and 55 °C, respectively. The heat exchanger has 15.25-in. I.D. shell, and 18 BWG and 0.75-in. O.D. tubes. A total of 160 tubes are laid out on a 15/16-in. triangular pitch. By assuming R_{fo} = 1.76×10^{-4} (m^2.K)/W, A_oR_w = 1.084×10^{-5} (m^2.K)/W , h_o = 686 W/(m^2.K), A_o/A_i = 1.1476, and R_{fi} = 8.8×10^{-5} (m^2.K)/W, find the

a. the heat transfer coefficient inside the tubes.

b. the total surface area of the heat exchanger.

<u>GIVEN:</u>

- A shell-and-tube oil cooler, cooling water flowing in tube.
- Engine oil inlet temperature ($T_{h,1}$) = 65 °C
- Engine oil outlet temperature ($T_{h,2}$) = 55 °C
- Cooling water inlet temperature ($T_{c,1}$) = 25 °C
- Cooling water outlet temperature ($T_{c,2}$) =35 °C
- Mass flow rate of cooling water ($\dot{m}_c$) = 8.16 kg/s
- Shell side heat transfer coefficient (h_o) = 686 W/(m^2.K)
- Fouling resistance: R_{fo} = 0.000176 (m^2.K/W)
  R_{fi} = 0.000088 (m^2.K/W)
- Ratio of outer surface and inner surface area (A_o/A_i) = 1.1476
- Thermal resistance of tube material (A_oR_w) = 1.084×10^{-5} W/(m.K)
- Tube diameter: d_o = 3/4 in. = 0.01905 m
  d_i = 0.652 in. = 0.01656 m
  (for 3/4 O.D., 18 BWG tubes given)
- Number of tubes (N_t) = 160 tubes
- Number of tube passes (N_p) = 2
- Tube arrangement: 15/16-in. triangular pitch.

<u>SOLUTION:</u>

a.

Properties of cooling water in tube side at 30 °C:

$$c_{p,h} = 4179 \ \text{J/kg.K} \qquad \text{Pr} = 5.44$$

$$\mu = 7.98\times10^{-4} \ \text{N.s/m}^2 \qquad k = 0.614 \ \text{W/m}^2\text{.K}$$

$$\rho = 996 \ \text{kg/m}^3$$

<u>Tube side heat transfer coefficient:</u>

$$u_m = \frac{\dot{m}_c}{\rho\left(\pi d_i^2/4\right)\left(N_t/N_p\right)} = \frac{8.16}{996\times\left(\pi\times0.01656^2/4\right)\left(160/2\right)} = 0.475 \ \text{m/s}$$

$$\text{Re} = \frac{\rho u_m d_i}{\mu} = \frac{996\times0.475\times0.01656}{7.98\times10^{-4}} = 9817$$

Using Genielinski's Correlation:

$$f = (1.58\ln\text{Re} - 3.28)^{-2} = \left[1.58\ln(9817) - 3.28\right]^{-2} = 0.0079$$

$$\text{Nu} = \frac{(f/2)(\text{Re}-1000)\text{Pr}}{1.0+12.7(f/2)^{1/2}\left(\text{Pr}^{2/3}-1\right)} = \frac{(0.0079/2)\times(9817-1000)\times5.44}{1+12.7\times(0.0079/2)^{1/2}\times(5.44-1)} = 70.94$$

$$h_i = \frac{Nu \cdot k}{d_i} = \frac{70.94 \times 0.614}{0.01656} = 2630 \ \ W/m^2K$$

b.

<u>Overall heat transfer coefficient:</u>

$$U_o = \left[\frac{A_o}{A_i h_i} + \frac{A_o R_{fi}}{A_i} + A_o R_w + R_{fo} + \frac{1}{h_o}\right]^{-1}$$

$$= \left[\frac{1.1476}{2630} + 1.1476 \times 0.000088 + 1.084 \times 10^{-5} + 0.000176 + \frac{1}{686}\right]^{-1}$$

$$= 458 \ \ W/m^2K$$

Heat balance:

$$R = \frac{T_{h1} - T_{h2}}{T_{c2} - T_{c1}} = \frac{65-55}{35-25} = 1.$$

$$P = \frac{T_{c2} - T_{c1}}{T_{h1} - T_{c1}} = \frac{35-25}{65-25} = 0.25$$

Then, F = 0.97

$$\Delta T_{lm,cf} = \Delta T_1 = \Delta T_2 = 30 \ \ ^oC$$

Thus, the total heat transfer area A_o is:

$$A_o = \frac{\dot{Q}}{U_o \cdot \Delta T_{lm,cf} \cdot F} = \frac{\dot{m}_c c_{p,c} \Delta T_c}{U_o \cdot \Delta T_{lm,cf} \cdot F} = \frac{8.16 \times 4179 \times 10}{458 \times 30 \times 0.97} = 25.59 \ \ m^2$$

Problem 9.7

A water-to-water system is used to test the effects of changing tube length, baffle spacing, tube pitch, pitch layout, and tube diameter. Cold water at 25 °C and 100,000 kg/h is heated by hot water at 100 °C, and also at 100,000 kg/h. The exchanger has a 31-in. I.D. shell. Perform calculations on this 1 to 2 shell-and-tube heat exchanger for the following conditions, as outlined in the examples; and put together an overall comparison chart.
a. 3/4 in. OD tubes, 12 BWG laid out on a 1 in. triangular pitch; 3 baffles per meter of tube length. Analyze the exchanger for tube lengths of 2 m, 3 m, 4 m, and 5 m.
b. 3/4 in. OD tubes, 12 BWG laid out on a 1 in. triangular pitch; 2 m long. Analyze for baffle placement of 1 baffle per meter of tube length, 2 baffles per meter of tube length, 3 baffles per meter of tube length, 4 baffles per meter of tube length, and 5 baffles per meter of tube length.
c. 3/4 in. OD tubes, 12 BWG; 2 m long; 4 baffles per meter of tube length. Analyze the exchanger for tube layouts of 15/16 in. triangular pitch, 1 in. triangular pitch, and 1 in. square pitch.
d. 1 m OD tubes, 12 BWG; 4 m long; 9 baffles. Analyze the exchanger for tube layouts of 1.25 in. triangular pitch and 1.25 in. square pitch.
e. 3/4 in. OD tubes, 12 BWG laid out on a 1 in. triangular pitch; 2 m long; 4 baffles per meter of tube length. Analyze the exchanger for 2, 4, 6, and 8 tube passes, and compare to the case of true counterflow (i.e., 1 tube pass).

GIVEN:

- Hot water inlet temperature ($T_{h,1}$) = 100 °C
- Cold water inlet temperature ($T_{c,1}$) = 25 °C
- Mass flow rate of hot water ($\dot{m}_h$) = 100,000 kg/hr = 27.778 kg/s
- Mass flow rate of cold water ($\dot{m}_c$) = 100,000 kg/hr = 27.778 kg/s
- Shell I.D. (D_i) = 31 in. = 0.7874 m
- Tube: 3/4-in O.D. (12 BWG)
  - d_o = 0.75 in. = 0.01905 m
  - d_i = 0.532 in. = 0.01351 m
- Number of tube pass (N_p) = 2

SOLUTION:

Properties of cold water in shell side at 25 °C:

$$c_{p,c} = 4180 \text{ J/kg.K} \qquad Pr = 6.16$$
$$\mu = 8.92\times10^{-4} \text{ N.s/m}^2 \qquad k = 0.606 \text{ W/m}^2\text{.K}$$
$$\rho = 997 \text{ kg/m}^3$$

Properties of hot water in tube side at 100 °C:

$$c_{p,h} = 4222 \text{ J/kg.K} \qquad Pr = 1.729$$
$$\mu = 2.79\times10^{-4} \text{ N.s/m}^2 \qquad k = 0.681 \text{ W/m}^2\text{.K}$$
$$\rho = 958 \text{ kg/m}^3$$

Approximate the tube wall temperature by $\frac{100+25}{2} = 62.5\,^{\circ}\text{C}$, then $\mu_w = 4.51\times10^{-4}$ Pa.s.

a. Comparing effects of different tube length:

From Table 9.3, for shell I.D. = 31 in., 2-P heat exchanger with 3/4-in. O.D. tube in 1-in. triangular pitch, the number of the tubes N_t = 728 tubes.

Tube side heat transfer coefficient:

$$u_m = \frac{\dot{m}_h}{\rho\left(\pi d_i^{\,2}/4\right)\left(N_t/N_p\right)} = \frac{27.788}{958\times\left(\pi\times 0.01351^2/4\right)\left(728/2\right)} = 0.556 \ \ m/s$$

$$Re = \frac{\rho u_m d_i}{\mu} = \frac{958\times 0.556\times 0.01351}{2.79\times 10^{-4}} = 25792 > 2300$$

So, it is turbulent flow.

Using Prandtle's Correlation:

$$f = (1.58\ln Re - 3.28)^{-2} = \left[1.58\ln\left(25792\right) - 3.28\right]^{-2} = 0.00613$$

$$Nu = \frac{\left(f/2\right)Re Pr}{1.0 + 8.7\left(f/2\right)^{1/2}\left(Pr - 1\right)} = \frac{\left(0.00613/2\right)\times 25792\times 1.729}{1 + 8.7\times\left(0.00613/2\right)^{1/2}\times(1.729-1)} = 184.77$$

$$h_i = \frac{Nu\cdot k}{d_i} = \frac{184.77\times 0.681}{0.01351} = 9314 \ \ W/m^2K$$

Shell side heat transfer:

The equivalent diameter is:

$$D_e = \frac{4A_c}{P_w} = \frac{4\left(\frac{P_T^{\,2}\sqrt{3}}{4} - \frac{\pi d_o^{\,2}}{8}\right)}{\pi d_o/2} = \frac{4\left[\frac{\left(2.54\times 10^{-2}\right)^2\sqrt{3}}{4} - \frac{\pi\times 0.01905^2}{8}\right]}{\pi\times 0.01905/2} = 0.01829 \ \ m$$

$$B = \frac{1}{(N_B + 1)} = \frac{1}{3+1} = 0.25 \ \ m$$

$$A_s = \frac{D_s C B}{P_T} = \frac{(31\times 0.0254)\times(0.0254 - 0.019)\times 0.25}{0.0254} = 0.0492 \ \ m^2$$

$$G_s = \frac{\dot{m}_c}{A_s} = \frac{27.778}{0.0492} = 564.45 \ \ kg/(m^2.s)$$

$$Re = \frac{G_s D_e}{\mu} = \frac{564.45\times 0.01892}{8.92\times 10^{-4}} = 11972$$

Assuming constant properties:

$$Nu = 0.36\,Re^{0.55}\,Pr^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.14} = 0.36\times 11972^{0.55}\times 6.16^{0.4}\left(\frac{8.92}{4.51}\right)^{0.14} = 143.4$$

$$h_s = \frac{Nu_s k}{D_e} = \frac{143.4\times 0.606}{0.01892} = 4594 \ \ W/m^2.K$$

Overall heat transfer coefficient: (assume the wall thermal conductivity k_w = 52 W/mK)

$$U_o = \left[\frac{d_o}{d_i h_i} + \frac{d_o \ln\left(d_o/d_i\right)}{2k_w} + \frac{1}{h_o}\right]^{-1}$$

$$= \left[\frac{0.01905}{0.01351\times 9314} + \frac{0.01905\ln\left(0.01905/0.01351\right)}{2\times 52} + \frac{1}{4594}\right]^{-1}$$

$$= 2315 \ \ W/m^2K$$

ε-NTU method for exit temperatures of cold and hot water:

$$A_o = (\pi d_o L) N_t = \pi \times 0.01905 \times L \times 728 = (43.57) \cdot L$$

$$C_h = (\dot{m} c_p)_h = 27.778 \times 4222 = 117279 \ \text{kJ/(s.K)}$$

$$C_c = (\dot{m} c_p)_c = 27.778 \times 4180 = 116306 \ \text{kJ/(s.K)}$$

$$\therefore \ C_{min} = (\dot{m} c_p)_c = 116306 \ \text{kJ/(s.K)}$$

$$C^* = \frac{C_{min}}{C_{max}} = \frac{116306}{117279} = 0.99 \approx 1$$

$$NTU = \frac{A_o U_o}{C_{min}}$$

$$\dot{Q} = \varepsilon \cdot C_{min} (T_{h1} - T_{c1})$$

$$\dot{Q} = (\dot{m} c_p)_c (T_{c2} - T_{c1}) = (\dot{m} c_p)_h (T_{h1} - T_{h2})$$

$$T_{c2} = T_{c1} + \frac{\dot{Q}}{(\dot{m} c_p)_c}$$

$$T_{h2} = T_{h1} - \frac{\dot{Q}}{(\dot{m} c_p)_h}$$

<u>Pressure drop in tube side:</u>

$$\Delta p_t = 4f \frac{L N_p}{d_i} \cdot \frac{\rho u_m^2}{2} = 4 \times 0.00613 \times \frac{L \times 2}{0.01351} \times \frac{958 \times 0.556^2}{2} = (537.5) \cdot L \ \ \text{Pa}$$

Pressure drop in shell side:

Shell-side friction factor:

$$f = \exp(0.576 - 0.19 \ln Re_s) = \exp[0.576 - 0.19 \ln(11972)] = 0.299$$

Shell-side pressure drop:

$$\phi_s = \left(\frac{\mu_b}{\mu_w}\right)^{0.14} = \left(\frac{0.000892}{0.000451}\right)^{0.14} = 1.1$$

$$N_B + 1 = \frac{L}{B} = \frac{L}{0.25} = 4L$$

$$\Delta p_s = \frac{f G_s^2 (N_b + 1) D_s}{2 \rho D_e \phi_s} = \frac{0.299 \times 564.45^2 \times (4L) \times (31 \times 0.0254)}{2 \times 997 \times 0.01829 \times 1.1} = (7479) \cdot L \ \ \text{Pa}$$

For different length (L =2, 3, 4, 5 m), the calculation is tabulated as follow:

	L = 2 m	L = 3 m	L = 4 m	L = 5 m
A_o (m^2)	87.14	130.71	174.28	217.85
NTU	1.73	2.60	3.47	4.34
ε (Fig 2.15)	0.54	0.57	0.59	0.59
Q (W)	4710393	4972082	5146540	5146540
T_{c2} (^oC)	65.5	66.2	69.2	69.2
T_{h2} (^oC)	59.8	57.6	56.1	56.1
Δp_t (Pa)	1075	1612	2150	2688
Δp_s (Pa)	14958	22437	29916	37395

b. Comparing effects of different baffle spacing :

From Table 9.3, for shell I.D. = 31 in., 2-P heat exchanger with 3/4-in. O.D. tube in 1-in. triangular pitch, the number of the tubes N_t = 728 tubes.

Tube side heat transfer coefficient:

$$u_m = \frac{\dot{m}_h}{\rho\left(\pi d_i^{\,2}/4\right)\left(N_t/N_p\right)} = \frac{27.788}{958\times\left(\pi\times 0.01351^2/4\right)\left(728/2\right)} = 0.556 \text{ m/s}$$

$$Re = \frac{\rho u_m d_i}{\mu} = \frac{958\times 0.556\times 0.01351}{2.79\times 10^{-4}} = 25792 > 2300$$

So, it is turbulent flow.

Using Prandtle's Correlation:

$$f = (1.58\ln Re - 3.28)^{-2} = \left[1.58\ln(25792) - 3.28\right]^{-2} = 0.00613$$

$$Nu = \frac{(f/2)RePr}{1.0 + 8.7(f/2)^{1/2}(Pr-1)} = \frac{(0.00613/2)\times 25792\times 1.729}{1 + 8.7\times(0.00613/2)^{1/2}\times(1.729-1)} = 184.77$$

$$h_i = \frac{Nu\cdot k}{d_i} = \frac{184.77\times 0.681}{0.01351} = 9314 \text{ W/m}^2\text{K}$$

Shell side heat transfer:

The equivalent diameter is:

$$D_e = \frac{4A_c}{P_w} = \frac{4\left(\frac{P_T^{\,2}\sqrt{3}}{4} - \frac{\pi d_o^{\,2}}{8}\right)}{\pi d_o/2} = \frac{4\left[\frac{\left(2.54\times 10^{-2}\right)^2\sqrt{3}}{4} - \frac{\pi\times 0.01905^2}{8}\right]}{\pi\times 0.01905/2} = 0.01829 \text{ m}$$

$$B = \frac{1}{(N_B + 1)} \text{ m}$$ (N_B is the number of baffles per meter of tube length)

$$A_s = \frac{D_s CB}{P_T} = \frac{(31\times 0.0254)\times(0.0254-0.01905)\times B}{0.0254} = (0.19685)\cdot B\ \ m^2$$

$$G_s = \frac{\dot{m}_c}{A_s} = \frac{27.778}{0.19685\cdot B} = \frac{141.11}{B}\ \ kg/(m^2.s)$$

$$Re = \frac{G_s D_e}{\mu} = \frac{\left(\frac{141.11}{B}\right)\times 0.01892}{8.92\times 10^{-4}} = \frac{2993.05}{B}$$

Assuming constant properties:

$$Nu = 0.36\,Re^{0.55}\ Pr^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.14} = 0.36\times\left(\frac{2993.05}{B}\right)^{0.55}\times 6.16^{0.4}\left(\frac{8.92}{4.51}\right)^{0.14} = \frac{66.906}{B^{0.55}}$$

$$h_s = \frac{Nu_s k}{D_e} = \frac{\left(\frac{66.906}{B^{055}}\right)\times 0.606}{0.01892} = \frac{2143}{B^{0.55}}\ \ W/m^2.K$$

Overall heat transfer coefficient: (assume the wall thermal conductivity k_w = 52 W/mK)

$$U_o = \left[\frac{d_o}{d_i h_i} + \frac{d_o \ln(d_o/d_i)}{2k_w} + \frac{1}{h_o}\right]^{-1}$$

$$= \left[\frac{0.01905}{0.01351\times 9314} + \frac{0.01905\ln(0.01905/0.01351)}{2\times 52} + \frac{B^{0.55}}{2143}\right]^{-1}$$

$$= \left[0.00021434 + \frac{B^{0.55}}{2143}\right]^{-1}\ \ W/m^2K$$

ε-NTU method for exit temperatures of cold and hot water:

$$A_o = (\pi d_o L)N_t = \pi\times 0.01905\times 2\times 728 = 87.14\ \ m^2$$

$$C_h = (\dot{m}c_p)_h = 27.778\times 4222 = 117279\ \ kJ/(s.K)$$

$$C_c = (\dot{m}c_p)_c = 27.778\times 4180 = 116306\ \ kJ/(s.K)$$

$$\therefore\ C_{min} = (\dot{m}c_p)_c = 116306\ \ kJ/(s.K)$$

$$C^* = \frac{C_{min}}{C_{max}} = \frac{116306}{117279} = 0.99 \approx 1$$

$$NTU = \frac{A_o U_o}{C_{min}}$$

$$\dot{Q} = \varepsilon\cdot C_{min}(T_{h1} - T_{c1})$$

$$\dot{Q} = (\dot{m}c_p)_c(T_{c2} - T_{c1}) = (\dot{m}c_p)_h(T_{h1} - T_{h2})$$

$$T_{c2} = T_{c1} + \frac{\dot{Q}}{(\dot{m}c_p)_c}$$

$$T_{h2} = T_{h1} - \frac{\dot{Q}}{\left(\dot{m}c_p\right)_h}$$

<u>Pressure drop in tube side:</u>

$$\Delta p_t = 4f\frac{LN_p}{d_i}\cdot\frac{\rho u_m^{\ 2}}{2} = 4\times 0.00613\times\frac{2\times 2}{0.01351}\times\frac{958\times 0.556^2}{2} = 1075 \ \text{ Pa}$$

Pressure drop in shell side:

Shell-side friction factor:

$$f = \exp\left(0.576 - 0.19\ln \text{Re}_s\right) = \exp\left[0.576 - 0.19\ln\left(\frac{2993.05}{B}\right)\right]$$

Shell-side pressure drop:

$$\phi_s = \left(\frac{\mu_b}{\mu_w}\right)^{0.14} = \left(\frac{0.000892}{0.000451}\right)^{0.14} = 1.1$$

$$\Delta p_s = \frac{fG_s^{\ 2}\left(\frac{2}{N_B+1}\right)D_s}{2\rho D_e\phi_s}$$

For baffle spacing (N_B /meter = 1, 2, 3, 4, 5), the calculation is tabulated as follow:

N_B/meter	1	2	3	4	5
B (m)	0.5	1/3	0.25	0.2	1/6
U_o (W/m^{2}K)	1876	2130	2315	2458	2574
NTU	1.40	1.60	1.73	1.84	1.93
ε (Fig 2.15)	0.52	0.53	0.54	0.55	0.56
Q (W)	4535934	4623164	4710393	4797622	4884852
T_{c2} (^{o}C)	64	64.75	65.5	66.25	67
T_{h2} (^{o}C)	61.3	60.6	59.8	59.1	58.3
Δp_t (Pa)	1075	1075	1075	1075	1075
Δp_s (Pa)	2060	6437	14447	27046	45144

Follow the similar procedure of heat exchanger rating, as in (a). and (b)., comparisons in (c), (d), (e) can be finished.

Problem 9.8

A sugar solution (ρ=1080 kg/m^3, c_p = 3601 J/kg.K, k_f = 0.5764 W/m.K, μ = 1.3×10^{-3} N.s/m^2) flows at rate of 60,000 kg/hr and is to be heated from 25 ^{o}C to 50 ^{o}C. Water at 95 ^{o}C is available at a flow rate of 75,000 kg/hr (c_p = 4004 J/kg.K). It is proposed to use one shell-pass and two tube-pass shell-and-tube heat exchanger containing 3/4 in. OD, 16 BWG tubes. Velocity of the sugar solution through the tube is 1.5 m/s, and the length of the heat exchanger should not be more than 3 m because of the space limitations. Assume that the shell side heat transfer coefficient is 700 W/m^2.K. Thermal conductivity of the tube material is 52 W/m.K.

a. Calculate the number of tubes, the tube side (sugar) heat transfer coefficient.
b. Calculate the overall heat transfer coefficient.
c. Calculate the length of this heat exchanger. Is this heat exchanger acceptable?

GIVEN:

- A shell and tube heat exchanger, with sugar solution through tubes and water through shell side.
- Hot water inlet temperature ($T_{h,1}$) = 95 ^{o}C
- Mass flow rate of hot water ($\dot{m}_h$) = 75,000 kg/hr = 20.833 kg/s
- Specific heat of hot water (c_p) = 4004 J/kg.K
- Sugar solution inlet temperature ($T_{c,1}$) = 25 ^{o}C
- Sugar solution outlet temperature ($T_{c,2}$) =50 ^{o}C
- Mass flow rate of sugar solution ($\dot{m}_c$) = 60,000 kg/hr = 16.667 kg/s
- Shell side heat transfer coefficient (h_o) = 700 W/m^2.K
- Velocity inside the tube (u) = 1.5 m/s
- Thermal conductivity of tube material (k) = 52 W/m.K
- Tube diameter: d_o = 3/4 in. = 0.01905 m
  d_i = 0.62 in. = 0.01575 m
  (for 3/4 O.D., 16 BWG tubes given)
- Length limitation (L) = 3 m

SOLUTION:

a.

Tube side heat transfer coefficient:

$$u_m = 1.5 \text{ m/s}$$

$$\mathrm{Pr} = \frac{\mu c_p}{k} = \frac{1.3\times10^{-3}\times3601}{0.5764} = 8.12$$

$$\mathrm{Re} = \frac{\rho u_m d_i}{\mu} = \frac{1080\times1.5\times0.01575}{1.3\times10^{-3}} = 19627 > 2300$$

So, it is turbulent flow.

Using Prandtle's Correlation:

$$f = (1.58\ln\mathrm{Re}-3.28)^{-2} = \left[1.58\ln(19627)-3.28\right]^{-2} = 0.00657$$

$$\mathrm{Nu} = \frac{(f/2)\mathrm{Re}\,\mathrm{Pr}}{1.0+8.7(f/2)^{1/2}(\mathrm{Pr}-1)} = \frac{(0.00657/2)\times19627\times8.12}{1+8.7\times(0.00657/2)^{1/2}\times(8.12-1)} = 115.05$$

$$h_i = \frac{Nu \cdot k}{d_i} = \frac{115.05 \times 0.5764}{0.01575} = 4210.5 \ \ W/m^2K$$

b.

Overall heat transfer coefficient:

$$U_o = \left[\frac{d_o}{d_i h_i} + \frac{d_o \ln(d_o/d_i)}{2k_w} + \frac{1}{h_o}\right]^{-1}$$

$$= \left[\frac{0.01905}{0.01575 \times 4210.5} + \frac{0.01905 \ln(0.01905/0.01575)}{2 \times 52} + \frac{1}{700}\right]^{-1}$$

$$= 571.21 \ \ W/m^2K$$

LMTD:

Number of tubes:

$$\dot{m}_c = \frac{\pi d_i^2}{4} \cdot \frac{N_T}{2} \cdot \rho u_m$$

$$N_T = \frac{8\dot{m}_c}{\pi d_i^2 \rho u_m} = \frac{8 \times 16.667}{\pi \times 0.01575^2 \times 1.5 \times 1080} = 105.61 \approx 106$$

Heat balance:

$$\dot{Q} = (\dot{m}c_p)_c (T_{c2} - T_{c1}) = (\dot{m}c_p)_h (T_{h1} - T_{h2})$$

$$\dot{Q} = (\dot{m}c_p)_c (T_{c2} - T_{c1}) = 16.667 \times 3601 \times (50 - 25) = 1500447 \ \ W$$

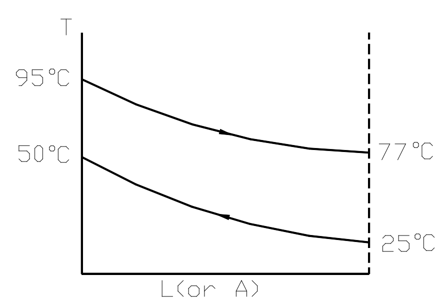
$$T_{h2} = T_{h1} - \frac{(\dot{m}c_p)_c (T_{c2} - T_{c1})}{(\dot{m}c_p)_h} = 95 - \frac{16.667 \times 3601 \times (50 - 25)}{20.833 \times 4004} = 77 \ \ ^oC$$

$$R = \frac{T_{h1} - T_{h2}}{T_{c2} - T_{c1}} = \frac{95 - 77}{50 - 25} = 0.72$$

$$P = \frac{T_{c2} - T_{c1}}{T_{h1} - T_{c1}} = \frac{50 - 25}{95 - 25} = 0.357$$

Then, F = 0.96

$$\Delta T_{lm,cf} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = \frac{45 - 52}{\ln(45/52)} = 48.42 \ \ ^oC$$



$$A_o = \frac{\dot{Q}}{U_o \cdot \Delta T_{lm,cf} \cdot F} = \frac{1500447}{571.21 \times 48.41 \times 0.96} = 56.51 \ \ m^2$$

$$A_o = \pi d_o N_T L$$

c.

$$\Rightarrow L = \frac{A_o}{\pi d_o N_T} = \frac{56.51}{\pi \times 0.01905 \times 106} = 8.9 \ \ m$$

Because L > 3m, $\therefore$ this heat exchanger is **unacceptable!**

Problem 10.1

Air at 1 atm and 400 K and with a velocity of 10 m/s flows across the compact heat exchanger shown in Figure 10.6a and exits with a mean temperature of 300 K. The core is 0.6 m long. Calculate the total frictional pressure drop between the air inlet and outlet and the average heat transfer coefficient on the air side.

SOLUTION:

At $T_m = \dfrac{T_{in} + T_{out}}{2} = \dfrac{400+300}{2} = 350$ K, 1 atm, properties of air from Appendix are:

$$\rho = 1.002 \text{ kg/m}^3$$
$$\mu = 2.08\times10^{-5} \text{ kg/m.s}$$
$$c_p = 1009 \text{ J/kg.K}$$
$$Pr = 0.708$$

From Fig. 10.6 A, we have

$$\frac{A_{min}}{A_{fr}} = \sigma = 0.443$$
$$D_h = 0.5477 \text{ cm}$$

Then

$$G = \frac{\rho U_\infty A_{fr}}{A_{min}} = \frac{\rho U_\infty}{\sigma} = \frac{1.002\times10}{0.443} = 22.62 \text{ kg/(m}^2\text{.s)}$$

$$Re = \frac{GD_h}{\mu} = \frac{22.62\times0.005477}{2.08\times10^{-5}} = 5956$$

From Fig. 10.6 A, for Re = 5956, we can obtain:

$$\frac{h}{Gc_p}Pr^{2/3} = 0.0068$$

$$h = 0.0068\times\frac{Gc_p}{Pr^{2/3}} = 0.0068\times\frac{22.62\times1009}{0.708^{2/3}} = 195.38 \text{ W/m}^2\text{.K}$$

For Re = 5956, from Fig. 10.6 A, f = 0.026

$$\Delta p_f = f\frac{G^2}{2\rho_a}\frac{A_t}{A_{min}}$$

$$\frac{A_t}{A_{min}} = \frac{4L}{D_h} = \frac{4\times0.6}{0.005477} = 438.196$$

Then

$$\Delta p_f = 0.026\times\frac{22.62^2}{2\times1.002}\times438.196 = 2909 \text{ Pa}$$

Problem 10.2

Air enters the core of a finned tube heat exchanger of the type shown in Figure 10.4 at 2 atm and 150°C. The air mass flow rate is 10 kg/s and flows perpendicular to the tubes. The core is 0.5 m long with a 0.30 m² frontal area. The height of the core is 0.5 m. Water at 15°C and at a flow rate of 50 kg/s flows inside the tubes. Airside data are given on Figure 10.4. For water side data, assume that σ_w = 0.129, D_h = 0.373 cm, and waterside heat transfer area/total volume = 138 m²/m³. This is a rating problem. Calculate:

a. The airside and waterside heat transfer coefficients
b. Overall heat transfer coefficient based on the outer (airside) surface area
c. Total heat transfer rate and outlet temperatures of air and water

GIVEN:

Air side:

p = 2 atm, $T_{in,a}$ = 150 °C, $\dot{m}_a$ = 10 kg/s

A_f = 0.30 m², Core length = 0.5 m, Core height = 0.5 m.

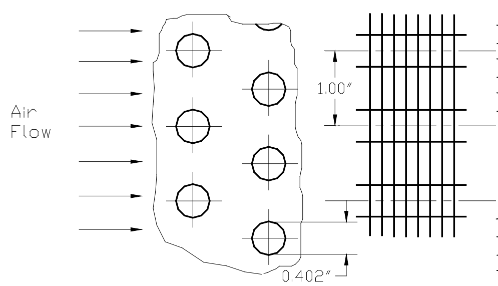
Water side:

$T_{in,w}$ = 15 °C, $\dot{m}_w$ = 50 kg/s

σ_w = 0.129, $D_{h,w}$ = 0.373, Water side heat transfer area/total volume = 138 m²/m³

FIND:

a. h_a = ?, h_w = ?
b. U_a = ?
c. $T_{out,a}$ = ?, $T_{out,w}$ = ?



Fin thickness, t = 0.033 cm
Fin Area/ Total area = 0.839
Fin Length L:

$$L = \frac{(1.00 - 0.402)}{2} = 0.299'' = 0.7595 \text{ cm}$$

Air passage hydraulic diameter, D_h=0.3633 m
Free flow area/Frontal area, σ = 0.534
Heat transfer area/ Total volume = 587 m²/m³

SOLUTION:

For the air side, we assume a mean temperature 80°C and determine the physical properties of air at p_a = 2 atm.

$$\rho_a = 1.9816 \text{ kg/m}^3$$
$$\mu_a = 2.1 \times 10^{-5} \text{ kg/m.s}$$
$$c_{p,a} = 1009 \text{ J/kg.K}$$
$$Pr_a = 0.708$$

Assuming an average temperature of 15°C for the water, we take the physical properties as:

$$\rho_w = 999 \ \ \text{kg / m}^3$$
$$\mu_w = 11.54 \times 10^{-4} \ \ \text{kg / m.s}$$
$$c_{p,w} = 4186 \ \ \text{J / kg.K}$$
$$\text{Pr}_w = 8.20$$
$$k_w = 0.591 \ \ \text{W / m.K}$$

a. Heat transfer coefficient:

Air side:

$$G_a = \frac{\dot{m}_a}{A_{min}} = \frac{\dot{m}_a}{\sigma_a \cdot A_{fr}} = \frac{10}{0.534 \times 0.30} = 62.42 \ \ \text{kg / (m}^2\text{.s)}$$

$$\text{Re} = \frac{G_a D_{h,a}}{\mu_a} = \frac{62.42 \times 0.003633}{2.1 \times 10^{-5}} = 10799$$

From Fig. 10.4, for Re = 10799, we have:

$$\frac{h_a}{G_a c_{p,a}} \text{Pr}_a^{\ 2/3} = 0.004$$

$$h_a = 0.004 \times \frac{G_a c_{p,a}}{\text{Pr}_a^{\ 2/3}} = 0.004 \times \frac{62.42 \times 1009}{0.708^{2/3}} = 317.1 \ \ \text{W / (m}^2\text{. K)}$$

Water side:

$$A_{fr,w} = 0.5 \times 0.5 = 0.25 \ \ \text{m}^2$$

$$G_w = \frac{\dot{m}_w}{A_{\min}} = \frac{\dot{m}_w}{\sigma_w \cdot A_{fr,w}} = \frac{50}{0.129 \times 0.3} = 1550.39 \ \ \text{kg/(m}^2.s)$$

$$\text{Re}_w = \frac{G_w D_{h,w}}{\mu_w} = \frac{1550.39 \times 0.00373}{11.54 \times 10^{-4}} = 5011$$

Using Gnielinski’s correlation, we have:

$$f = (1.58 \ln \text{Re} - 3.28)^{-2} = 0.00965$$
$$Nu = \frac{(f/2)(\text{Re} - 1000)\text{Pr}}{1 + 12.7(f/2)^{1/2}(\text{Pr}^{2/3} - 1)} = \frac{(0.00965/2) \times (5011 - 1000) \times 8.2}{1 + 12.7 \times (0.00965/2)^{0.5}(8.2^{2/3} - 1)} = 42.8$$

So,

$$h_w = \frac{k_w}{D_e} \cdot Nu = \frac{0.591}{0.00373} \times 42.8 = 6781.4 \ \ \text{W/(m}^2.K)$$

b. Overall heat transfer coefficient U_a:

To determine U_{air}, based on the air-side surface, the fin efficiency η should be determined. The reason is that the effective temperature difference between the fluid and the fin surface is lower than that between the fluid and the fin base. Only the air side has extended surfaces which can be regarded as plate fins, the thermal conductivity of fin material, k is given as:

k = 212 W/(m.K) for aluminum

The fin efficiency η for a plate fin can be calculated as:

$$\eta_f = \frac{\tanh(mL)}{mL}$$

where $m = \sqrt{\dfrac{2h_a}{k\ t}}$

Then $mL = \sqrt{\dfrac{2 \times 317.1}{212 \times 0.033 \times 10^{-2}}} \times 0.7595 \times 10^{-2} = 0.7231$

and $\eta_f = \dfrac{\tanh(mL)}{mL} = \dfrac{\tanh(0.7231)}{0.7231} = 0.856$

The air-weighted fin efficiency is determined by the following Eq.:

$$\eta = \left[1 - \frac{A_f}{A}\left(1 - \eta_f\right)\right] = \left[1 - 0.839 \times \left(1 - 0.856\right)\right] = 0.879$$

The overall heat transfer coefficient, U_a, based on the air-side surface, is:

$$\frac{1}{U_a} = \frac{1}{\eta h_a} + \frac{1}{\left(A_w / A_a\right) h_w}$$

Here the ratio of the water-side to air-side heat transfer surfaces is found to be

$$\frac{A_w}{A_a} = \frac{\text{water - side heat transfer area / total volume}}{\text{air - side heat transfer area / total volume}} = \frac{138}{587} = 0.235$$

Then

$$\frac{1}{U_a} = \frac{1}{0.879 \times 317.1} + \frac{1}{0.235 \times 6781.4}$$

$$U_a = 238\ \text{W/(m}^2.K)$$

c. Total heat transfer rate and outlet temperatures of air and water:

$$C_a = \dot{m}_a c_{p,a} = 10 \times 1009 = 10090\ \text{W / K}$$

$$C_w = \dot{m}_w c_{p,w} = 50 \times 4186 = 209300\ \text{W / K}$$

So, $C_{min} = C_a = 10090\ \text{W / K}$

and $\dfrac{C_{min}}{C_{max}} = \dfrac{10090}{209300} = 0.048$

Total volume of matrix, $V = A_{fr} \times$ (core length) = 0.3×0.5 = 0.15 m^3.

$$\frac{A_a}{V} = 587 \quad \text{(from Fig. 9.4)}$$

Then $A_a = 587 \cdot V = 587 \times 0.15 = 88.05\ \text{m}^2$

$$NTU = \frac{A_a U_a}{C_{min}} = \frac{88.05 \times 230}{10090} = 2.0$$

From Fig. 2.15, for a cross-flow heat exchanger with both fluids unmixed, for NTU = 2.0, and $C_{min}/C_{max} = 0.048$, we obtain:

$$\varepsilon = 0.85$$

Then the total heat transfer rate, Q becomes:

$$\begin{aligned} Q &= \varepsilon C_{min}\left(T_{in,a} - T_{in,w}\right) \\ &= 0.85 \times 10090 \times \left(150 - 15\right) \\ &= 1158\ \text{kW} \end{aligned}$$

From the heat balance

$$Q = \dot{m}_a c_{p,a}\left(T_{in,a} - T_{out,a}\right) = \dot{m}_w c_{p,w}\left(T_{out,w} - T_{in,w}\right)$$

the outlet temperature can be obtained as:

$$T_{out,a} = T_{in,a} - \frac{Q}{\dot{m}_a c_{p,a}} = 150 - \frac{1158 \times 10^3}{10090} = 35.2\ ^{o}C$$

$$T_{out,w} = T_{in,w} + \frac{Q}{\dot{m}_w c_{p,w}} = 15 + \frac{1158 \times 10^3}{209300} = 20.5\ ^{o}C$$

Problem 10.3

Hot air at 2 atm and 500 K at a rate of 8 kg/s flows across a circular finned-tube matrix configuration shown in Figure 10.6. The frontal area of the heat exchanger is 0.8 m × 0.5 m and the core is 0.5 m long. Geometrical configurations are shown in Figure 10.6. Calculate:

a. the heat transfer coefficient;

b. the total frictional pressure drop between the air inlet and outlet.

GIVEN:

Hot Air flows across a circular finned-tube matrix configuration:

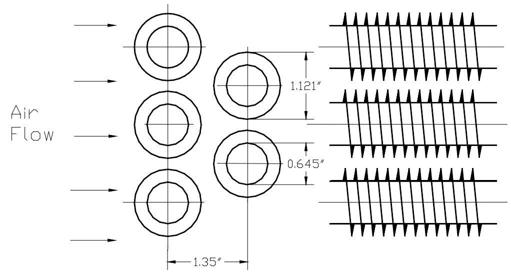
$p = 2$ atm, $T_{in,a} = 500$ K, $\dot{m}_a = 8$ kg/s

$A_f = 0.8 \times 0.5$ m^2, Core length = 0.5 m.

FIND:

a. $h = ?$

b. $\Delta p_f = ?$



Air-passage hydraulic Diameter, $D_h = 0.5477$ cm

Free flow area/ frontal area, $\sigma = 0.443$

Heat transfer area/total volume = 323.8 m^2/m^3

SOLUTION:

At 500 K, 2 atm, properties of air from Appendix are:

$$\rho = 1.416 \ \text{kg/m}^3$$

$$\mu = 2.687 \times 10^{-5} \ \text{kg/m.s}$$

$$c_p = 1030 \ \text{J/kg.K}$$

$$Pr = 0.68$$

From Fig. 10.6 A, we have

$$\frac{A_{min}}{A_{fr}} = \sigma = 0.443$$

$$D_h = 0.5477 \ \text{cm}$$

Then

$$G = \frac{\dot{m}_a}{A_{min}} = \frac{\dot{m}_a}{\sigma \cdot A_{fr}} = \frac{8}{0.443 \times 0.8 \times 0.5} = 45.15 \ \text{kg/(m}^2\text{.s)}$$

$$Re = \frac{GD_h}{\mu} = \frac{45.15 \times 0.005477}{2.687 \times 10^{-5}} = 9203$$

From Fig. 10.6 A, for Re = 9203, we can obtain:

$$\frac{h}{Gc_p} Pr^{2/3} = 0.0058$$

$$h = 0.0058 \times \frac{Gc_p}{Pr^{2/3}} = 0.0058 \times \frac{45.15 \times 1030}{0.68^{2/3}} = 349\ W/m^2.K$$

For Re = 9203, from Fig. 10.6 A, f = 0.025

$$\Delta p_f = f\frac{G^2}{2\rho_a}\frac{A_t}{A_{min}}$$

$$\frac{A_t}{A_{min}} = \frac{4L}{D_h} = \frac{4 \times 0.5}{0.005477} = 365.2$$

Then

$$\Delta p_f = 0.025 \times \frac{45.15^2}{2 \times 1.416} \times 365.2 = 6571\,Pa$$

Problem 10.4

Repeat Problem 10.3 for a finned-tube matrix shown in Figure 10.5 and discuss the results.

GIVEN:

Hot Air flows across a circular finned tube matrix configuration:

$p = 2$ atm, $T_{in,a} = 500$ K, $\dot{m}_a = 8$ kg/s

$A_f = 0.8 \times 0.5$ m^2, Core length = 0.5 m.

FIND:

a. h = ?

b. Δp_f = ?

Air-passage hydraulic Diameter, $D_h = 0.443$ cm

Free flow area/ frontal area, $\sigma = 0.494$

Heat transfer area/total volume = 446 m^2/m^3

SOLUTION:

At 500 K, 2 atm, properties of air from Appendix are:

$$\rho = 1.416 \ \ \text{kg/m}^3$$

$$\mu = 2.687 \times 10^{-5} \ \ \text{kg/m.s}$$

$$c_p = 1030 \ \ \text{J/kg.K}$$

$$\text{Pr} = 0.68$$

Then

$$G = \frac{\dot{m}_a}{A_{min}} = \frac{\dot{m}_a}{\sigma \cdot A_{fr}} = \frac{8}{0.494 \times 0.8 \times 0.5} = 40.48 \ \ \text{kg/(m}^2\text{.s)}$$

$$\text{Re} = \frac{GD_h}{\mu} = \frac{40.48 \times 0.00443}{2.687 \times 10^{-5}} = 6674$$

From Fig. 10.5, for Re = 6674, we can obtain:

$$\frac{h}{Gc_p}\text{Pr}^{2/3} = 0.0067$$

$$h = 0.0067 \times \frac{Gc_p}{\text{Pr}^{2/3}} = 0.0067 \times \frac{40.48 \times 1030}{0.68^{2/3}} = 361 \ \text{W/m}^2\text{.K}$$

For Re = 6674, from Fig. 10.5, f = 0.035

$$\Delta p_f = f\frac{G^2}{2\rho_a}\frac{A_t}{A_{min}}$$

$$\frac{A_t}{A_{min}} = \frac{4L}{D_h} = \frac{4 \times 0.5}{0.00443} = 451.5$$

Then

$$\Delta p_f = 0.035 \times \frac{40.64^2}{2 \times 1.416} \times 451.5 = 9216 \ \text{Pa}$$

Problem 10.5

Repeat Problem 10.1 for the heat exchanger matrix configuration in Figure 10.8 for the matrix 9.1-0.737-s as given in Table 10.1.

GIVEN:

Hot Air flows across a circular finned-tube matrix configuration:

$p = 2$ atm, $T_{in,a} = 500$ K, $\dot{m}_a = 8$ kg/s

$A_f = 0.8\times0.5\ m^2$, Core length = 0.5 m.

FIND:

a. $h = ?$

b. $\Delta p_f = ?$

SOLUTION:

At 500 K, 2 atm, properties of air from Appendix are:

$$\rho = 1.416\ \ kg/m^3$$

$$\mu = 2.687\times10^{-5}\ \ kg/m.s$$

$$c_p = 1030\ \ J/kg.K$$

$$Pr = 0.68$$

From Table 10.1, we have

$$\frac{A_{min}}{A_{fr}} = \sigma = 0.788$$

$$D_h = 0.3565\ cm$$

Then
$$G = \frac{\dot{m}_a}{A_{min}} = \frac{\dot{m}_a}{\sigma\cdot A_{fr}} = \frac{8}{0.788\times0.8\times0.5} = 25.38\ \ kg/(m^2.s)$$

$$Re = \frac{GD_h}{\mu} = \frac{25.38\times0.003565}{2.687\times10^{-5}} = 3367$$

From Fig. 10.8 and Fig. 10.7, for Re = 3367, we can obtain:

$$\frac{h}{Gc_p}Pr^{2/3} = 0.0057$$

$$h = 0.0057\times\frac{Gc_p}{Pr^{2/3}} = 0.0057\times\frac{25.38\times1029.5}{0.68^{2/3}} = 192.7\ W/m^2.K$$

For Re =3367, from Fig. 10.8 and Fig. 10.7, f = 0.018

$$\Delta p_f = f\frac{G^2}{2\rho_a}\frac{A_t}{A_{min}}$$

$$\frac{A_t}{A_{min}} = \frac{4L}{D_h} = \frac{4\times0.5}{0.003565} = 561$$

Then
$$\Delta p_f = 0.018\times\frac{25.38^2}{2\times1.416}\times561 = 2297\ \ Pa$$

Problem 10.6

Repeat Problem 10.2 for the heat exchanger matrix configuration shown in Figure 10.8 for the surface 11.32-0.737-S-R (see Table 10.1).

GIVEN:

Hot Air flows across a circular finned-tube matrix configuration:

$p = 2$ atm, $T_{in,a} = 500$ K, $\dot{m}_a = 8$ kg/s

$A_f = 0.8 \times 0.5$ m^2, Core length = 0.5 m.

FIND:

a. $h = ?$

b. $\Delta p_f = ?$

SOLUTION:

At 500 K, 2 atm, properties of air from Appendix are:

$$\rho = 1.416 \ \ \text{kg/m}^3$$

$$\mu = 2.687 \times 10^{-5} \ \ \text{kg/m.s}$$

$$c_p = 1030 \ \ \text{J/kg.K}$$

$$\text{Pr} = 0.68$$

From Table 10.1, we have

$$\frac{A_{min}}{A_{fr}} = \sigma = 0.78$$

$$D_h = 0.3434 \text{ cm}$$

Then $$G = \frac{\dot{m}_a}{A_{min}} = \frac{\dot{m}_a}{\sigma \cdot A_{fr}} = \frac{8}{0.78 \times 0.8 \times 0.5} = 25.64 \ \ \text{kg/(m}^2\text{.s)}$$

$$\text{Re} = \frac{G D_h}{\mu} = \frac{25.64 \times 0.003434}{2.687 \times 10^{-5}} = 3277$$

From Fig. 10.8 and Fig. 10.7, for Re = 3277, we can obtain:

$$\frac{h}{G c_p} \text{Pr}^{2/3} = 0.0063$$

$$h = 0.0063 \times \frac{G c_p}{\text{Pr}^{2/3}} = 0.0063 \times \frac{25.64 \times 1030}{0.68^{2/3}} = 215 \text{ W/m}^2\text{.K}$$

For Re =3277, from Fig. 10.8 and Fig. 10.7, f = 0.025

$$\Delta p_f = f \frac{G^2}{2\rho_a} \frac{A_t}{A_{min}}$$

$$\frac{A_t}{A_{min}} = \frac{4L}{D_h} = \frac{4 \times 0.5}{0.003434} = 582.4$$

Then $$\Delta p_f = 0.025 \times \frac{25.64^2}{2 \times 1.416} \times 582.4 = 3380 \ \ \text{Pa}$$

Problem 10.7

Air at 2 atm and 500 K and at a rate of 12 kg/s flows across a plain plate-fin matrix of the configuration shown in Figure 10.8 and in Table 10.1 for the surface 9.68-0.870. The frontal area is 0.8 m × 0.6 m and the length of the matrix is 0.6 m. Calculate:
a. the heat transfer coefficient;
b. friction coefficient.

GIVEN:

Hot Air flows across a plain plate-fin matrix configuration:

$p = 2$ atm, $T_{in,a} = 500$ K, $\dot{m}_a = 12$ kg/s

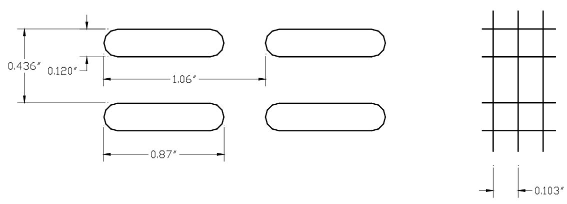
$A_f = 0.8 \times 0.6$ m^2, Core length = 0.6 m.

FIND:

a. $h = ?$

b. $\Delta p_f = ?$

SOLUTION:



Surface 9.68-0.870

At 500 K, 2 atm, properties of air from Appendix are:

$$\rho = 1.416 \ \text{kg/m}^3$$

$$\mu = 2.687 \times 10^{-5} \ \text{kg/m.s}$$

$$c_p = 1030 \ \text{J/kg.K}$$

$$Pr = 0.68$$

From Table 10.1, we have

$$\frac{A_{min}}{A_{fr}} = \sigma = 0.697$$

$$D_h = 0.2997 \text{ cm}$$

Then
$$G = \frac{\dot{m}_a}{A_{min}} = \frac{\dot{m}_a}{\sigma \cdot A_{fr}} = \frac{12}{0.697 \times 0.8 \times 0.6} = 35.9 \ \text{kg/(m}^2\text{.s)}$$

$$Re = \frac{GD_h}{\mu} = \frac{35.9 \times 0.002997}{2.687 \times 10^{-5}} = 4001$$

From Fig. 10.8 and Fig. 10.7, for Re = 4001, we can obtain:

$$\frac{h}{Gc_p} Pr^{2/3} = 0.0038$$

$$h = 0.0038 \times \frac{Gc_p}{Pr^{2/3}} = 0.0038 \times \frac{35.9 \times 1030}{0.68^{2/3}} = 181.7 \ \text{W/m}^2\text{.K}$$

For Re =4001, from Fig. 10.8 and Fig. 10.7, f = 0.015

$$\Delta p_f = f\frac{G^2}{2\rho_a}\frac{A_t}{A_{min}}$$

$$\frac{A_t}{A_{min}} = \frac{4L}{D_h} = \frac{4 \times 0.6}{0.002997} = 800.8$$

Then $$\Delta p_f = 0.015 \times \frac{35.9^2}{2 \times 1.416} \times 800.8 = 5467 \text{ Pa}$$

Problem 10.8

Repeat Problem 10.5 for a finnedtube matrix of the configuration shown in Figure 10.8 for the surface 9.68-0.870-R (see Figure 10.7) geometrical configuration given in Table 10.1.

GIVEN:

Hot Air flows across a plain plate-fin matrix configuration:

$p = 2$ atm, $T_{in,a} = 500$ K, $\dot{m}_a = 12$ kg/s

$A_f = 0.8 \times 0.6$ m^2, Core length = 0.6 m.

FIND:

a. $h = ?$

b. $\Delta p_f = ?$

SOLUTION:

At 500 K, 2 atm, properties of air from Appendix are:

$$\rho = 1.416 \ \text{kg/m}^3$$

$$\mu = 2.687 \times 10^{-5} \ \text{kg/m.s}$$

$$c_p = 1030 \ \text{J/kg.K}$$

$$\text{Pr} = 0.68$$

From Table 10.1, we have

$$\frac{A_{min}}{A_{fr}} = \sigma = 0.697$$

$$D_h = 0.2997 \text{ cm}$$

Then $$G = \frac{\dot{m}_a}{A_{min}} = \frac{\dot{m}_a}{\sigma \cdot A_{fr}} = \frac{12}{0.697 \times 0.8 \times 0.6} = 35.9 \ \text{kg/(m}^2\text{.s)}$$

$$\text{Re} = \frac{GD_h}{\mu} = \frac{35.9 \times 0.002997}{2.687 \times 10^{-5}} = 4001$$

From Fig. 10.8 and Fig. 10.7, for Re = 4001, we can obtain:

$$\frac{h}{Gc_p}\text{Pr}^{2/3} = 0.005$$

$$h = 0.0063 \times \frac{Gc_p}{\text{Pr}^{2/3}} = 0.005 \times \frac{35.9 \times 1030}{0.68^{2/3}} = 239.1 \,\text{W/m}^2.K$$

For Re =4001, from Fig. 10.8 and Fig. 10.7, f = 0.023

$$\Delta p_f = f \frac{G^2}{2\rho_a}\frac{A_t}{A_{min}}$$

$$\frac{A_t}{A_{min}} = \frac{4L}{D_h} = \frac{4 \times 0.6}{0.002997} = 800.8$$

Then $$\Delta p_f = 0.023 \times \frac{35.9^2}{2 \times 1.416} \times 800.8 = 8382 \ \text{Pa}$$

Problem 10.9

An air-to-water compact heat exchanger is to be designed to serve as an intercooler in a gas turbine plant. Geometrical details of the proposed surface (surface 9.29-0.737-S-R) for the air side are given in Figure 10.8 and Table 10.1. Hot air at 2 atm and 400 K with a flow rate of 20 kg/s flows across the matrix. The outlet temperature of air is 300 K and the allowable pressure drop is 0.3 bar. Water at 17°C and a flow rate of 50 kg/s flows inside the flat tubes. Water velocity is 1.5 m/s. Waterside geometrical details are:

$$D_h = 0.373 \text{ m}$$

$$\sigma_w = 0.129$$

$$\frac{\text{Water} - \text{side heat transfer are}}{\text{total volume}} = 138 \text{ m}^2/\text{m}^3$$

Calculate:

a. Overall heat transfer coefficient based on air side;

b. Air flow frontal area, A_{fr};

c. Flow length and core size.

GIVEN:

An compact heat exchanger with surface 9.29-0.737-S-R.

Air side:

p = 2 atm, $T_{in,a}$ = 400K, $T_{out,a}$ = 300K, $\dot{m}_a$ = 20 kg/s

Water side:

$T_{in,w}$ = 17 °C, $\dot{m}_w$ = 50 kg/s

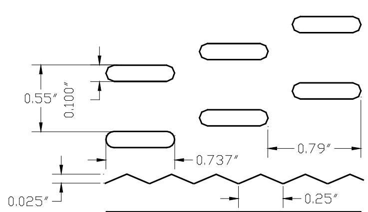
σ_w = 0.129, $D_{h,w}$ = 0.373, Water side heat transfer area/total volume = 138 m^2/m^3

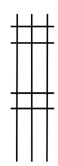
FIND:

a. h_a = ?, h_w = ? U_a = ?

b. A_{fr} = ?

c. L = ?





Fin thickness, t = 0.0102 cm

Fin Area/ Total area = 0.845

Fin Length L:

$$L = \frac{(0.55 - 0.100)}{2} = 0.225'' = 0.5715 \text{ cm}$$

Air passage hydraulic diameter, D_h=0.351 cm

Free flow area/Frontal area, σ = 0.788

Heat transfer area/ Total volume = 885.8 m^2/m^3

SOLUTION:

Properties of air at p_a = 2 atm, $T_m = \frac{400+300}{2} = 350\text{K}$:

$$\rho_a = 2.004 \text{ kg/m}^3, \quad c_{p,a} = 1009 \text{ J/(kg.K)}, \quad \mu_a = 20.8 \times 10^{-6} \text{ Pa.s}$$

$$Pr_a = 0.708$$

Properties of water at 17 °C (or 273.15+17=290.15K):

$$\rho_w = 999 \text{ kg/m}^3, \quad c_{p,w} = 4184 \text{ J/(kg.K)}, \quad \mu_w = 10.84 \times 10^{-4} \text{ Pa.s}$$

$$Pr_w = 7.65, \quad k_w = 0.594$$

The thermal conductivity of aluminum is taken as:

k_{al} = 250 (W/m.K)

The solution requires iteration. For the first iteration, let $Re = 10^4 = \frac{G_a D_h}{\mu_a}$:

<u>Mass velocity:</u>

$$G_a = \frac{Re \cdot \mu_a}{D_h} = \frac{10^4 \times 20.8 \times 10^{-6}}{0.00351} = 59.26 \ \ kg/(m^2.s)$$

<u>Air side heat transfer coefficient:</u>

From Fig. 10.7, for Re = 10^4, we have:

$$\frac{h_a}{G_a c_{p,a}} Pr_a^{2/3} = 0.0053, \qquad f = 0.021$$

$$h_a = 0.0053 \times \frac{G_a c_{p,a}}{Pr_a^{2/3}} = 0.0053 \times \frac{59.26 \times 1009}{0.708^{2/3}} = 399 \ \ W/(m^2.K)$$

<u>Fin efficiency:</u>

$$\eta_f = \frac{\tanh(mL)}{mL}$$

where $$m = \sqrt{\frac{2h_a}{k \ t}}$$

Then $$mL = \sqrt{\frac{2 \times 399}{250 \times 0.0102 \times 10^{-2}}} \times 0.5715 \times 10^{-2} = 1.01$$

and $$\eta_f = \frac{\tanh(mL)}{mL} = \frac{\tanh(1.03)}{1.03} = 0.76$$

so, the total fin efficiency is:

$$\eta = \left[1 - \frac{A_f}{A}\left(1 - \eta_f\right)\right] = \left[1 - 0.845 \times (1 - 0.76)\right] = 0.80$$

Since $G_a = \frac{\dot{m}_a}{A_{free\ flow}}$, then

$$A_{free\ flow} = \frac{\dot{m}_a}{G_a} = \frac{20}{59.26} = 0.3375 \ \ m^2$$

$$A_{fr} = \frac{A_{free\ flow}}{\sigma} = \frac{0.3375}{0.788} = 0.428 \ \ m^2$$

<u>Water-side heat transfer coefficient:</u>

Assume the frontal areas for water side and air side are the same,

$$G_w = \frac{\dot{m}_w}{A_w} = \frac{\dot{m}_w}{\sigma_w \cdot A_{fr}} = \frac{50}{0.129 \times 0.428} = 905.6$$

$$Re_w = \frac{G_w D_{h,w}}{\mu_w} = \frac{905.6 \times 0.00373}{10.84 \times 10^{-4}} = 3116$$

Using Gnielinski's correlation for low Re ($2300 < Re < 10^4$)

$$f = (1.58 \ln \text{Re} - 3.28)^{-2} = 0.0112$$

$$Nu = \frac{(f/2)(\text{Re}-1000)\text{Pr}}{1+12.7(f/2)^{1/2}(\text{Pr}^{2/3}-1)} = \frac{(0.0112/2)\times(3116-1000)\times 7.65}{1+12.7\times(0.0112/2)^{1/2}(7.65^{2/3}-1)} = 24.2$$

$$h_w = \frac{k_w}{D_e} Nu = \frac{0.594}{0.00373}\times 24.2 = 3854 \text{ W/(m}^2.K)$$

Overall heat transfer coefficient:

The overall heat transfer coefficient, U_a, based on the air-side surface, is:

$$\frac{1}{U_a} = \frac{1}{\eta h_a} + \frac{1}{(A_w/A_a)h_w}$$

where

$$\frac{A_w}{A_a} = \frac{\text{water-side heat transfer area / total volume}}{\text{air-side heat transfer area / total volume}} = \frac{138}{885.8} = 0.156$$

Then

$$\frac{1}{U_a} = \frac{1}{0.80\times 399} + \frac{1}{0.156\times 3854}$$

$$U_a = 208.5 \text{ W/(m}^2.K)$$

Heat balance:

$$Q = \dot{m}_a c_{p,a}(T_{in,a} - T_{out,a}) = \dot{m}_w c_{p,w}(T_{out,w} - T_{in,w})$$

$$T_{out,w} = T_{in,w} + \frac{\dot{m}_a c_{p,a}}{\dot{m}_w c_{p,w}}(T_{in,a} - T_{out,a}) = 17 + \frac{20\times 1009}{50\times 4184}(400-300) = 26.6\ ^oC = 300K$$

$$\varepsilon = \frac{C_h(T_{in,a} - T_{out,a})}{C_{min}(T_{in,a} - T_{in,w})} = \frac{400-300}{400-290.15} = 0.91$$

$$\frac{C_{min}}{C_{max}} = \frac{20\times 1009}{50\times 4184} = 0.096$$

from Fig. 2.15, we obtain NTU = 3.5

$$NTU = \frac{A_a U_a}{C_{min}}$$

$$A_a = \frac{NTU\cdot C_{min}}{U_a} = \frac{3.5\times(20\times 1009)}{208.5} = 339 \text{ m}^2$$

Heat exchange volume:

$$V = \frac{A_a}{748} = \frac{339}{855.8} = 0.40 \text{ m}^3$$

Core length:

$$L = \frac{V}{A_{fr}} = \frac{0.40}{0.428} = 0.9 \text{ m}$$

The core pressure drop:

$$\Delta p = f \cdot \frac{G^2}{2\rho} \cdot \frac{4L}{D_h} = 0.021 \times \frac{59.26^2}{2 \times 2.004} \times \frac{4 \times 0.9}{0.00351} = 18872 \text{ Pa} < 0.3 \text{ bar}$$

The pressure drop is less than the limitation given in problem, so this is an acceptable design.

Problem 10.10

Repeat Problem 10.7 for air at a flow rate of 15 kg/s while everything else remains the same.

GIVEN:

Hot Air flows across a plain plate-fin matrix configuration:

$p = 2$ atm, $T_{in,a} = 500$ K, $\dot{m}_a = 15$ kg/s

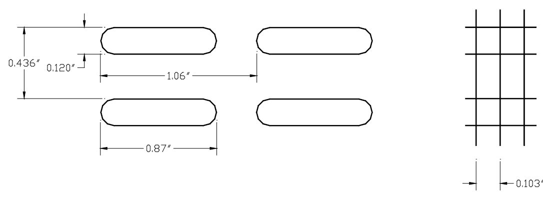
$A_f = 0.8 \times 0.6$ m^2, Core length = 0.6 m.

FIND:

a. $h = ?$

b. $\Delta p_f = ?$

SOLUTION:



Surface 9.68-0.870

At 500 K, 2 atm, properties of air from Appendix are:

$$\rho = 1.416 \ \text{kg/m}^3$$

$$\mu = 2.687 \times 10^{-5} \ \text{kg/m.s}$$

$$c_p = 1030 \ \text{J/kg.K}$$

$$Pr = 0.68$$

From Table 10.1, we have

$$\frac{A_{min}}{A_{fr}} = \sigma = 0.697$$

$$D_h = 0.2997 \text{ cm}$$

Then $$G = \frac{\dot{m}_a}{A_{min}} = \frac{\dot{m}_a}{\sigma \cdot A_{fr}} = \frac{15}{0.697 \times 0.8 \times 0.6} = 44.8 \ \text{kg/(m}^2\text{.s)}$$

$$Re = \frac{GD_h}{\mu} = \frac{44.8 \times 0.002997}{2.687 \times 10^{-5}} = 4997$$

From Fig. 10.8 and Fig. 10.7, for Re = 4997, we can obtain:

$$\frac{h}{Gc_p} Pr^{2/3} = 0.0037$$

$$h = 0.0037 \times \frac{Gc_p}{Pr^{2/3}} = 0.0037 \times \frac{44.8 \times 1030}{0.68^{2/3}} = 220.8 \ \text{W/m}^2\text{.K}$$

For Re = 4997, from Fig. 10.8 and Fig. 10.7, f = 0.014

$$\Delta p_f = f \frac{G^2}{2\rho_a} \frac{A_t}{A_{min}}$$

$$\frac{A_t}{A_{min}} = \frac{4L}{D_h} = \frac{4 \times 0.6}{0.002997} = 800.8$$

Then $$\Delta p_f = 0.014 \times \frac{44.8^2}{2 \times 1.416} \times 800.8 = 7945 \text{ Pa}$$

Problem 10.11

An aircooled refrigerant condenser is to be designed. A flattened tube with corrugated fins will be used. The surface selected for the matrix is similar to Figure 10.4. The cooling load (heat duty) is 125 kW. The refrigerant 134A condenses inside the tubes at 310 K. Air enters the condenser at 18°C and leaves at 26°C. The mean pressure is 2 atm. Calculate:
a. Air-side heat transfer coefficient
b. Tube-side condensation heat transfer coefficient
c. Overall heat transfer coefficient
d. Core dimensions based on the following parameters:
Air-side geometric configurations for the surface 7.75-5/8 T^3
Tube O.D. = 1.717 m
Tube arrangement = staggered
Fins/in. = 7.75
Fin type = plain
Fin thickness = 0.41×10^{-3} m
Minimum flow area/frontal area, σ = 0.481
Hydraulic diameter, $D_h = 3.48 \times 10^{-3}$ m
Heat transfer area/total volume, α = 554 m^2/m^3.

GIVEN:

Cold Air flows across a plain plate-fin matrix configuration similar to that in Fig. 10.4:
p_m = 2 atm, $T_{in,a}$ = 18 °C, $T_{out,a}$ = 26 °C
Refrigerant side:
T = 310 K
Cooling load: Q = 125 kW.

Geometrical size of the core is listed in the problem statement

SOLUTION:

Fins per in. = 7.75
D_h *= 0.00348 m*
Fin thickness t = 0.00041cm
Heat transfer area/Total volume α *= 554* m^2/m^3
Free flow area/ frontal area σ *= 0.481*
Tube O.D. = 1.717 cm

Refrigerant side:
Assume the tube wall thickness to be 0.2 cm.
Frontal area associated with one tube: 0.866"×1.0"=0.866 $in^2 = 5.587\times10^{-4}\ m^2$

$$\sigma_w = \frac{A_t}{5.587\times10^{-4}} = \frac{(\pi/4)\times d_i^{\,2}}{5.587\times10^{-4}} = \frac{(\pi/4)\times 0.01317^2}{5.587\times10^{-4}} = 0.244$$

$$\text{Heat transfer area/Volume} = \frac{\pi\times 0.01317}{5.587\times10^{-4}} = 74$$

At $\frac{18+26}{2} = 22\,^{o}C$, p_m = 2 atm, properties of air from Appendix are:

$$\rho = 2.2265 \ \ kg/m^3$$
$$\mu = 2.02 \times 10^{-5} \ \ kg/m.s$$
$$c_p = 1009 \ \ J/kg.K$$
$$Pr = 0.71$$
$$k = 0.0274$$

Properties of refrigerant from Appendix are (310K):

$$p_{sat} = 9.33 \ \ bar$$
$$\rho_l = 1156 \ \ kg/m^3 \qquad \rho_g = 45.45 \ \ kg/m^3$$
$$\mu_l = 1.89 \times 10^{-4} \ \ kg/m.s \qquad \mu_g = 0.125 \times 10^{-4} \ \ kg/m.s$$
$$c_p = 1.497 \ \ J/kg.K$$
$$Pr_l = 3.98$$
$$k_l = 0.071 \ \ W/(m.K)$$
$$h_{fg} = 165.3 \ \ kJ/kg$$

Thermal conductivity of aluminum is taken as k_{al} = 250 W/(m.K)

Mass flow rates of air and refrigerant:

$$Q = \dot{m}_f h_{fg} = \dot{m}_a c_{p,a}\left(T_{c2} - T_{c1}\right)$$
$$\dot{m}_f = \frac{Q}{h_{fg}} = \frac{125}{165.3} = 0.7562 \quad kg/s$$
$$\dot{m}_a = \frac{Q}{c_{p,a}\left(T_{c2} - T_{c1}\right)} = \frac{125}{1.009 \times (26-18)} = 15.5 \quad kg/s$$

Assume Re for air flow to be 7000, i.e. Re = $\frac{G_a D_{h,a}}{\mu_a} = 7000$.

Mass velocity:

$$G_a = \frac{Re \cdot \mu_a}{D_{h,a}} = \frac{7000 \times 2.02 \times 10^{-5}}{0.00348} = 40.63 \ \ kg/(m^2.s)$$

Air side heat transfer coefficient:

From Fig. 10.4, for Re = 7000, we can obtain:

$$\frac{h}{Gc_p} Pr^{2/3} = 0.0048$$
$$f = 0.02$$
$$h_a = 0.0048 \times \frac{G_a c_{p,a}}{Pr_a^{2/3}} = 0.0048 \times \frac{40.63 \times 1009}{0.71^{2/3}} = 247.25 \ \ W/m^2.K$$

Fin efficiency:

Fin length (from fig. 10.4):

$$L = \frac{1 \times 0.0254 - 0.01717}{2} = 0.004115 \ \ m$$

$$mL = \sqrt{\frac{Ph_a}{k_{al}A}} \cdot L \approx \sqrt{\frac{2h_a}{k_{al} \cdot t}} \cdot L = \sqrt{\frac{2 \times 247.25}{250 \times 0.41 \times 10^{-3}}} \times 0.004115 = 0.2858$$

$$\eta_f = \frac{\tanh(mL)}{mL} = 0.97$$

The total efficiency:

The ratio of finned area/ total surface area:

$$A_u = \left(1 \times 0.0254 - 7.75 \times 0.41 \times 10^{-3}\right) \times \pi \times 0.01717 = 0.0011987071$$

$$A_f = 7.75 \times 2 \times \left(0.866 \times 1.0 \times 0.0254^2 - \frac{\pi \times 0.01717^2}{4}\right) = 0.0050710759$$

$$\frac{A_f}{A_t} = \frac{A_f}{A_f + A_u} = 0.809$$

$$\eta = \left[1 - \frac{A_f}{A_t}(1 - \eta_f)\right] = \left[1 - 0.809 \times (1 - 0.97)\right] = 0.98$$

Air side frontal area:

$$G_a = \frac{\dot{m}_a}{A_{free}} = \frac{\dot{m}_a}{\sigma \cdot A_{fr}}$$

$$A_{fr} = \frac{\dot{m}_a}{\sigma \cdot G_a} = \frac{15.5}{0.481 \times 40.63} = 0.7931 \ \ m^2$$

Water side heat transfer coefficient:

Assume: (1). tube wall thickness to be t = 0.002 m, so the I.D. of tube is d = 0.01317 m.

(2). average quality x = 0.5.

(3). condensate velocity inside tube u_l = 0.05 m/s,

so the mass flux G = $\rho_l u_l$ = 1156 × 0.05 = 57.8 kg/(m^2.s).

Then,

$$Re_l = \frac{G \cdot (1 - x) \cdot d}{\mu_l} = \frac{57.8 \times (1 - 0.5) \times 0.01317}{1.89 \times 10^{-4}} = 2014$$

$$Re_g = \frac{G \cdot x \cdot d}{\mu_g} = \frac{57.8 \times (1 - 0.5) \times 0.01317}{0.125 \times 10^{-4}} = 30449$$

$$Re_{eq} = Re_g \left(\frac{\mu_g}{\mu_l}\right)\left(\frac{\rho_l}{\rho_g}\right)^{0.5} + Re_l = 30449 \times \left(\frac{0.125}{1.89}\right) \times \left(\frac{1156}{45.45}\right)^{0.5} + 2014 = 12170$$

$$h_{TP} = 0.05 Re_{eq}^{0.8} Pr_l^{0.33} \frac{k_l}{d} = 0.05 \times 12170^{0.8} \times 3.98^{0.33} \times \frac{0.071}{0.01317} = 788.6 \ \ W/(m^2.K)$$

Overall heat transfer coefficient:

$$U_a = \left[\frac{A_o}{A_i}\frac{1}{h_i} + A_o R_w + \frac{1}{\eta h_o}\right]^{-1}$$

$$= \left[\frac{554}{74} \times \frac{1}{788.6} + \frac{1.717 \times 10^{-2}}{2 \times 250} \ln\left(\frac{1.717}{1.317}\right) + \frac{1}{0.98 \times 247.25} \right]^{-1}$$

$$= 73.4 \ \ W/(m^2.K)$$

Heat balance:

$$C_{min} = C_{air} = \dot{m}_a c_{p,a} = 15.5 \times 1009 = 15639.5 \ \ W/K$$

$$\varepsilon = \frac{26-18}{(310-273.15)-18} = 0.4244$$

$$\frac{C_{min}}{C_{max}} = \frac{15639.5}{\infty} = 0$$

From Fig. 2.15: NTU = 0.5

$$A_a = \frac{NTU \cdot C_{min}}{U_a} = \frac{0.5 \times 15639.5}{73.4} = 106.5 \ \ m^2$$

Heat exchanger size:

$$V = \frac{A_a}{554} = \frac{106.5}{554} = 0.192 \ \ m^3$$

Core length L_1:

$$L_1 = \frac{V}{A_{fr,a}} = \frac{0.192}{0.7931} = 0.242 \ \ m$$

Core length L_2:

$$G = \frac{\dot{m}_f}{A_{fluid}} = \rho_f u_f$$

$$A_{fluid} = \frac{\dot{m}_f}{\rho_f u_f} = \frac{0.7562}{1156 \times 0.05} = 0.013 \ \ m^2$$

$$A_{fr,f} = \frac{A_{fluid}}{\sigma_1} = \frac{0.013}{0.244} = 0.053 \ \ m^2$$

$$L_2 = \frac{V}{A_{fr,w}} = \frac{0.192}{0.053} = 3.6 \ \ m$$

$$L_3 = \frac{V}{L_1 L_2} = \frac{0.192}{0.242 \times 3.6} = 0.22 \ \ m$$

Problem 10.12

Design an air-water compact heat exchanger to serve as an intercooler for a gas turbine plant. The heat exchanger is to meet the following heat transfer and pressure drop performance specifications:

flow rate: 25 kg/s
inlet temperature: 500°C
outlet temperature: 350°C
inlet pressure: 2×10^5 N/m^2
pressure drop rate: 8%
water side operating conditions: flow rate 50 kg/s
inlet temperature: 290 K

GIVEN:

The heat exchanger surface for this heat exchanger is given in Fig. 10.8 (surface 9.1-0.737-S) with flattened tube-fin compact surface. Fins are continuous aluminum fins. The geometrical data for air side is given in Table 10.1. Water side: the flatted tube is 0.2 cm×1.6 cm. The inside diameter of the tubes before it was flatted was 1.23 cm, with a wall thickness of 0.025 cm. Water velocity inside is 1.5 m/s. The design should specify the core size, the core pressure drop.

Hot Air flows across a plain plate-fin matrix configuration (9.1-0.737-S):

$p_1 = 2\times10^5$ N/m^2, $T_{in,a}$ = 500 °C, $T_{out,\,a}$ = 350 °C, $\dot{m}_a$ = 25 kg/s

Δp = 8%

Water side:

$T_{in,w}$ = 290 K, $\dot{m}_w$ = 50 kg/s

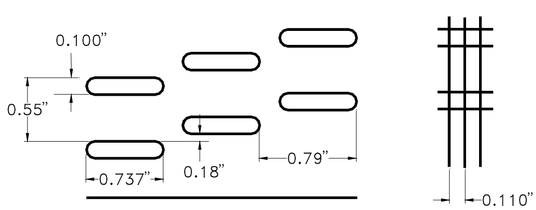
Geometrical size:

Water side:

Flatted tube size: 0.2×1.6 cm^2, Diameter of tube before flatted: d_t = 1.23 cm,

Tube wall thickness: δ = 0.025 cm, Water velocity u_w = 1.5 m/s

SOLUTION:



Surface 9.1–0.737–S

Fins per cm = 23.114
D_h = 0.3565 cm
Fin thickness t = 0.0102 cm
Extended area/Total area: 0.813
Area/Core volume β = 734.9 m^2/m^3
Free flow area/ frontal area σ = 0.788

Water side:

Frontal area associated with one tube: 0.79”×0.55”=0.4345 in^2 = 2.803×10^{-4} m^2

$$\sigma_w = \frac{A_t}{2.803\times10^{-4}} = \frac{\left[(1.6-0.2)\times0.2+(\pi/4)\times0.2^2\right]\times10^{-4}}{2.803\times10^{-4}} = 0.111$$

$$\textit{Heat transfer area/Volume} = \frac{\pi\times0.0123}{2.803\times10^{-4}} = 138$$

At $\frac{500+350}{2} = 425\,^{o}C$, $p_1 = 2\times10^5\ N/m^2$, properties of air from Appendix are:

$$\rho = 1.012\ \ kg/m^3$$
$$\mu = 3.36\times10^{-5}\ \ kg/m.s$$
$$c_p = 1074\ \ J/kg.K$$
$$Pr = 0.682$$

From Table 10.1, we have

$$\frac{A_{min}}{A_{fr}} = \sigma = 0.788$$
$$D_h = 0.3565\,cm$$
$$\beta = 734.9\,m^2/m^3$$

Properties of water from Appendix are (Assume mean temperature as being 300K):

$$\rho_w = 996\ \ kg/m^3$$
$$\mu_w = 8.56\times10^{-4}\ \ kg/m.s$$
$$c_{p,w} = 4179\ \ J/kg.K$$
$$Pr_w = 5.88$$
$$k_w = 0.609\ \ W/(m.K)$$

Thermal conductivity of aluminum is taken as k_{al} = 250 W/(m.K)

The trial-and-error method is needed for air side calculation, for the first attempting, let Re for air flow be 7000, i.e. Re = $\frac{G_a D_{h,a}}{\mu_a} = 7000$

<u>Mass velocity:</u>

$$G_a = \frac{Re\cdot\mu_a}{D_{h,a}} = \frac{7000\times3.36\times10^{-5}}{0.003565} = 65.97\ \ kg/(m^2.s)$$

<u>Air side heat transfer coefficient:</u>

From Fig. 10.8 and Fig. 10.7, for Re = 7000, we can obtain:

$$\frac{h}{Gc_p}Pr^{2/3} = 0.0044$$
$$f = 0.015$$
$$h_a = 0.0044\times\frac{G_a c_{p,a}}{Pr_a^{\,2/3}} = 0.0044\times\frac{65.97\times1074}{0.682^{2/3}} = 402.36\ \ W/m^2.K$$

<u>Fin efficiency:</u>

$$mL = \sqrt{\frac{Ph_a}{k_{al}A}}\cdot L \approx \sqrt{\frac{2h_a}{k_{al}\cdot t}}\cdot L = \sqrt{\frac{2\times402.36}{250\times0.0102\times10^{-2}}}\times0.5715\times10^{-2} = 1.015$$

$$\eta_f = \frac{\tanh(mL)}{mL} = 0.76$$

The total efficiency is then:

$$\eta = \left[1 - \frac{A_f}{A}(1-\eta_f)\right] = \left[1 - 0.813(1-0.76)\right] = 0.80$$

Air side frontal area:

$$G_a = \frac{\dot{m}_a}{A_{free}} = \frac{\dot{m}_a}{\sigma \cdot A_{fr}}$$

$$A_{fr} = \frac{\dot{m}_a}{\sigma \cdot G_a} = \frac{25}{0.788 \times 65.97} = 0.4809 \ \ m^2$$

Water side heat transfer coefficient:

$$\mathrm{Re}_w = \frac{\rho_w u_w D_{h,w}}{\mu_w}$$

$$D_{h,w} = \frac{4A}{P_w} = \frac{4 \times \left[\frac{\pi \times 0.002^2}{4} + (1.6-0.2) \times 0.2 \times 10^{-4}\right]}{\pi \times 0.0123} = 0.003224 \ \ m$$

$$\mathrm{Re}_w = \frac{996 \times 1.5 \times 0.003224}{8.54 \times 10^{-4}} = 5640$$

Using Gnielinski's correlation for 2300<Re<10000, we obtain:

$$f = (1.58 \ln \mathrm{Re} - 3.28)^{-2} = 0.0093$$

$$\mathrm{Nu} = \frac{(f/2)(\mathrm{Re}-1000)\,\mathrm{Pr}}{1+12.7(f/2)^{0.5}(\mathrm{Pr}^{2/3}-1)} = \frac{(0.0093/2) \times (5640-1000) \times 5.88}{1+12.7 \times (0.0093/2)^{0.5}(5.88^{2/3}-1)} = 42.9$$

$$h_w = \frac{k \cdot \mathrm{Nu}}{D_{h,w}} = \frac{0.609 \times 42.9}{0.003224} = 8104 \ \ W/(m^2.K)$$

Overall heat transfer coefficient:

$$U_a = \left[\frac{A_o}{A_i}\frac{1}{h_i} + A_o R_w + \frac{1}{\eta h_o}\right]^{-1}$$

$$= \left[\frac{734.9}{138} \times \frac{1}{8104} + \frac{1.28 \times 10^{-2}}{2 \times 250} \ln\left(\frac{12.8}{1.23}\right) + \frac{1}{0.8 \times 402.36}\right]^{-1}$$

$$= 265.6 \ \ W/(m^2.K)$$

Heat balance:

$$T_{out,w} = 290 + \frac{25 \times 1074}{50 \times 4179} \times (500-350) = 309 \ \ K$$

$$\varepsilon = \frac{500-350}{500+273-290} = 0.31$$

$$\frac{C_{min}}{C_{max}} = \frac{20 \times 1074}{50 \times 4179} = 0.13$$

From Fig. 2.15: NTU = 0.4

$$A_a = \frac{NTU \cdot C_{min}}{U_a} = \frac{0.4 \times (25 \times 1074)}{265.6} = 40.44 \ \ m^2$$

Heat exchanger size:

$$V = \frac{A_a}{734.9} = \frac{40.44}{734.9} = 0.055 \ \ m^3$$

Core length L_1:

$$L_1 = \frac{V}{A_{fr,a}} = \frac{0.055}{0.4809} = 0.114 \ \ m$$

Core length L_2:

$$G_w = \frac{\dot{m}_w}{A_w} = \rho_w u_w$$

$$A_w = \frac{\dot{m}_w}{\rho_w u_w} = \frac{50}{996 \times 1.5} = 0.03347 \quad m^2$$

$$A_{fr,w} = \frac{A_w}{\sigma_w} = \frac{0.03347}{0.111} = 0.302 \ \ m^2$$

$$L_2 = \frac{V}{A_{fr,w}} = \frac{0.055}{0.302} = 0.182 \ \ m$$

$$L_3 = \frac{V}{L_1 L_2} = \frac{0.055}{0.114 \times 0.182} = 2.65 \ \ m$$

Pressure drop:

$$\frac{\Delta p_f}{p_1} = \frac{G^2}{2\rho_a p_1}\left[(1+\sigma^2)\left(\frac{\rho_1}{\rho_2} - 1\right) + f\frac{A_t}{A_{min}}\left(\frac{\rho_1}{\rho_m}\right)\right]$$

To calculate ρ_1, ρ_2 from p_1, a pressure drop assumption must be made and, then, verified. A pressure drop of 2% will be used as the initial guess:

$$p_1 = 1.02 p_2,$$

$$p_2 = \frac{p_1}{1.02} = 1.96 \times 10^5 \, Pa, \quad \text{and } p_m = \frac{p_1 + p_2}{2} = 1.98 \times 10^5 \quad Pa$$

$$\frac{\rho_1}{\rho_2} = \frac{p_1}{p_2} = 1.02$$

$$\frac{\rho_1}{\rho_m} = \frac{p_1}{p_m} = 1.01$$

$$\frac{A_t}{A_{min}} = \frac{4L}{D_h} = \frac{4 \times 0.114}{0.003565} = 128$$

Then

$$\frac{\Delta p_f}{p_1} = \frac{G^2}{2\rho_a p_1}\left[(1+\sigma^2)\left(\frac{\rho_1}{\rho_2} - 1\right) + f\frac{A_t}{A_{min}}\left(\frac{\rho_1}{\rho_m}\right)\right]$$

$$= \frac{65.97^2}{2 \times 2 \times 10^5 \times 0.996}\left[\left(1 + 0.788^2\right)(1.02 - 1) + 0.015 \times 128 \times 1.01\right] = 0.0215$$

The final result is almost the same as we assume, so $\frac{\Delta p_f}{p_1} = 2\%$. The pressure drop 2% is less than the pressure drop limitation 8%, and the design is acceptable.

Problem 10.13

Design a heat exchanger asking the surface given in Figure 10.8 (surface 9.29-0.737-S-R). Fins are continuous aluminum. The geometrical data for the air side are given in Table 10.1. On the water side, the flatted tube is 0.2 cm × 1.6 cm. The inside diameter of the tube before it was flattened was 1.23 cm, with a wall thickness of 0.025 cm. Water velocity inside is 1.5 m/s. The design should specify the core size and the core pressure drop.

GIVEN:

The heat exchanger surface for this heat exchanger is given in Fig. 10.8 (surface 9.1-0.737-S) with flattened tube-fin compact surface. Fins are continuous aluminum fins. The geometrical data for air side is given in Table 10.1. Water side: the flatted tube is 0.2 cm×1.6 cm. The inside diameter of the tubes before it was flatted was 1.23 cm, with a wall thickness of 0.025 cm. Water velocity inside is 1.5 m/s. The design should specify the core size, the core pressure drop.

Hot Air flows across a plain plate-fin matrix configuration (9.1-0.737-S):

$p_1 = 2\times10^5$ N/m^2, $T_{in,a}$ = 500 °C, $T_{out,\,a}$ = 350 °C, $\dot{m}_a$ = 25 kg/s

Δp = 8%

Water side:

$T_{in,w}$ = 290 K, $\dot{m}_w$ = 50 kg/s

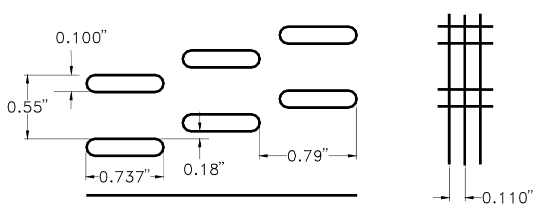
Geometrical size:

Water side:

Flatted tube size: 0.2×1.6 cm^2, Diameter of tube before flatted: d_t = 1.23 cm,

Tube wall thickness: δ = 0.025 cm, Water velocity u_w = 1.5 m/s

SOLUTION:



Fins per cm = 23.114
D_h = 0.3565 cm
Fin thickness t = 0.0102 cm
Extended area/Total area: 0.813
Area/Core volume β = 734.9 m^2/m^3
Free flow area/ frontal area σ = 0.788

Water side:

Frontal area associated with one tube: 0.79"×0.55"=0.4345 in^2 = 2.803×10^{-4} m^2

$$\sigma_w = \frac{A_t}{2.803\times10^{-4}} = \frac{\left[(1.6-0.2)\times0.2+(\pi/4)\times0.2^2\right]\times10^{-4}}{2.803\times10^{-4}} = 0.111$$

$$\textit{Heat transfer area/Volume} = \frac{\pi\times0.0123}{2.803\times10^{-4}} = 138$$

At $\frac{500+350}{2}=425\,^{o}$C, $p_1=2\times10^5$ N/m^2, properties of air from Appendix are:

$$\rho = 1.012 \ \ \text{kg}/\text{m}^3$$
$$\mu = 3.36\times10^{-5} \ \ \text{kg}/\text{m.s}$$
$$c_p = 1074 \ \ \text{J}/\text{kg.K}$$
$$\text{Pr} = 0.682$$

From Table 10.1, we have

$$\frac{A_{min}}{A_{fr}} = \sigma = 0.788$$
$$D_h = 0.3565\,\text{cm}$$
$$\beta = 734.9\,\text{m}^2/m^3$$

Properties of water from Appendix are (Assume mean temperature as being 300K):

$$\rho_w = 996 \ \ \text{kg/m}^3$$
$$\mu_w = 8.56\times10^{-4} \ \ \text{kg/m.s}$$
$$c_{p,w} = 4179 \ \ \text{J/kg.K}$$
$$\text{Pr}_w = 5.88$$
$$k_w = 0.609 \ \ \text{W/(m.K)}$$

Thermal conductivity of aluminum is taken as k_{al} = 250 W/(m.K)

The trial-and-error method is needed for air side calculation, for the first attempting, let Re for air flow be 7000, i.e. Re = $\frac{G_a D_{h,a}}{\mu_a}=7000$

Mass velocity:

$$G_a = \frac{\text{Re}\cdot\mu_a}{D_{h,a}} = \frac{7000\times3.36\times10^{-5}}{0.003565} = 65.97 \ \ \text{kg}/(\text{m}^2.\text{s})$$

Air side heat transfer coefficient:

From Fig. 10.8 and Fig. 10.7, for Re = 7000, we can obtain:

$$\frac{h}{Gc_p}\text{Pr}^{2/3} = 0.0044$$
$$f = 0.015$$
$$h_a = 0.0044\times\frac{G_a c_{p,a}}{\text{Pr}_a^{\ 2/3}} = 0.0044\times\frac{65.97\times1074}{0.682^{2/3}} = 402.36 \ \ \text{W}/\text{m}^2.\text{K}$$

Fin efficiency:

$$mL = \sqrt{\frac{Ph_a}{k_{al}A}}\cdot L \approx \sqrt{\frac{2h_a}{k_{al}\cdot t}}\cdot L = \sqrt{\frac{2\times402.36}{250\times0.0102\times10^{-2}}}\times0.5715\times10^{-2} = 1.015$$

$$\eta_f = \frac{\tanh(mL)}{mL} = 0.76$$

The total efficiency is then:

$$\eta = \left[1 - \frac{A_f}{A}(1 - \eta_f)\right] = [1 - 0.813(1 - 0.76)] = 0.80$$

Air side frontal area:

$$G_a = \frac{\dot{m}_a}{A_{free}} = \frac{\dot{m}_a}{\sigma \cdot A_{fr}}$$

$$A_{fr} = \frac{\dot{m}_a}{\sigma \cdot G_a} = \frac{25}{0.788 \times 65.97} = 0.4809 \ \ m^2$$

Water side heat transfer coefficient:

$$Re_w = \frac{\rho_w u_w D_{h,w}}{\mu_w}$$

$$D_{h,w} = \frac{4A}{P_w} = \frac{4 \times \left[\frac{\pi \times 0.002^2}{4} + (1.6 - 0.2) \times 0.2 \times 10^{-4}\right]}{\pi \times 0.0123} = 0.003224 \ \ m$$

$$Re_w = \frac{996 \times 1.5 \times 0.003224}{8.54 \times 10^{-4}} = 5640$$

Using Gnielinski's correlation for 2300<Re<10000, we obtain:

$$f = (1.58 \ln Re - 3.28)^{-2} = 0.0093$$

$$Nu = \frac{(f/2)(Re - 1000)\, Pr}{1 + 12.7(f/2)^{0.5}(Pr^{2/3} - 1)} = \frac{(0.0093/2) \times (5640 - 1000) \times 5.88}{1 + 12.7 \times (0.0093/2)^{0.5}(5.88^{2/3} - 1)} = 42.9$$

$$h_w = \frac{k \cdot Nu}{D_{h,w}} = \frac{0.609 \times 42.9}{0.003224} = 8104 \ \ W/(m^2.K)$$

Overall heat transfer coefficient:

$$U_a = \left[\frac{A_o}{A_i}\frac{1}{h_i} + A_o R_w + \frac{1}{\eta h_o}\right]^{-1}$$

$$= \left[\frac{734.9}{138} \times \frac{1}{8104} + \frac{1.28 \times 10^{-2}}{2 \times 250}\ln\left(\frac{12.8}{1.23}\right) + \frac{1}{0.8 \times 402.36}\right]^{-1}$$

$$= 265.6 \ \ W/(m^2.K)$$

Heat balance:

$$T_{out,w} = 290 + \frac{25 \times 1074}{50 \times 4179} \times (500 - 350) = 309 \ \ K$$

$$\varepsilon = \frac{500 - 350}{500 + 273 - 290} = 0.31$$

$$\frac{C_{min}}{C_{max}} = \frac{20 \times 1074}{50 \times 4179} = 0.13$$

From Fig. 2.15: NTU = 0.4

$$A_a = \frac{NTU \cdot C_{min}}{U_a} = \frac{0.4 \times (25 \times 1074)}{265.6} = 40.44 \ \ m^2$$

Heat exchanger size:

$$V = \frac{A_a}{734.9} = \frac{40.44}{734.9} = 0.055 \ \text{m}^3$$

Core length L_1:

$$L_1 = \frac{V}{A_{fr,a}} = \frac{0.055}{0.4809} = 0.114 \ \text{m}$$

Core length L_2:

$$G_w = \frac{\dot{m}_w}{A_w} = \rho_w u_w$$

$$A_w = \frac{\dot{m}_w}{\rho_w u_w} = \frac{50}{996 \times 1.5} = 0.03347 \quad \text{m}^2$$

$$A_{fr,w} = \frac{A_w}{\sigma_w} = \frac{0.03347}{0.111} = 0.302 \ \text{m}^2$$

$$L_2 = \frac{V}{A_{fr,w}} = \frac{0.055}{0.302} = 0.182 \ \text{m}$$

$$L_3 = \frac{V}{L_1 L_2} = \frac{0.055}{0.114 \times 0.182} = 2.65 \ \text{m}$$

Pressure drop:

$$\frac{\Delta p_f}{p_1} = \frac{G^2}{2\rho_a p_1}\left[(1+\sigma^2)\left(\frac{\rho_1}{\rho_2} - 1\right) + f\frac{A_t}{A_{\min}}\left(\frac{\rho_1}{\rho_m}\right)\right]$$

To calculate ρ_1, ρ_2 from p_1, a pressure drop assumption must be made and, then, verified. A pressure drop of 2% will be used as the initial guess:

$$p_1 = 1.02 p_2,$$

$$p_2 = \frac{p_1}{1.02} = 1.96 \times 10^5 \text{ Pa}, \quad \text{and } p_m = \frac{p_1 + p_2}{2} = 1.98 \times 10^5 \ \text{Pa}$$

$$\frac{\rho_1}{\rho_2} = \frac{p_1}{p_2} = 1.02$$

$$\frac{\rho_1}{\rho_m} = \frac{p_1}{p_m} = 1.01$$

$$\frac{A_t}{A_{\min}} = \frac{4L}{D_h} = \frac{4 \times 0.114}{0.003565} = 128$$

Then

$$\frac{\Delta p_f}{p_1} = \frac{G^2}{2\rho_a p_1}\left[(1+\sigma^2)\left(\frac{\rho_1}{\rho_2} - 1\right) + f\frac{A_t}{A_{\min}}\left(\frac{\rho_1}{\rho_m}\right)\right]$$

$$= \frac{65.97^2}{2 \times 2 \times 10^5 \times 0.996}\left[\left(1 + 0.788^2\right)(1.02 - 1) + 0.015 \times 128 \times 1.01\right] = 0.0215$$

The final result is almost the same as we assume, so $\frac{\Delta p_f}{p_1} = 2\%$. The pressure drop 2% is less than the pressure drop limitation 8%, and the design is acceptable.

Problem 11.1

The following constructional information is available for a gasket-plate heat exchanger:

chevron angle	**50 º**
Enlargement factor	**1.17**
All port diameters	**15 cm**
Plate thickness	**0.0006 m**
Vertical port distance	**1.50 m**
Horizontal port distance	**0.50 m**
Plate pitch	**0.0035 m**

Calculate:

a. Mean channel flow gap

b. One-channel flow area

c. Channel equivalent diameter

d. Projected plate area

e. Effective surface area per plate

SOLUTION:

a. Mean channel flow gap:

$$b = p - t = 0.0035 - 0.0006 = 0.0029 \quad \text{m}$$

b. One-channel flow area:

$$A_{1c} = b \cdot L_w = b \cdot \left(L_h + D_p\right) = 0.0029 \times (0.5 + 0.15) = 0.001885 \ \ \text{m}^2$$

c. Channel equivalent diameter:

$$D_e = \frac{4\left(b \cdot L_w\right)}{2\left(b + L_w \phi\right)} = \frac{2b}{\phi} = \frac{2 \times 0.0029}{1.17} = 0.004957 \quad \text{m}$$

d. Projected plate area:

$$A_{1p} = L_p \cdot L_w = \left(L_h + D_p\right)\left(L_v - D_p\right) = (0.5 + 0.15)(1.5 - 0.15) = 0.8775 \ \ \text{m}^2$$

e. Effective surface area per plate:

$$A_1 = \phi \cdot A_{1p} = 1.17 \times 0.8775 = 1.0267 \ \ \text{m}^2$$

Problem 11.2

A gasketed-plate heat exchanger will be used for heating city water (R_{fc} = 0.00006 m^2.K/W) using the wastewater available at 90 °C. The vertical distance between the ports of the plate is 1.60 m and the width of the plate is 0.50 m with a gap of 6 mm between the plates. The enhancement factor is provided by the manufacturer as 1.17 and the Chevron angle is 50 °C. The plates are made of titanium (k = 20 W/m.K) with a thickness of 0.0006m. The port diameter is 0.15 m. The cold water enters to the plate heat exchanger at 15 °C and leaves at 45 °C at a rate of 6 kg/s; and it will be heated by the hot water available at 90 °C, flowing at a rate of 12 kg/s. By considering single-pass arrangements for both streams, calculate:
a. The effective surface area and the number of plates of this heat exchangers
b. The pressure drop for both streams

GIVEN:

-Inlet temperature of hot wastewater (T_{h1}) = 90 °C
-Mass flow rate of hot water ($\dot{m}_h$) = 12 kg/s
-Inlet temperature of cold water (T_{c1}) = 15 °C
-Outlet temperature of cold water (T_{c2}) = 45 °C
-Mass flow rate of cold water ($\dot{m}_c$) = 6 kg/s
-Fouling resistance (R_{fc}) = 0.00006 m^2.K/W
-Plate thermal conductivity (Titanium , k_w) = 20 W/(m.K)
-Gasket heat exchanger configuration:

L_v = 1.60 m	L_w = 0.50 m
b = 6 mm	ϕ = 1.17
β = 50°	t = 0.0006 m
D_p = 0.15 m	

SOLUTION:

Cold water properties at $T_m = \dfrac{T_{in} + T_{out}}{2} = \dfrac{15+45}{2} = 30$ °C:

$$\rho = 996 \ \text{kg/m}^3$$
$$\mu = 8.15\times 10^{-4} \ \text{kg/m.s}$$
$$c_p = 4179 \ \text{J/kg.K}$$
$$\text{Pr} = 5.58$$
$$k = 0.612 \ \text{W/(m.K)}$$

Hot water properties at T_{h1} = 90 °C:

$$\rho = 965 \ \text{kg/m}^3$$
$$\mu = 3.16\times 10^{-4} \ \text{kg/m.s}$$
$$c_p = 4205 \ \text{J/kg.K}$$
$$\text{Pr} = 1.96$$
$$k = 0.675 \ \text{W/(m.K)}$$

a.

Heat duty:

$$Q = (\dot{m}c_p)_c (T_{c2} - T_{c1}) = (\dot{m}c_p)_h (T_{h1} - T_{h2})$$
$$= 6 \times 4179 \times (45 - 15) = 752220 \text{ W}$$
$$T_{h2} = T_{h1} - \frac{(\dot{m}c_p)_c (T_{c2} - T_{c1})}{(\dot{m}c_p)_h} = 90 - \frac{6 \times 4179 \times (45 - 15)}{12 \times 4205} = 75 \ ^oC$$

Log-mean temperature difference:

$$\Delta T_{lm,cf} = \frac{(T_{h1} - T_{c2}) - (T_{h2} - T_{c1})}{\ln \frac{(T_{h1} - T_{c2})}{(T_{h2} - T_{c1})}} = \frac{(90 - 45) - (75 - 15)}{\ln \frac{90 - 45}{75 - 15}} = 52.14 \ ^oC$$

To solve for the number of plates needed, a trial-and-error method has to be used. First, an estimated heat transfer coefficient is assumed, which is verified and iterated later until the assumed value matches the value resulted from it.

Assume the overall heat transfer coefficient U_c to be 5000 W/(m^2.K) (From table 11.2).

For safety factor C_s = 1.4,

$$Q_f = C_s \cdot Q_r = 1.4 \times 752220 = 1053108 \text{ W}$$

where $Q_f = U_f \cdot A_e F \Delta T_{lm,cf}$, then:

$$A_e = \frac{Q_f}{U_f \cdot F \Delta T_{lm,cf}} = \frac{1053108}{5000 \times 1 \times 52.14} = 4.0395 \text{ m}^2$$

The single-plate projected area:

$$A_{1p} = L_w \cdot L_p = (L_v - D_p) \cdot L_w = (1.6 - 0.15)(0.50) = 0.725 \text{ m}^2$$

The single-plate heat transfer area:

$$A_1 = A_{1p} \cdot \phi = 0.725 \times 1.17 = 0.8483 \text{ m}^2$$

The number of plates:

$$N_e = \frac{A_e}{A_1} = \frac{4.0395}{0.8483} \approx 5 \text{ plates}$$

$$\therefore N_t = N_e + 2 = 5 + 2 = 7 \text{ plates}$$

Verify the assumed heat transfer coefficient:

One channel flow area:

$$A_{ch} = b \times L_w = 0.006 \times 0.5 = 0.003 \text{ m}^2$$

Channel equivalent diameter:

$$D_e = \frac{4(b \cdot L_w)}{2(b + L_w \phi)} = \frac{2b}{\phi} = \frac{2 \times 0.006}{1.17} = 0.01026 \text{ m}$$

The number of channels per pass, N_{cp}:

$$N_{cp} = \frac{N_t - 1}{2 \cdot N_p} = \frac{7 - 1}{2 \times 1} = 3$$

Mass flow rate per channel:

$$\dot{m}_{ch,h} = \frac{12}{3} = 4 \quad kg/s$$

$$\dot{m}_{ch,c} = \frac{6}{3} = 2 \quad kg/s$$

Mass velocity, G_{ch}:

$$G_{ch,h} = \frac{4}{0.003} = 1333.34 \quad kg/(m^2 s)$$

$$G_{ch,c} = \frac{2}{0.003} = 666.67 \quad kg/(m^2 s)$$

Hot and cold fluid Reynolds numbers for correlation proposed by Kumar are:

$$Re_h = \frac{G_{ch,h} \cdot D_e}{\mu_h} = \frac{1333.34 \times 0.01026}{3.16 \times 10^{-4}} = 43291$$

$$Re_c = \frac{G_{ch,c} \cdot D_e}{\mu_c} = \frac{666.67 \times 0.01026}{8.15 \times 10^{-4}} = 8393$$

Heat transfer coefficients, h_h—This can be obtained by referring to Table 11.6 C_h=0.13, and n= 0.732 for $\beta = 50\,^o$:

$$Nu_h = \frac{h_h D_e}{k} = 0.13 \cdot (Re_h)^{0.732} (Pr)^{1/3} \left(\frac{\mu_b}{\mu_w}\right)^{0.17}$$

By assuming $\mu_b = \mu_w$, then:

$$Nu_h = 0.13 \cdot (43291)^{0.732} (1.96)^{1/3} = 402.9$$

$$h_h = \frac{Nu_h k}{D_e} = \frac{402.9 \times 0.675}{0.01026} = 26507 \;\; W/(m^2 \cdot K)$$

$$Nu_c = 0.13 \cdot (8393)^{0.732} (5.58)^{1/3} = 171.8$$

$$h_c = \frac{Nu_c k}{D_e} = \frac{171.8 \times 0.612}{0.01026} = 10247.7 \;\; W/(m^2 \cdot K)$$

Overall heat transfer coefficient:

$$\frac{1}{U_c} = \frac{1}{h_c} + \frac{1}{h_h} + \frac{0.0006}{20}$$

$$U_c = \left[\frac{1}{h_c} + \frac{1}{h_h} + \frac{0.0006}{20}\right]^{-1} = \left[\frac{1}{10247.7} + \frac{1}{26507} + \frac{0.0006}{20}\right]^{-1} = 6049.3 \;\; W/(m^2.K)$$

Fouled overall heat transfer coefficient:

$$\frac{1}{U_f} = \frac{1}{U_c} + 0.00006$$

$$U_f = \left[\frac{1}{U_c} + 0.00006\right]^{-1} = \left[\frac{1}{6049.3} + 0.0006\right]^{-1} = 4438.4 \;\; W/(m^2.K)$$

So, the resulting heat transfer coeffcient is less than the assumed 5000 W/(m^2.K), the procedure has to be repeated, assuming the U_f to be 4438.4 W/(m^2.K) as follows:

$$A_e = \frac{Q_f}{U_f \cdot F\Delta T_{lm,cf}} = \frac{1053108}{4438.4 \times 1 \times 52.14} = 4.5506 \ \text{m}^2$$

The number of plates:

$$N_e = \frac{A_e}{A_1} = \frac{4.5506}{0.8483} = 5.36 \approx 7$$ plates (round off to odd number to make equal number of channel passes for cold and hot fluid)

So the number of plates we get is:

$$\therefore \ N_t = N_e + 2 = 7 + 2 = 9 \text{ plates}$$

and the effective surface area is:

$$A_e = N_e \cdot A_1 = (9-2) \times 0.8483 = 5.9381 \ \text{m}^2$$

b.

Hot and cold fluid friction coefficients: —This can be obtained by referring to Table 11.6 K_p=0.772, and m= 0.161 for $\beta = 50^o$ and Re > 300:

$$f_h = \frac{0.772}{(Re_h)^{0.161}} = \frac{0.772}{43291^{0.161}} = 0.138$$

$$f_c = \frac{0.772}{(Re_c)^{0.161}} = \frac{0.772}{6456^{0.161}} = 0.180$$

Hot and cold fluid frictional pressure drop:

$$\Delta p_c = 4f \frac{L_{eff} N_p}{D_e} \frac{G_c^{\,2}}{2\rho} \left(\frac{\mu_b}{\mu_w}\right)^{-0.17}$$

$$(\Delta p_c)_h = 4 \times 0.138 \times \frac{1.6 \times 1}{0.01026} \times \frac{1333.34^2}{2 \times 965} = 79293 \ \text{Pa}$$

$$(\Delta p_c)_c = 4 \times 0.180 \times \frac{1.6 \times 1}{0.01026} \times \frac{666.67^2}{2 \times 996} = 25051 \ \text{Pa}$$

Port mass velocity:

$$G_{p,h} = \frac{\dot{m}}{\pi\left(\frac{D_p^{\,2}}{4}\right)} = \frac{12}{\pi \times \left(\frac{0.15^2}{4}\right)} = 679.06 \ \ \text{kg/(m}^2\text{s)}$$

$$G_{p,c} = \frac{\dot{m}_c}{\pi\left(\frac{D_p^{\,2}}{4}\right)} = \frac{6}{\pi \times \left(\frac{0.15^2}{4}\right)} = 339.53 \ \ \text{kg/(m}^2\text{s)}$$

Pressure drop in port ducts:

$$\Delta p_p = 1.4 N_p \frac{G_p^{\,2}}{2\rho}$$

$$(\Delta p_c)_h = 1.4 \times 1 \times \frac{679.06^2}{2 \times 965} = 334.49 \ \text{Pa}$$

$$\left(\Delta p_c\right)_c = 1.4 \times 1 \times \frac{339.53^2}{2 \times 996} = 81.02 \text{ Pa}$$

Total pressure drop for hot and cold fluids:

$$\Delta p_t = \Delta p_C + \Delta p_P$$

$$\left(\Delta p_t\right)_h = \left(\Delta p_c\right)_h + \left(\Delta p_p\right)_h = 79293 + 334.49 = 79627.5 \text{ Pa}$$

$$\left(\Delta p_t\right)_c = \left(\Delta p_c\right)_c + \left(\Delta p_p\right)_c = 25051 + 81.02 = 25132.02 \text{ Pa}$$

Problem 11.3

Solve problem 11.2 with the use of the correlations proposed by Kumar, and Mulay and Manglik. List the resutls in a Table and compare the results.

SOLUTION:

Hot fluid properties:

$$\rho = 986 \ \ \mathrm{kg/m^3}$$
$$\mu = 5.11\times 10^{-4} \ \ \mathrm{kg/m.s}$$
$$c_p = 4183 \ \ \mathrm{J/kg.K}$$
$$\mathrm{Pr} = 3.32$$
$$k = 0.635 \ \ \mathrm{W/(m.K)}$$

Cold water properties:

$$\rho = 995 \ \ \mathrm{kg/m^3}$$
$$\mu = 7.68\times 10^{-4} \ \ \mathrm{kg/m.s}$$
$$c_p = 4178 \ \ \mathrm{J/kg.K}$$
$$\mathrm{Pr} = 5.21$$
$$k = 0.616 \ \ \mathrm{W/(m.K)}$$

Stepwise performance analysis:

Heat duty:

$$Q_{rc} = \left(\dot{m}c_p\right)_c\left(T_{c2} - T_{c1}\right) = 140\times 4178\times\left(42-22\right) = 11698.4 \ \ \mathrm{W}$$

$$Q_{rh} = \left(\dot{m}c_p\right)_h\left(T_{h1} - T_{h2}\right) = 140\times 4183\times\left(65-45\right) = 11712.4 \ \ \mathrm{W}$$

Log-mean temperature difference:

$$\Delta T_1 = \Delta T_2 = \Delta T_{lm,cf} = 23 \ {}^{o}\mathrm{C}$$

The effective number of plate is:

$$N_e = N_t - 2 = 103$$

Plate pitch:

$$p = \frac{L_c}{N_t} = \frac{0.38}{105} = 0.0036 \ \ \mathrm{m}$$

Mean channel flow gap:

$$b = 0.0036 - 0.0006 = 0.003 \ \ \mathrm{m}$$

One channel flow area:

$$A_{ch} = b\times L_w = 0.003\times 0.63 = 0.00189 \ \ \mathrm{m^2}$$

The single-plate heat transfer area:

$$A_1 = \frac{A_e}{N_e} = \frac{110}{103} = 1.067 \ \ \mathrm{m^2}$$

Enlargement factor:

$$\phi = \frac{1.067}{0.85} = 1.255$$

Channel equivalent diameter:

$$D_e = \frac{4(b \cdot L_w)}{2(b + L_w \phi)} = \frac{2b}{\phi} = \frac{2 \times 0.003}{1.255} = 0.00478 \quad \text{m}$$

The number of channel per pass:

$$N_{cp} = \frac{N_t - 1}{2N_p} = \frac{105 - 1}{2 \times 1} = 52$$

Mass flow rate per channel:

$$\dot{m}_{ch} = \frac{140}{52} = 2.69 \quad \text{kg/s}$$

Mass velocity, G_{ch}:

$$G_{ch} = \frac{2.69}{0.00189} = 1423.3 \quad \text{kg/(m}^2\text{s)}$$

Hot and cold fluid Reynolds numbers are:

$$Re_h = \frac{G_{ch} \cdot D_e}{\mu_h} = \frac{1423.3 \times 0.00478}{5.11 \times 10^{-4}} = 13314$$

$$Re_c = \frac{G_{ch} \cdot D_e}{\mu_c} = \frac{1423.3 \times 0.00478}{7.68 \times 10^{-4}} = 8859$$

For β = 0.60:

Hot fluid heat transfer coefficient, h_h—This can be obtained by referring to Table 11.6 C_h=0.108, and n= 0.703 for β = 60 °C:

$$Nu_h = \frac{h_h D_e}{k} = 0.108 \cdot (Re_h)^{0.703} (Pr)^{1/3} \left(\frac{\mu_b}{\mu_w}\right)^{0.17}$$

By assuming $\mu_b = \mu_w$, then:

$$Nu_h = 0.108 \cdot (13314)^{0.703} (3.32)^{1/3} = 127.8$$

$$h_h = \frac{Nu_h k}{D_e} = \frac{127.8 \times 0.635}{0.00478} = 16977 \quad \text{W/(m}^2 \cdot \text{K)}$$

$$Nu_c = 0.108 \cdot (8859)^{0.703} (5.21)^{1/3} = 111.5$$

$$h_c = \frac{Nu_c k}{D_e} = \frac{111.5 \times 0.616}{0.00478} = 14372 \quad \text{W/(m}^2 \cdot \text{K)}$$

Overall heat transfer coefficient:

$$\frac{1}{U_c} = \frac{1}{h_c} + \frac{1}{h_h} + \frac{0.0006}{17.5}$$

$$U_c = \left[\frac{1}{h_c} + \frac{1}{h_h} + \frac{0.0006}{17.5}\right]^{-1} = \left[\frac{1}{14372} + \frac{1}{16977} + \frac{0.0006}{17.5}\right]^{-1} = 6144 \quad \text{W/(m}^2\text{.K)}$$

Fouled overall heat transfer coefficient:

$$\frac{1}{U_f} = \frac{1}{U_c} + 0.00006$$

$$U_f = \left[\frac{1}{U_c} + 0.00006\right]^{-1} = \left[\frac{1}{6144} + 0.0005\right]^{-1} = 4700 \ \ W/(m^2.K)$$

The corresponding cleanliness factor is:

$$CF = \frac{U_f}{U_c} = \frac{4700}{6144} = 0.76$$

Actual heat duties for clean surface are:

$$Q_c = U_c A_e \Delta T_m = 6144 \times 110 \times 23 = 15544 \ \ kW$$

$$Q_f = U_f A_e \Delta T_m = 4700 \times 110 \times 23 = 11891 \ \ kW$$

The safety factor is:

$$C_s = \frac{Q_f}{Q_r} = \frac{11891}{11698} = 1.01$$

The percentage over surface design is:

$$OS = 100 U_c R_{ft} = 100 \times 6144 \times 0.00005 = 30.72\%$$

Hot and cold fluid friction coefficients:

$$f_h = \frac{0.76}{(Re_h)^{0.215}} = \frac{0.76}{13314^{0.215}} = 0.099$$

$$f_c = \frac{0.76}{(Re_c)^{0.215}} = \frac{0.76}{6456^{0.215}} = 0.108$$

Hot and cold fluid frictional pressure drop:

$$\Delta p_c = 4f \frac{L_{eff} N_p}{D_e} \frac{G_c{}^2}{2\rho} \left(\frac{\mu_b}{\mu_w}\right)^{-0.17}$$

$$(\Delta p_c)_h = 4 \times 0.099 \times \frac{1.55 \times 1}{0.00478} \times \frac{1423.3^2}{2 \times 986} = 131912 \ \ Pa$$

$$(\Delta p_c)_c = 4 \times 0.108 \times \frac{1.5 \times 1}{0.00478} \times \frac{1423.3^2}{2 \times 995} = 142602 \ \ Pa$$

Port mass velocity:

$$G_{p,h} = G_{p,c} = \frac{\dot{m}}{\pi\left(\frac{D_p{}^2}{4}\right)} = \frac{140}{\pi \times \left(\frac{0.2^2}{4}\right)} = 4458.6 \ \ kg/(m^2 s)$$

Pressure drop in port ducts:

$$\Delta p_p = 1.4 N_p \frac{G_p{}^2}{2\rho}$$

$$(\Delta p_c)_h = 1.4 \times 1 \times \frac{4458.6^2}{2 \times 986} = 14112.96 \ \ Pa$$

$$(\Delta p_c)_c = 1.4 \times 1 \times \frac{4458.6^2}{2 \times 995} = 13985.31 \ \ Pa$$

Total pressure drop for hot and cold fluids:

$$\Delta p_t = \Delta p_C + \Delta p_P$$

$$\left(\Delta p_t\right)_h = \left(\Delta p_c\right)_h + \left(\Delta p_p\right)_h = 131912 + 14112.96 = 146025 \text{ Pa}$$

$$\left(\Delta p_t\right)_c = \left(\Delta p_c\right)_c + \left(\Delta p_p\right)_c = 142602 + 13985.31 = 156587 \text{ Pa}$$

For β = 0.65:

Hot fluid heat transfer coefficient, h_h—This can be obtained by referring to Table 11.6 C_h=0.087, and n= 0.718 for β = 65 ^{o}C:

$$Nu_h = \frac{h_h D_e}{k} = 0.087 \cdot \left(Re_h\right)^{0.718} \left(Pr\right)^{1/3} \left(\frac{\mu_b}{\mu_w}\right)^{0.17}$$

By assuming $\mu_b = \mu_w$, then:

$$Nu_h = 0.087 \cdot \left(13314\right)^{0.718} \left(3.32\right)^{1/3} = 118.7$$

$$h_h = \frac{Nu_h k}{D_e} = \frac{118.7 \times 0.635}{0.00478} = 15770 \text{ W/(m}^2 \cdot \text{K)}$$

$$Nu_c = 0.087 \cdot \left(8859\right)^{0.718} \left(5.21\right)^{1/3} = 103$$

$$h_c = \frac{Nu_c k}{D_e} = \frac{103 \times 0.616}{0.00478} = 13269 \text{ W/(m}^2 \cdot \text{K)}$$

Overall heat transfer coefficient:

$$\frac{1}{U_c} = \frac{1}{h_c} + \frac{1}{h_h} + \frac{0.0006}{17.5}$$

$$U_c = \left[\frac{1}{h_c} + \frac{1}{h_h} + \frac{0.0006}{17.5}\right]^{-1} = \left[\frac{1}{13269} + \frac{1}{15770} + \frac{0.0006}{17.5}\right]^{-1} = 5778 \text{ W/(m}^2\text{.K)}$$

Fouled overall heat transfer coefficient:

$$\frac{1}{U_f} = \frac{1}{U_c} + 0.00006$$

$$U_f = \left[\frac{1}{U_c} + 0.00006\right]^{-1} = \left[\frac{1}{5778} + 0.0005\right]^{-1} = 4483 \text{ W/(m}^2\text{.K)}$$

The corresponding cleanliness factor is:

$$CF = \frac{U_f}{U_c} = \frac{4483}{5778} = 0.78$$

Actual heat duties for clean surface are:

$$Q_c = U_c A_e \Delta T_m = 5778 \times 110 \times 23 = 14618 \text{ kW}$$

$$Q_f = U_f A_e \Delta T_m = 4483 \times 110 \times 23 = 11342 \text{ kW}$$

The safety factor is:

$$C_s = \frac{Q_f}{Q_r} = \frac{11342}{11698} = 0.97$$

The percentage over surface design is:

$$OS = 100 U_c R_{ft} = 100 \times 5778 \times 0.00005 = 28.89\%$$

Hot and cold fluid friction coefficients:

$$f_h = \frac{0.639}{(\mathrm{Re}_h)^{0.213}} = \frac{0.639}{13314^{0.213}} = 0.084$$

$$f_c = \frac{0.639}{(\mathrm{Re}_c)^{0.213}} = \frac{0.639}{6456^{0.213}} = 0.099$$

<u>Hot and cold fluid frictional pressure drop:</u>

$$\Delta p_c = 4f \frac{L_{eff} N_p}{D_e} \frac{G_c^2}{2\rho} \left(\frac{\mu_b}{\mu_w}\right)^{-0.17}$$

$$(\Delta p_c)_h = 4 \times 0.084 \times \frac{1.55 \times 1}{0.00478} \times \frac{1423.3^2}{2 \times 986} = 111926 \text{ Pa}$$

$$(\Delta p_c)_c = 4 \times 0.099 \times \frac{1.5 \times 1}{0.00478} \times \frac{1423.3^2}{2 \times 995} = 131912 \text{ Pa}$$

<u>Port mass velocity:</u>

$$G_{p,h} = G_{p,c} = \frac{\dot{m}}{\pi\left(\frac{D_p^2}{4}\right)} = \frac{140}{\pi \times \left(\frac{0.2^2}{4}\right)} = 4458.6 \text{ kg/(m}^2\text{s)}$$

<u>Pressure drop in port ducts:</u>

$$\Delta p_p = 1.4 N_p \frac{G_p^2}{2\rho}$$

$$(\Delta p_c)_h = 1.4 \times 1 \times \frac{4458.6^2}{2 \times 986} = 14112.96 \text{ Pa}$$

$$(\Delta p_c)_c = 1.4 \times 1 \times \frac{4458.6^2}{2 \times 995} = 13985.31 \text{ Pa}$$

<u>Total pressure drop for hot and cold fluids:</u>

$$\Delta p_t = \Delta p_C + \Delta p_P$$

$$(\Delta p_t)_h = (\Delta p_c)_h + (\Delta p_p)_h = 111926 + 14112.96 = 126039 \text{ Pa}$$

$$(\Delta p_t)_c = (\Delta p_c)_c + (\Delta p_p)_c = 131912 + 13985.31 = 145897 \text{ Pa}$$

Problem 11.4

Repeat Example 11.1 for 30% or less oversurface design.

SOLUTION:

Hot fluid properties:

$$\rho = 986 \ \text{kg/m}^3$$
$$\mu = 5.11\times10^{-4} \ \text{kg/m.s}$$
$$c_p = 4183 \ \text{J/kg.K}$$
$$Pr = 3.32$$
$$k = 0.635 \ \text{W/(m.K)}$$

Cold water properties:

$$\rho = 995 \ \text{kg/m}^3$$
$$\mu = 7.68\times10^{-4} \ \text{kg/m.s}$$
$$c_p = 4178 \ \text{J/kg.K}$$
$$Pr = 5.21$$
$$k = 0.616 \ \text{W/(m.K)}$$

Stepwise performance analysis:

Heat duty:

$$Q_{rc} = \left(\dot{m}c_p\right)_c\left(T_{c2} - T_{c1}\right) = 140\times4178\times\left(42-22\right) = 11698.4 \ \text{W}$$
$$Q_{rh} = \left(\dot{m}c_p\right)_h\left(T_{h1} - T_{h2}\right) = 140\times4183\times\left(65-45\right) = 11712.4 \ \text{W}$$

Log-mean temperature difference:

$$\Delta T_1 = \Delta T_2 = \Delta T_{lm,cf} = 23 \ ^oC$$

The effective number of plate is:

$$N_e = N_t - 2 = 103$$

Plate pitch:

$$p = \frac{L_c}{N_t} = \frac{0.38}{105} = 0.0036 \ \text{m}$$

Mean channel flow gap:

$$b = 0.0036 - 0.0006 = 0.003 \ \text{m}$$

One channel flow area:

$$A_{ch} = b\times L_w = 0.003\times0.63 = 0.00189 \ \text{m}^2$$

The single-plate heat transfer area:

$$A_1 = \frac{A_e}{N_e} = \frac{110}{103} = 1.067 \ \text{m}^2$$

Enlargement factor:

$$\phi = \frac{1.067}{0.85} = 1.255$$

Channel equivalent diameter:

$$D_e = \frac{4(b \cdot L_w)}{2(b + L_w\phi)} = \frac{2b}{\phi} = \frac{2 \times 0.003}{1.255} = 0.00478 \quad \text{m}$$

<u>The number of channel per pass:</u>

$$N_{cp} = \frac{N_t - 1}{2N_p} = \frac{105 - 1}{2 \times 1} = 52$$

<u>Mass flow rate per channel:</u>

$$\dot{m}_{ch} = \frac{140}{52} = 2.69 \quad \text{kg/s}$$

<u>Mass velocity, G_{ch}:</u>

$$G_{ch} = \frac{2.69}{0.00189} = 1423.3 \quad \text{kg/(m}^2\text{s)}$$

<u>Hot and cold fluid Reynolds numbers are:</u>

$$Re_h = \frac{G_{ch} \cdot D_e}{\mu_h} = \frac{1423.3 \times 0.00478}{5.11 \times 10^{-4}} = 13314$$

$$Re_c = \frac{G_{ch} \cdot D_e}{\mu_c} = \frac{1423.3 \times 0.00478}{7.68 \times 10^{-4}} = 8859$$

<u>Hot fluid heat transfer coefficient, h_h</u>—This can be obtained by referring to Table 11.6 C_h=0.3, and n= 0.633 for β = 45 °C:

$$Nu_h = \frac{h_h D_e}{k} = 0.3 \cdot (Re_h)^{0.633} (Pr)^{1/3} \left(\frac{\mu_b}{\mu_w}\right)^{0.17}$$

By assuming $\mu_b = \mu_w$, then:

$$Nu_h = 0.3 \cdot (13314)^{0.633} (3.32)^{1/3} = 242.8$$

$$h_h = \frac{Nu_h k}{D_e} = \frac{242.8 \times 0.635}{0.00478} = 32255 \quad \text{W/(m}^2 \cdot \text{K)}$$

$$Nu_c = 0.3 \cdot (8859)^{0.633} (5.21)^{1/3} = 215.4$$

$$h_c = \frac{Nu_c k}{D_e} = \frac{215.4 \times 0.616}{0.00478} = 27759 \quad \text{W/(m}^2 \cdot \text{K)}$$

<u>Overall heat transfer coefficient:</u>

$$\frac{1}{U_c} = \frac{1}{h_c} + \frac{1}{h_h} + \frac{0.0006}{17.5}$$

$$U_c = \left[\frac{1}{h_c} + \frac{1}{h_h} + \frac{0.0006}{17.5}\right]^{-1} = 9870 \quad \text{W/(m}^2\text{.K)}$$

<u>For 30% of oversurface design:</u>

$$OS = 100 U_c R_{ft} = 30\%$$

$$R_{ft} = \frac{OS}{100 U_c} = \frac{0.3}{100 \times 9870} = 0.0000003$$

<u>Fouled overall heat transfer coefficient:</u>

$$\frac{1}{U_f} = \frac{1}{U_c} + 0.0000003$$

$$U_f = \left[\frac{1}{U_c} + 0.0000003\right]^{-1} = \left[\frac{1}{9870} + 0.0000003\right]^{-1} = 9840 \ \text{W/(m}^2\text{.K)}$$

The corresponding cleanliness factor is:

$$CF = \frac{U_f}{U_c} = \frac{9840}{9870} = 0.997$$

Actual heat duties for clean surface are:

$$Q_c = U_c A_e \Delta T_m = 9870 \times 110 \times 23 = 24971 \ \text{kW}$$

$$Q_f = U_f A_e \Delta T_m = 9840 \times 110 \times 23 = 24895 \ \text{kW}$$

The safety factor is:

$$C_s = \frac{Q_f}{Q_r} = \frac{24895}{11698} = 2.13$$

Hot and cold fluid friction coefficients:

$$f_h = \frac{1.441}{(Re_h)^{0.206}} = \frac{1.441}{13314^{0.206}} = 0.204$$

$$f_c = \frac{1.441}{(Re_c)^{0.206}} = \frac{1.441}{8859^{0.206}} = 0.222$$

Hot and cold fluid frictional pressure drop:

$$\Delta p_c = 4f \frac{L_{eff} N_p}{D_e} \frac{G_c^{\ 2}}{2\rho} \left(\frac{\mu_b}{\mu_w}\right)^{-0.17}$$

$$(\Delta p_c)_h = 4 \times 0.204 \times \frac{1.55 \times 1}{0.00478} \times \frac{1423.3^2}{2 \times 986} = 271819 \ \text{Pa}$$

$$(\Delta p_c)_c = 4 \times 0.222 \times \frac{1.5 \times 1}{0.00478} \times \frac{1423.3^2}{2 \times 995} = 293128 \ \text{Pa}$$

Port mass velocity:

$$G_{p,h} = G_{p,c} = \frac{\dot{m}}{\pi\left(\frac{D_p^{\ 2}}{4}\right)} = \frac{140}{\pi \times \left(\frac{0.2^2}{4}\right)} = 4458.6 \ \text{kg/(m}^2\text{s)}$$

Pressure drop in port ducts:

$$\Delta p_p = 1.4 N_p \frac{G_p^{\ 2}}{2\rho}$$

$$(\Delta p_c)_h = 1.4 \times 1 \times \frac{4458.6^2}{2 \times 986} = 14112.96 \ \text{Pa}$$

$$(\Delta p_c)_c = 1.4 \times 1 \times \frac{4458.6^2}{2 \times 995} = 13985.31 \ \text{Pa}$$

Total pressure drop for hot and cold fluids:

$$\Delta p_t = \Delta p_C + \Delta p_P$$

$$(\Delta p_t)_h = (\Delta p_c)_h + (\Delta p_p)_h = 271819 + 14112.96 = 285932 \ \text{Pa}$$

$$\left(\Delta p_t\right)_c = \left(\Delta p_c\right)_c + \left(\Delta p_p\right)_c = 293128 + 13985.31 = 307113 \text{ Pa}$$

Problem 11.5

Solve Problem 11.2 for two-pass/two-pass arrangement.

GIVEN:

- -Inlet temperature of hot wastewater (T_{h1}) = 90 ^oC
- -Mass flow rate of hot water ($\dot{m}_h$) = 12 kg/s
- -Inlet temperature of cold water (T_{c1}) = 15 oC
- -Outlet temperature of cold water (T_{c2}) = 45 oC
- -Mass flow rate of cold water ($\dot{m}_c$) = 6 kg/s
- -Fouling resistance (R_{fc}) = 0.00006 m^2.K/W
- -Plate thermal conductivity (Titanium , k_w) = 20 W/(m.K)
- -Gasket heat exchanger configuration:

L_v = 1.60 m	L_w = 0.50 m
b = 6 mm	ϕ = 1.17
β = 50 ^oC	t = 0.0006 m
D_p = 0.15 m	

SOLUTION:

Cold water properties at $T_m = \dfrac{T_{in} + T_{out}}{2} = \dfrac{15+45}{2} = 30\ ^oC$:

$$\rho = 996 \text{ kg/m}^3$$
$$\mu = 8.15\times10^{-4} \text{ kg/m.s}$$
$$c_p = 4179 \text{ J/kg.K}$$
$$Pr = 5.58$$
$$k = 0.612 \text{ W/(m.K)}$$

Hot water properties at T_{h1} = 90 ^oC:

$$\rho = 965 \text{ kg/m}^3$$
$$\mu = 3.16\times10^{-4} \text{ kg/m.s}$$
$$c_p = 4205 \text{ J/kg.K}$$
$$Pr = 1.96$$
$$k = 0.675 \text{ W/(m.K)}$$

a.

Heat duty:

$$Q = (\dot{m}c_p)_c (T_{c2} - T_{c1}) = (\dot{m}c_p)_h (T_{h1} - T_{h2})$$
$$= 6\times4179\times(45-15) = 752220 \text{ W}$$

$$T_{h2} = T_{h1} - \frac{(\dot{m}c_p)_c (T_{c2}-T_{c1})}{(\dot{m}c_p)_h} = 90 - \frac{6\times4179\times(45-15)}{12\times4205} = 75\ ^oC$$

Log-mean temperature difference:

$$\Delta T_{lm,cf} = \frac{(T_{h1} - T_{c2}) - (T_{h2} - T_{c1})}{\ln\frac{(T_{h1} - T_{c2})}{(T_{h2} - T_{c1})}} = \frac{(90-45)-(75-15)}{\ln\frac{90-45}{75-15}} = 52.14 \ {}^{o}C$$

To solve for the number of plates needed, a trial-and-error method has to be used. First, an estimated heat transfer coefficient is assumed, which is verified and iterated later until the assumed value matches the value resulted from it.

Assume the overall heat transfer coefficient U_c to be 7000 W/(m^2.K) (From table 11.2).

For safety factor C_s = 1.4,

$$Q_f = C_s \cdot Q_r = 1.4 \times 752220 = 1053108 \ \text{W}$$

where $Q_f = U_f \cdot A_e F \Delta T_{lm,cf}$, then:

$$A_e = \frac{Q_f}{U_f \cdot F\Delta T_{lm,cf}} = \frac{1053108}{7000 \times 1 \times 52.14} = 2.885 \ \text{m}^2$$

The single-plate projected area:

$$A_{1p} = L_w \cdot L_p = (L_v - D_p) \cdot L_w = (1.6 - 0.15) \times 0.5 = 0.725 \ \text{m}^2$$

The single-plate heat transfer area:

$$A_1 = A_{1p} \cdot \phi = 0.725 \times 1.17 = 0.848 \,\text{m}^2$$

The number of plates:

$$N_e = \frac{A_e}{A_1} = \frac{2.885}{0.848} = 3.4 \approx 5 \ \text{plates} \qquad \text{(odd number)}$$

$$\therefore N_t = N_e + 2 = 5 + 2 = 7 \,\text{plates}$$

Verify the assumed heat transfer coefficient:

One channel flow area:

$$A_{ch} = b \times L_w = 0.006 \times 0.5 = 0.003 \ \ \text{m}^2$$

Channel equivalent diameter:

$$D_e = \frac{4(b \cdot L_w)}{2(b + L_w \phi)} = \frac{2b}{\phi} = \frac{2 \times 0.006}{1.17} = 0.01026 \ \ \text{m}$$

The number of channels per pass, N_{cp}:

$$N_{cp} = \frac{N_t - 1}{2 \cdot N_p} = \frac{7-1}{2 \times 2} = 2$$

mass flow rate per channel:

$$\dot{m}_{ch,h} = \frac{12}{2} = 6 \ \ \text{kg/s}$$

$$\dot{m}_{ch,c} = \frac{6}{2} = 6 \ \ \text{kg/s}$$

Mass velocity, G_{ch}:

$$G_{ch,h} = \frac{6}{0.0039} = 1538.5 \ \ \text{kg/(m}^2\text{s)}$$

$$G_{ch,c} = \frac{3}{0.0039} = 769.2 \ \ \text{kg/(m}^2\text{s)}$$

Hot and cold fluid Reynolds numbers are:

$$\mathrm{Re}_h = \frac{G_{ch,h} \cdot D_e}{\mu_h} = \frac{1538.5 \times 0.01026}{3.16 \times 10^{-4}} = 49953$$

$$\mathrm{Re}_c = \frac{G_{ch,c} \cdot D_e}{\mu_c} = \frac{769.2 \times 0.01026}{8.15 \times 10^{-4}} = 9684$$

Hot fluid heat transfer coefficient, h_h—This can be obtained by referring to Table 11.6 C_h=0.13, and n= 0.732 for $\beta = 50\ ^{o}C$:

$$\mathrm{Nu_h} = \frac{h_h D_e}{k} = 0.3 \cdot (\mathrm{Re_h})^{0.732} (\mathrm{Pr})^{1/3} \left(\frac{\mu_b}{\mu_w}\right)^{0.17}$$

By assuming $\mu_b = \mu_w$, then:

$$Nu_h = 0.3 \cdot (49953)^{0.732} (5.58)^{1/3} = 1455.0$$

$$h_h = \frac{Nu_h k}{D_e} = \frac{1455.0 \times 0.612}{0.01026} = 86788\ \mathrm{W/(m^2 \cdot K)}$$

$$Nu_c = 0.3 \cdot (9684)^{0.732} (1.96)^{1/3} = 310.0$$

$$h_c = \frac{Nu_c k}{D_e} = \frac{310.0 \times 0.675}{0.01026} = 20395\ \mathrm{W/(m^2 \cdot K)}$$

Overall heat transfer coefficient:

$$\frac{1}{U_c} = \frac{1}{h_c} + \frac{1}{h_h} + \frac{0.0006}{20}$$

$$U_c = \left[\frac{1}{h_c} + \frac{1}{h_h} + \frac{0.0006}{20}\right]^{-1} = \left[\frac{1}{20395} + \frac{1}{86788} + \frac{0.0006}{20}\right]^{-1} = 11043\ \mathrm{W/(m^2.K)}$$

Fouled overall heat transfer coefficient:

$$\frac{1}{U_f} = \frac{1}{U_c} + 0.00006$$

$$U_f = \left[\frac{1}{U_c} + 0.00006\right]^{-1} = \left[\frac{1}{11043} + 0.00006\right]^{-1} = 6642\ \mathrm{W/(m^2.K)}$$

So, the resulting heat transfer coefficient is less than the assumed 7000 W/(m^2.K), the procedure has to be repeated, assuming the U_f to be 6642 W/(m^2.K) as follows:

$$A_e = \frac{Q_f}{U_f \cdot F\Delta T_{lm,cf}} = \frac{1053108}{6642 \times 1 \times 52.14} = 3.04\ \mathrm{m^2}$$

The number of plates:

$$N_e = \frac{A_e}{A_1} = \frac{3.04}{0.848} = 3.6 \approx 5$$ plates (round off to odd number to make equal number of channel passes for cold and hot fluid)

$$\therefore N_t = N_e + 2 = 5 + 2 = 7 \text{ plates}$$

The following calculation is the same as the above procedure, where we obtain the fouled overall heat transfer coefficient as 7912 W/(m^2.K).

So the number of plates we get is:

$$\therefore N_t = N_e + 2 = 5 + 2 = 7 \text{ plates}$$

and the effective surface area is:

$$A_e = N_e \cdot A_1 = 5 \times 0.848 = 4.24 \ \text{m}^2$$

b.

Hot and cold fluid friction coefficients:

$$f_h = \frac{0.772}{(\text{Re}_h)^{0.161}} = \frac{0.772}{49953^{0.161}} = 0.135$$

$$f_c = \frac{0.772}{(\text{Re}_c)^{0.161}} = \frac{0.772}{9684^{0.161}} = 0.176$$

Hot and cold fluid frictional pressure drop:

$$\Delta p_c = 4f \frac{L_{eff} N_p}{D_e} \frac{G_c^2}{2\rho} \left(\frac{\mu_b}{\mu_w}\right)^{-0.17}$$

$$(\Delta p_c)_h = 4 \times 0.135 \times \frac{1.6 \times 2}{0.01026} \times \frac{1538.5^2}{2 \times 965} = 206554 \ \text{Pa}$$

$$(\Delta p_c)_c = 4 \times 0.176 \times \frac{1.6 \times 2}{0.01026} \times \frac{769.2^2}{2 \times 996} = 65218 \ \text{Pa}$$

Port mass velocity:

$$G_{p,h} = \frac{\dot{m}}{\pi\left(\frac{D_p^2}{4}\right)} = \frac{12}{\pi \times \left(\frac{0.15^2}{4}\right)} = 679.06 \quad \text{kg/(m}^2\text{s)}$$

$$G_{p,c} = \frac{\dot{m}_c}{\pi\left(\frac{D_p^2}{4}\right)} = \frac{6}{\pi \times \left(\frac{0.15^2}{4}\right)} = 339.53 \quad \text{kg/(m}^2\text{s)}$$

Pressure drop in port ducts:

$$\Delta p_p = 1.4 N_p \frac{G_p^2}{2\rho}$$

$$(\Delta p_c)_h = 1.4 \times 2 \times \frac{679.06^2}{2 \times 965} = 669.0 \ \text{Pa}$$

$$(\Delta p_c)_c = 1.4 \times 2 \times \frac{339.53^2}{2 \times 996} = 162.0 \ \text{Pa}$$

Total pressure drop for hot and cold fluids:

$$\Delta p_t = \Delta p_C + \Delta p_P$$

$$(\Delta p_t)_h = (\Delta p_c)_h + (\Delta p_p)_h = 206554 + 669 = 207223 \ \text{Pa}$$

$$(\Delta p_t)_c = (\Delta p_c)_c + (\Delta p_p)_c = 65218 + 162 = 65380 \ \text{Pa}$$

Problem 11.6

A heat exchanger is required to heat treated cooling water with a flow reate of 60 kg/s from 10 to 50 ^oC using the waste heat from water, cooling from 60 to 20 ^oC with the same mass flow rate as the cold water. The maximum allowable pressure drop for both streams is 120 kPa A gasketed-plate heat exchanger with 301 plates having a channel width of 50 cm, a vertical distance of 1.5 m between ports is proposed, and the plate pitch is 0.0035 m with an enlargement factor of 1.25. The spacing between them is 6 mm. Plates are made of stainless steel (316) (k = 16.5 W/m.K). For a two-pass/two-pass arrangement, analyze the problem to see if the proposed design is feasible. Could this heat exchanger be smaller or larger?

GIVEN:

-Inlet temperature of hot wastewater (T_{h1}) = 60 ^oC
-Outlet temperature of hot wastewater (T_{h2}) = 20 ^oC
-Mass flow rate of hot water ($\dot{m}_h$) = 60 kg/s
-Inlet temperature of cold water (T_{c1}) = 10 ^oC
-Outlet temperature of cold water (T_{c2}) = 50 ^oC
-Mass flow rate of cold water ($\dot{m}_c$) = 60 kg/s
-Fouling resistance (R_{fc}) = 0.00006 m^2.K/W
-Plate thermal conductivity (stainless steel, k_w) = 16.5 W/(m.K)
-Number of plates (N_t) = 301
-Allowable pressure drop(Δp) = 120 kPa
-Gasket heat exchanger configuration:

L_v = 1.5 m	L_w = 0.50 m
p = 0.0035 m	ϕ = 1.25
β = 60 ^{o}	t = 0.0006 m
D_p = 0.15 m	

SOLUTION:

Hot fluid properties at $T_m = \frac{T_{h1+}T_{h2}}{2} = \frac{60+20}{2} = 40\ ^oC$:

$\rho = 992.2\ kg/m^3$

$\mu = 6.53\ x\ 10^{-4}\ kg/m.s$

$c_p = 4179\ J/kg.K$

$\mathrm{Pr} = 4.34$

$k = 0.629\ W/m.K$

Cold water properties at $T_m = \frac{T_{c1+}T_{c2}}{2} = \frac{50+10}{2} = 30\ ^oC$:

$\rho = 995.7\ kg/m^3$

$\mu = 7.975\ x\ 10^{-4}\ kg/m.s$

$c_p = 4178\ J/kg.K$

$\mathrm{Pr} = 5.43$

$k = 0.614\ W/m.K$

Stepwise performance analysis:

Heat duty:

$$Q_{rc} = (\dot{m}c_p)_c (T_{c2} - T_{c1}) = 60\ x\ 4178\ x\ (50 - 10) = 10027.2\ kW$$

$$Q_{rh} = (\dot{m}c_p)_h (T_{h1} - T_{h2}) = 60\ x\ 4179 x\ (60 - 20) = 10029.6\ kW$$

Log -mean temperature difference:

$$\Delta T_1 = \Delta T_2 = \Delta T_{lm,cf} = 10\ ^oC$$

The effective number of plate is:

$$N_e = N_t - 2 = 299$$

Mean channel flow gap:

$$b = 0.0035 - 0.0006 = 0.0029\ m$$

One channel flow area:

$$A_{ch} = b\ x\ L_w = 0.0029\ x\ 0.5 = 0.00145\ m^2$$

The single plate projected area:

$$A_{1p} = L_w\ .\ L_p = (L_v - D_p)\ .\ L_w = (1.5 - 0.15)\ x\ 0.5 = 0.675\ m^2$$

The single plate heat transfer area:

$$A_1 = A_{1p}\ .\ \emptyset = 0.675\ x\ 1.25 = 0.844\ m^2$$

Channel equivalent diameter:

$$D_e = \frac{4(b.L_w)}{2(b + L_w\emptyset)} = \frac{2b}{\emptyset} = \frac{2\ x\ 0.0029}{1.25} = 0.00464\ m$$

The number of channel per pass:

$$N_{cp} = \frac{N_t - 1}{2N_p} = \frac{301 - 1}{2\ x\ 2} = 75$$

Mass flow rate per channel:

$$\dot{m}_{ch} = \frac{60}{75} = 0.8\ kg/s$$

Mass velocity, G_{ch}:

$$G_{ch} = \frac{0.8}{0.00145} = 551.7\ kg/(m^2 s)$$

Hot and cold fluid Reynolds numbers are:

$$Re_h = \frac{G_{ch}\ .\ D_e}{\mu_h} = \frac{551.7\ x\ 0.00464}{6.53\ x\ 10^{-4}} = 3920.2$$

$$Re_c = \frac{G_{ch}\ .\ D_e}{\mu_c} = \frac{551.7\ x\ 0.00464}{7.975\ x\ 10^{-4}} = 3209.8$$

Hot fluid and cold fluid heat transfer coefficient, h_h and h_c: This can be obtained by referring to Table 11.6 (Kumar's correlation) C_h=0.108, and n=0.703 for β=60° :

$$Nu_h = \frac{h_h \ x \ D_e}{k} = 0.108 \ x \ (Re_h)^{0.703}(Pr)^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.17}$$

By assuming $\mu_b=\mu_w$, then:

$$Nu_h = 0.108 \ x \ (3920.2)^{0.703}(4.34)^{1/3} = 59.16$$

$$h_h = \frac{Nu_h \ x \ k}{D_e} = \frac{59.16 \ x \ 0.629}{0.00464} = 8019.75 \ W/(m^2.K)$$

$$Nu_c = 0.108 \ x \ (3209.8)^{0.703}(5.43)^{1/3} = 55.39$$

$$h_c = \frac{Nu_c \ x \ k}{D_e} = \frac{55.39 \ x \ 0.614}{0.00464} = 7329.6 \ W/(m^2.K)$$

Overall heat transfer coefficient:

$$\frac{1}{U_c} = \frac{1}{h_h} + \frac{1}{h_c} + \frac{0.0006}{16.5}$$

$$U_c = \left[\frac{1}{8019.75} + \frac{1}{7329.6} + \frac{0.0006}{16.5}\right]^{-1} = 3361.5 \ W/(m^2.K)$$

For %30 of oversurface design:

$$OS = 100U_cR_{ft} = 30\%$$

$$R_{ft} = \frac{OS}{100U_c} = \frac{0.3}{100 \ x \ 3361.5} = 8.92 \ x \ 10^{-7} \ \ (m^2.K)/W$$

Fouled overall heat transfer coefficient:

$$\frac{1}{U_f} = \frac{1}{U_c} + 8.92 \ x \ 10^{-7}$$

$$U_f = \left[\frac{1}{U_c} + 8.92 \ x \ 10^{-7}\right]^{-1} = \left[\frac{1}{3361.5} + 8.92 \ x \ 10^{-7}\right]^{-1} = 3351.4 \, W/(m^2.K)$$

Corresponding cleanliness factor:

$$CF = \frac{U_f}{U_c} = \frac{3351.4}{3361.5} = 0.997$$

Actual heat duties for clean surface:

$$Q_c = U_cA_e\Delta T_m = 3361.5 \ x \ 299 \ x \ 0.844 \ x \ 10 = 8482.95 \ kW$$

$$Q_f = U_fA_e\Delta T_m = 3351.4 \ x \ 299 \ x \ 0.844 \ x \ 10 = 8457.49 \ kW$$

The safety factor:

$$C_s = \frac{Q_f}{Q_r} = \frac{8457.49}{10029.6} = 0.84$$

Hot and cold friction coefficients with Kumar's correlation from Table 11.6 for K_p=0.760 and m=0.215:

$$f_h = \frac{0.760}{(Re_h)^{0.215}} = \frac{0.760}{(3920.2)^{0.215}} = 0.128$$

$$f_c = \frac{0.760}{(Re_c)^{0.215}} = \frac{0.760}{(3209.8)^{0.215}} = 0.134$$

Hot and cold fluid frictional pressure drop:

$$\Delta P_c = 4f \frac{L_{eff} N_p}{D_e} \frac{G_c{}^2}{2\rho} \left(\frac{\mu_b}{\mu_w}\right)^{-0.17}$$

$$(\Delta P_c)_h = 4 \; x \; 0.128 \; x \; \frac{1.5 \; x \; 2}{0.00464} \; x \; \frac{551.7^2}{2 \; x \; 992.2} = 50775 Pa$$

$$(\Delta P_c)_c = 4 \; x \; 0.134 \; x \; \frac{1.5 \; x \; 2}{0.00464} \; x \; \frac{551.7^2}{2 \; x \; 995.7} = 52968.2 \; Pa$$

Port mass velocity:

$$G_{p,h} = G_{p,c} = \frac{\dot{m}}{\pi \left(\frac{D_p^2}{4}\right)} = \frac{60}{\pi \left(\frac{0.15^2}{4}\right)} = 3395.3 \; kg/(m^2 s)$$

Pressure drop in port ducts:

$$\Delta \mathrm{P_p} = 1.4 N_p \frac{G_p{}^2}{2\rho}$$

$$(\Delta P_p)_h = 1.4 \; x \; 2 \; x \; \frac{3395.3^2}{2 \; x \; 992.2} = 16266.2 \; Pa$$

$$(\Delta P_p)_c = 1.4 \; x \; 2 \; x \; \frac{3395.3^2}{2 \; x \; 995.7} = 16209 \; Pa$$

Total pressure drop for hot and cold fluids:

$$\Delta P_t = \Delta P_c + \Delta P_p$$

$$(\Delta P_t)_h = (\Delta P_c)_h + (\Delta P_p)_h = 50775 + 16266.2 = 67041.2 \; Pa$$

$$(\Delta P_t)_c = (\Delta P_c)_c + (\Delta P_p)_c = 52968.2 + 16209 = 69177.2 \; Pa$$

As a result, this heat exchanger can be smaller since pressure drop values are lower than allowable pressure drop. Heat exchanger size can be reduced until pressure drop values get close to allowable pressure drop value.

Problem 11.7

Analyze problem 11.6 with the use of the correlations proposed by Kumar and correlation proposed by Chisholm and Wanniarachchi to see if the proposed design is feasible. Then compare the results of both correlations. Could this heat exchanger be smaller or larger?

GIVEN:

-Inlet temperature of hot wastewater (T_{h1}) = 60 ^oC
-Outlet temperature of hot wastewater (T_{h2}) = 20 ^oC
-Mass flow rate of hot water ($\dot{m}_h$) = 60 kg/s
-Inlet temperature of cold water (T_{c1}) = 10 ^oC
-Outlet temperature of cold water (T_{c2}) = 50 ^oC
-Mass flow rate of cold water ($\dot{m}_c$) = 60 kg/s
-Fouling resistance (R_{fc}) = 0.00006 m^2.K/W
-Plate thermal conductivity (stainless steel, k_w) = 16.5 W/(m.K)
-Number of plates (N_t) = 301
-Allowable pressure drop(Δp) = 120 kPa
-Gasket heat exchanger configuration:

L_v = 1.5 m L_w = 0.50 m
p = 0.0035 m ϕ = 1.25
β = 60 ^{o} t = 0.0006 m
D_p = 0.15 m

SOLUTION:

Hot fluid properties at $T_m = \frac{T_{h1+}T_{h2}}{2} = \frac{60+20}{2} = 40\ ^oC$:

$\rho = 992.2\ kg/m^3$

$\mu = 6.53\ x\ 10^{-4}\ kg/m.s$

$c_p = 4179\ J/kg.K$

$\mathrm{Pr} = 4.34$

$k = 0.629\ W/m.K$

Cold water properties at $T_m = \frac{T_{c1+}T_{c2}}{2} = \frac{50+10}{2} = 30\ ^oC$:

$\rho = 995.7\ kg/m^3$

$\mu = 7.975\ x\ 10^{-4}\ kg/m.s$

$c_p = 4178\ J/kg.K$

$\mathrm{Pr} = 5.43$

$k = 0.614\ W/m.K$

Stepwise performance analysis:

<u>Heat duty:</u>

$$Q_{rc} = \left(\dot{m}c_p\right)_c (T_{c2} - T_{c1}) = 60\ x\ 4178\ x\ (50 - 10) = 10027.2\ kW$$

$$Q_{rh} = \left(\dot{m}c_p\right)_h (T_{h1} - T_{h2}) = 60\ x\ 4179 x\ (60 - 20) = 10029.6\ kW$$

Log -mean temperature difference:

$$\Delta T_1 = \Delta T_2 = \Delta T_{lm,cf} = 10\ ^oC$$

The effective number of plate is:

$$N_e = N_t - 2 = 299$$

Mean channel flow gap:

$$b = 0.0035 - 0.0006 = 0.0029\ m$$

One channel flow area:

$$A_{ch} = b\ x\ L_w = 0.0029\ x\ 0.5 = 0.00145\ m^2$$

The single plate projected area:

$$A_{1p} = L_w\ .\ L_p = \left(L_v - D_p\right).L_w = (1.5 - 0.15)\ x\ 0.5 = 0.675\ m^2$$

The single plate heat transfer area:

$$A_1 = A_{1p}\ .\emptyset = 0.675\ x\ 1.25 = 0.844\ m^2$$

Channel equivalent diameter for Kumar's correlation:

$$D_e = \frac{4(b.L_w)}{2(b + L_w\emptyset)} = \frac{2b}{\emptyset} = \frac{2\ x\ 0.0029}{1.25} = 0.00464\ m$$

Channel equivalent diameter for Chisholm and Wanniarachchi's correlations:

$$D_e = 2b = 2\ x\ 0.0029 = 0.0058 m$$

The number of channel per pass:

$$N_{cp} = \frac{N_t - 1}{2N_p} = \frac{301 - 1}{2\ x\ 2} = 75$$

Mass flow rate per channel:

$$\dot{m}_{ch} = \frac{60}{75} = 0.8\ \ kg/s$$

Mass velocity, G_{ch}:

$$G_{ch} = \frac{0.8}{0.00145} = 551.7\ \ kg/(m^2 s)$$

Solution with Kumar's Correlation from Table 11.6 for β=60º :

Hot and cold fluid Reynolds numbers are:

$$Re_h = \frac{G_{ch}\ .\ D_e}{\mu_h} = \frac{551.7\ x\ 0.00464}{6.53\ x\ 10^{-4}} = 3920.2$$

$$Re_c = \frac{G_{ch}\ .\ D_e}{\mu_c} = \frac{551.7\ x\ 0.00464}{7.975\ x\ 10^{-4}} = 3209.8$$

Hot fluid and cold fluid heat transfer coefficient, h_h and h_c: This can be obtained by referring to Table 11.6 (Kumar's correlation) C_h=0.108, and n=0.703 for β=60° :

$$Nu_h = \frac{h_h\ x\ D_e}{k} = 0.108\ x\ (Re_h)^{0.703}(Pr)^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.17}$$

By assuming $\mu_b=\mu_w$, then:

$$Nu_h = 0.108\ x\ (3920.2)^{0.703}(4.34)^{1/3} = 59.16$$

$$h_h = \frac{Nu_h\ x\ k}{D_e} = \frac{59.16\ x\ 0.629}{0.00464} = 8019.75\ W/(m^2.K)$$

$$Nu_c = 0.108\ x\ (3209.8)^{0.703}(5.43)^{1/3} = 55.39$$

$$h_c = \frac{Nu_c\ x\ k}{D_e} = \frac{55.39\ x\ 0.614}{0.00464} = 7329.6\ W/(m^2.K)$$

Overall heat transfer coefficient:

$$\frac{1}{U_c} = \frac{1}{h_h} + \frac{1}{h_c} + \frac{0.0006}{16.5}$$

$$U_c = \left[\frac{1}{8019.75} + \frac{1}{7329.6} + \frac{0.0006}{16.5}\right]^{-1} = 3361.5\ W/(m^2.K)$$

For %30 of oversurface design:

$$OS = 100U_cR_{ft} = 30\%$$

$$R_{ft} = \frac{OS}{100U_c} = \frac{0.3}{100\ x\ 3361.5} = 8.92\ x\ 10^{-7}\ \ (m^2.K)/W$$

Fouled overall heat transfer coefficient:

$$\frac{1}{U_f} = \frac{1}{U_c} + 8.92\ x\ 10^{-7}$$

$$U_f = \left[\frac{1}{U_c} + 8.92\ x\ 10^{-7}\right]^{-1} = \left[\frac{1}{3361.5} + 8.92\ x\ 10^{-7}\right]^{-1} = 3351.4\,W/(m^2.K)$$

Corresponding cleanliness factor:

$$CF = \frac{U_f}{U_c} = \frac{3351.4}{3361.5} = 0.997$$

Actual heat duties for clean surface:

$$Q_c = U_cA_e\Delta T_m = 3361.5\ x\ 299\ x\ 0.844\ x\ 10 = 8482.95\ kW$$

$$Q_f = U_fA_e\Delta T_m = 3351.4\ x\ 299\ x\ 0.844\ x\ 10 = 8457.49\ kW$$

The safety factor:

$$C_s = \frac{Q_f}{Q_r} = \frac{8457.49}{10029.6} = 0.84$$

Hot and cold friction coefficients with Kumar's correlation from Table 11.6 for K_p=0.760 and m=0.215:

$$f_h = \frac{0.760}{(Re_h)^{0.215}} = \frac{0.760}{(3920.2)^{0.215}} = 0.128$$

$$f_c = \frac{0.760}{(Re_c)^{0.215}} = \frac{0.760}{(3209.8)^{0.215}} = 0.134$$

Hot and cold fluid frictional pressure drop:

$$\Delta P_c = 4f \frac{L_{eff} N_p}{D_e} \frac{{G_c}^2}{2\rho} \left(\frac{\mu_b}{\mu_w}\right)^{-0.17}$$

$$(\Delta P_c)_h = 4 \, x \, 0.128 \, x \, \frac{1.5 \, x \, 2}{0.00464} \, x \, \frac{551.7^2}{2 \, x \, 992.2} = 50775 Pa$$

$$(\Delta P_c)_c = 4 \, x \, 0.134 \, x \, \frac{1.5 \, x \, 2}{0.00464} \, x \, \frac{551.7^2}{2 \, x \, 995.7} = 52968.2 \, Pa$$

Port mass velocity:

$$G_{p,h} = G_{p,c} = \frac{\dot{m}}{\pi \left(\frac{D_p^2}{4}\right)} = \frac{60}{\pi \left(\frac{0.15^2}{4}\right)} = 3395.3 \;\; kg/(m^2 s)$$

Pressure drop in port ducts:

$$\Delta \mathrm{P_p} = 1.4 N_p \frac{{G_p}^2}{2\rho}$$

$$(\Delta P_p)_h = 1.4 \, x \, 2 \, x \, \frac{3395.3^2}{2 \, x \, 992.2} = 16266.2 \, Pa$$

$$(\Delta P_p)_c = 1.4 \, x \, 2 \, x \, \frac{3395.3^2}{2 \, x \, 995.7} = 16209 \, Pa$$

Total pressure drop for hot and cold fluids:

$$\Delta P_t = \Delta P_c + \Delta P_p$$

$$(\Delta P_t)_h = (\Delta P_c)_h + (\Delta P_p)_h = 50775 + 16266.2 = 67041.2 \, Pa$$

$$(\Delta P_t)_c = (\Delta P_c)_c + (\Delta P_p)_c = 52968.2 + 16209 = 69177.2 \, Pa$$

Solution with Chisholm and Wanniarachchi's correlation from Table 11.5 for β=60º:

Hot and cold fluid Reynolds numbers are:

$$Re_h = \frac{G_{ch} \, . \, D_e}{\mu_h} = \frac{551.7 \, x \, 0.0058}{6.53 \, x \, 10^{-4}} = 4900.2$$

$$Re_c = \frac{G_{ch} \, . \, D_e}{\mu_c} = \frac{551.7 \, x \, 0.0058}{7.975 \, x \, 10^{-4}} = 4012.4$$

Hot fluid and cold fluid heat transfer coefficient, h_h and h_c: This can be obtained by referring to Table 11.5 for β=60º and Ø=1.25:

$$Nu_h = 0.72\ x\ Re_h^{0.59} Pr^{0.4} \emptyset^{0.41}\ (\beta/30)^{0.66}$$

$$Nu_h = 0.72\ x\ (4900.2)^{0.59}(4.34)^{0.4}(1.25)^{0.41}\ (60/30)^{0.66} = 337.3$$

$$h_h = \frac{Nu_h\ x\ k}{D_e} = \frac{337.3\ x\ 0.629}{0.0058} = 36579.6\ \ W/(m^2.K)$$

$$Nu_c = 0.72\ x\ Re_c^{0.59} Pr^{0.4} \emptyset^{0.41}\ (\beta/30)^{0.66}$$

$$Nu_c = 0.72\ x\ (4012.4)^{0.59}(5.43)^{0.4}(1.25)^{0.41}\ (60/30)^{0.66} = 327.8$$

$$h_c = \frac{Nu_c\ x\ k}{D_e} = \frac{327.8\ x\ 0.614}{0.0058} = 34701.6\ \ W/(m^2.K)$$

<u>Overall heat transfer coefficient:</u>

$$\frac{1}{U_c} = \frac{1}{h_h} + \frac{1}{h_c} + \frac{0.0006}{16.5}$$

$$U_c = \left[\frac{1}{36579.6} + \frac{1}{34701.6} + \frac{0.0006}{16.5}\right]^{-1} = 10808.7\, W/(m^2.K)$$

<u>For %30 of oversurface design:</u>

$$OS = 100 U_c R_{ft} = 30\%$$

$$R_{ft} = \frac{OS}{100U_c} = \frac{0.3}{100\ x\ 10808.7} = 2.78\ x\ 10^{-7}\ \ (m^2.K)/W$$

<u>Fouled overall heat transfer coefficient:</u>

$$\frac{1}{U_f} = \frac{1}{U_c} + 2.78x\ 10^{-7}$$

$$U_f = \left[\frac{1}{U_c} + 2.78\ x\ 10^{-7}\right]^{-1} = \left[\frac{1}{10808.7} + 2.78\ x\ 10^{-7}\right]^{-1} = 10776.3\ \ W/(m^2.K)$$

<u>Corresponding cleanliness factor:</u>

$$CF = \frac{U_f}{U_c} = \frac{10776.3}{10808.7} = 0.997$$

<u>Actual heat duties for clean surface with:</u>

$$Q_c = U_c A_e \Delta T_m = 10808.7\ x\ 299\ x\ 0.844\ x\ 10 = 27276.4\ kW$$

$$Q_f = U_f A_e \Delta T_m = 10776.3x\ 299\ x\ 0.844\ x\ 10 = 27194.6\ kW$$

<u>The safety factor:</u>

$$C_s = \frac{Q_f}{Q_r} = \frac{27194.6}{10029.6} = 2.71$$

<u>Hot and cold friction coefficients:</u>

$$f_h = 0.08(Re_h)^{-0.25} \emptyset^{1.25} (\beta/30)^{3.6} = \ 0.08(4900.2)^{-0.25}(1.25)^{1.25}(60/30)^{3.6}$$

$$= \ 0.153$$

$$f_c = 0.08(Re_c)^{-0.25}\emptyset^{1.25}(\beta/30)^{3.6} = 0.08(4012.4)^{-0.25}(1.25)^{1.25}(60/30)^{3.6}$$
$$= 0.161$$

Hot and cold fluid frictional pressure drop:

$$\Delta P_c = 4f\frac{L_{eff}N_p}{D_e}\frac{{G_c}^2}{2\rho}\left(\frac{\mu_b}{\mu_w}\right)^{-0.17}$$

$$(\Delta P_c)_h = 4\ x\ 0.153\ x\ \frac{1.5\ x\ 2}{0.0058}\ x\ \frac{551.7^2}{2\ x\ 992.2} = 48553.6\ Pa$$

$$(\Delta P_c)_c = 4\ x\ 0.161\ x\ \frac{1.5\ x\ 2}{0.0058}\ x\ \frac{551.7^2}{2\ x\ 995.7} = 50912.8\ Pa$$

Port mass velocity:

$$G_{p,h} = G_{p,c} = \frac{\dot{m}}{\pi\left(\frac{D_p^2}{4}\right)} = \frac{60}{\pi\left(\frac{0.15^2}{4}\right)} = 3395.3\ kg/(m^2s)$$

Pressure drop in port ducts:

$$\Delta \mathrm{P_p} = 1.4N_p\frac{{G_p}^2}{2\rho}$$

$$(\Delta P_p)_h = 1.4\ x\ 2\ x\ \frac{3395.3^2}{2\ x\ 992.2} = 16266.2\ Pa$$

$$(\Delta P_p)_c = 1.4\ x\ 2\ x\ \frac{3395.3^2}{2\ x\ 995.7} = 16209\ Pa$$

Total pressure drop for hot and cold fluids with:

$$\Delta P_t = \Delta P_c + \Delta P_p$$

$$(\Delta P_t)_h = (\Delta P_c)_h + (\Delta P_p)_h = 48553.6 + 16266.2 = 64819.8\ Pa$$

$$(\Delta P_t)_c = (\Delta P_c)_c + (\Delta P_p)_c = 50912.8 + 16209 = 67121.8\ Pa$$

Table 1. Comparison of the results

	Kumar's Correlation	Chisholm and Wanniarachchi's Correlation
Heat Transfer Coefficient (Hot Fluid) [W/m^{2}K]	8019.8	36579.6
Heat Transfer Coefficient (Cold Fluid)[W/m^{2}K]	7329.6	34701.6
Overall Heat Transfer Coefficient[W/m^{2}K]	3361.5	10808.7
Friction Coefficient (Hot Fluid)	0.1	0.2
Friction Coefficient (Cold Fluid)	0.1	0.2
Frictional Pressure Drop (Hot Fluid) [Pa]	50775.0	48553.6
Frictional Pressure Drop (Cold Fluid) [Pa]	52968.2	50912.8
Pressure Drop in Port Ducts (Hot Fluid) [Pa]	16266.2	16266.2
Pressure Drop in Port Ducts (Cold Fluid) [Pa]	16209.0	16209.0
Total Pressure Drop (Hot Fluid) [Pa]	67041.2	64819.8
Total Pressure Drop (Cold Fluid) [Pa]	69177.2	67121.8

As a result, this heat exchanger can be smaller since pressure drop values are lower than allowable pressure drop. Heat exchanger size can be reduced until pressure drop values get close to allowable pressure drop value.

Problem 11.8

Repeat Example 11.1 for different types of water conditions:
a. Coastal ocean (R_{fc} = 0.0005 K.m^2/W), demineralized closed loop (0.000001 K.m^2/W)
b. Wastewater (0.00001 K.m^2/W), cooling tower (R_{fc} = 0.000069 m^2.K/W)
Write you conclusions.

SOLUTION:

Hot fluid properties:

$$\rho = 986 \ \ kg/m^3$$
$$\mu = 5.11 \times 10^{-4} \ \ kg/m.s$$
$$c_p = 4183 \ \ J/kg.K$$
$$Pr = 3.32$$
$$k = 0.635 \ \ W/(m.K)$$

Cold water properties:

$$\rho = 995 \ \ kg/m^3$$
$$\mu = 7.68 \times 10^{-4} \ \ kg/m.s$$
$$c_p = 4178 \ \ J/kg.K$$
$$Pr = 5.21$$
$$k = 0.616 \ \ W/(m.K)$$

Stepwise performance analysis:

Heat duty:

$$Q_{rc} = \left(\dot{m}c_p\right)_c \left(T_{c2} - T_{c1}\right) = 140 \times 4178 \times (42 - 22) = 11698.4 \ \ W$$
$$Q_{rh} = \left(\dot{m}c_p\right)_h \left(T_{h1} - T_{h2}\right) = 140 \times 4183 \times (65 - 45) = 11712.4 \ \ W$$

Log-mean temperature difference:

$$\Delta T_1 = \Delta T_2 = \Delta T_{lm,cf} = 23 \ {}^oC$$

The effective number of plate is:

$$N_e = N_t - 2 = 103$$

Plate pitch:

$$p = \frac{L_c}{N_t} = \frac{0.38}{105} = 0.0036 \ \ m$$

Mean channel flow gap:

$$b = 0.0036 - 0.0006 = 0.003 \ \ m$$

One channel flow area:

$$A_{ch} = b \times L_w = 0.003 \times 0.63 = 0.00189 \ \ m^2$$

The single-plate heat transfer area:

$$A_1 = \frac{A_e}{N_e} = \frac{110}{103} = 1.067 \ \ m^2$$

Enlargement factor:

$$\phi = \frac{1.067}{0.85} = 1.255$$

Channel equivalent diameter:

$$D_e = \frac{4(b \cdot L_w)}{2(b + L_w\phi)} = \frac{2b}{\phi} = \frac{2 \times 0.003}{1.255} = 0.00478 \quad \text{m}$$

The number of channel per pass:

$$N_{cp} = \frac{N_t - 1}{2N_p} = \frac{105 - 1}{2 \times 1} = 52$$

Mass flow rate per channel:

$$\dot{m}_{ch} = \frac{140}{52} = 2.69 \quad \text{kg/s}$$

Mass velocity, G_{ch}:

$$G_{ch} = \frac{2.69}{0.00189} = 1423.3 \quad \text{kg/(m}^2\text{s)}$$

Hot and cold fluid Reynolds numbers are:

$$Re_h = \frac{G_{ch} \cdot D_e}{\mu_h} = \frac{1423.3 \times 0.00478}{5.11 \times 10^{-4}} = 13314$$

$$Re_c = \frac{G_{ch} \cdot D_e}{\mu_c} = \frac{1423.3 \times 0.00478}{7.68 \times 10^{-4}} = 8859$$

a.

Hot fluid heat transfer coefficient, h_h—This can be obtained by referring to Table 11.6 C_h=0.3, and n= 0.633 for β = 45 °C:

$$Nu_h = \frac{h_h D_e}{k} = 0.3 \cdot (Re_h)^{0.633} (Pr)^{1/3} \left(\frac{\mu_b}{\mu_w}\right)^{0.17}$$

By assuming $\mu_b = \mu_w$, then:

$$Nu_h = 0.3 \cdot (13314)^{0.633} (3.32)^{1/3} = 242.8$$

$$h_h = \frac{Nu_h k}{D_e} = \frac{242.8 \times 0.635}{0.00478} = 32255 \ \text{W/(m}^2 \cdot \text{K)}$$

$$Nu_c = 0.3 \cdot (8859)^{0.633} (5.21)^{1/3} = 215.4$$

$$h_c = \frac{Nu_c k}{D_e} = \frac{215.4 \times 0.616}{0.00478} = 27759 \ \text{W/(m}^2 \cdot \text{K)}$$

Overall heat transfer coefficient:

$$\frac{1}{U_c} = \frac{1}{h_c} + \frac{1}{h_h} + \frac{0.0006}{17.5}$$

$$U_c = \left[\frac{1}{h_c} + \frac{1}{h_h} + \frac{0.0006}{17.5}\right]^{-1} = 9870 \ \text{W/(m}^2\text{.K)}$$

Fouled overall heat transfer coefficient:

$$\frac{1}{U_f} = \frac{1}{U_c} + 0.0005 + 0.000001$$

$$U_f = \left[\frac{1}{U_c} + 0.000501\right]^{-1} = \left[\frac{1}{9870} + 0.000501\right]^{-1} = 1660 \;\; W/(m^2.K)$$

The corresponding cleanliness factor is:

$$CF = \frac{U_f}{U_c} = \frac{1660}{9870} = 0.17 \quad m^2$$

Actual heat duties for clean surface are:

$$Q_c = U_c A_e \Delta T_m = 9870 \times 110 \times 23 = 24971 \quad kW$$

$$Q_f = U_f A_e \Delta T_m = 1660 \times 110 \times 23 = 4200 \quad kW$$

The safety factor is:

$$C_s = \frac{Q_f}{Q_r} = \frac{4200}{11698} = 0.359$$

Can not be used for the specific heat transfer purpose, since the fouled overall heat transfer is too low to meet the requirement.

b.

Fouled overall heat transfer coefficient:

$$\frac{1}{U_f} = \frac{1}{U_c} + 0.00001 + 0.000069$$

$$U_f = \left[\frac{1}{U_c} + 0.000079\right]^{-1} = \left[\frac{1}{9870} + 0.000079\right]^{-1} = 5546 \;\; W/(m^2.K)$$

The corresponding cleanliness factor is:

$$CF = \frac{U_f}{U_c} = \frac{5546}{9870} = 0.56 \quad m^2$$

Actual heat duties for clean surface are:

$$Q_c = U_c A_e \Delta T_m = 9870 \times 110 \times 23 = 24971 \quad kW$$

$$Q_f = U_f A_e \Delta T_m = 5546 \times 110 \times 23 = 14031 \quad kW$$

The safety factor is:

$$C_s = \frac{Q_f}{Q_r} = \frac{14031}{11698} = 1.2$$

This heat exchanger is acceptable for the heat transfer design. From a. and b., it is seen that there is no use to keep the fouling resistance of only one side very low, and should keep the total fouling resistance below certain value to meet the heat transfer requirements.

Problem 11.9

A one-pass countercurrent flow heat exchanger has 201 plates. The exchanger has a vertical port distance of 2 m and is 0.6 m wide, with a gap between the plates of 6 mm. The heat exchanger will be used for the following process: cold water from the city supply with an inlet temperature of 10 °C is fed to the heat exchanger at a rate of 15 kg/s that will be heated to 75°C with a wastewater entering at an inlet temperature of 90 °C. The flow rate of hot water is 30 kg/s, which is a distilled water. The other construction parameters are given as in Problem 11.2. There is no limitation on the pressure drop. Is this heat exchanger suitable for this purpose (larger or smaller)?

GIVEN:

- -Inlet temperature of hot wastewater (T_{h1}) = 90 °C
- -Mass flow rate of hot water ($\dot{m}_h$) = 30 kg/s
- -Inlet temperature of cold water (T_{c1}) = 10 °C
- -Outlet temperature of cold water (T_{c2}) = 75 °C
- -Mass flow rate of cold water ($\dot{m}_c$) = 15 kg/s
- -Plate thermal conductivity (Titanium , k_w) = 20 W/(m.K)
- -Gasket heat exchanger configuration:

L_v = 2.0 m	L_w = 0.60 m
b = 6 mm	ϕ = 1.17
β = 50°	t = 0.0006 m
D_p = 0.15 m	

SOLUTION:

Cold water properties at $T_m = \dfrac{T_{in} + T_{out}}{2} = \dfrac{10+75}{2} = 42.5\,^oC$:

$$\rho = 991 \ \ kg/m^3$$
$$\mu = 6.24\times10^{-4} \ \ kg/m.s$$
$$c_p = 4179 \ \ J/kg.K$$
$$Pr = 4.14$$
$$k = 0.630 \ \ W/(m.K)$$

Hot water properties at T_{h1} = 90 °C:

$$\rho = 965 \ \ kg/m^3$$
$$\mu = 3.16\times10^{-4} \ \ kg/m.s$$
$$c_p = 4205 \ \ J/kg.K$$
$$Pr = 1.96$$
$$k = 0.675 \ \ W/(m.K)$$

Heat duty:

$$Q = (\dot{m}c_p)_c (T_{c2} - T_{c1}) = (\dot{m}c_p)_h (T_{h1} - T_{h2})$$
$$= 15\times4179\times(75-10) = 4074525 \ \ W$$

$$T_{h2} = T_{h1} - \frac{(\dot{m}c_p)_c (T_{c2} - T_{c1})}{(\dot{m}c_p)_h} = 90 - \frac{15\times4179\times(75-10)}{30\times4205} = 57.7 \ \ ^oC$$

Log-mean temperature difference:

$$\Delta T_{lm,cf} = \frac{\left(T_{h1}-T_{c2}\right)-\left(T_{h2}-T_{c1}\right)}{\ln\frac{\left(T_{h1}-T_{c2}\right)}{\left(T_{h2}-T_{c1}\right)}} = \frac{(90-75)-(57.7-10)}{\ln\frac{90-75}{57.7-10}} = 21.7\ {}^{o}C$$

The effective number of plate is:

$$N_e = N_t - 2 = 199$$

One channel flow area:

$$A_{ch} = b\times L_w = 0.006\times 0.60 = 0.0036 \quad m^2$$

The single-plate projected area:

$$A_{1p} = L_w\cdot L_p = \left(L_v - D_p\right)\cdot L_w = (2.0-0.15)\times 0.6 = 1.11\ \ m^2$$

The single-plate heat transfer area:

$$A_1 = A_{1p}\cdot\phi = 1.11\times 1.17 = 1.2987 \quad m^2$$

Channel equivalent diameter at ϕ = 1.17:

$$D_e = \frac{4\left(b\cdot L_w\right)}{2\left(b+L_w\phi\right)} = \frac{2b}{\phi} = \frac{2\times 0.006}{1.17} = 0.01025 \quad m$$

The number of channel per pass:

$$N_{cp} = \frac{N_t - 1}{2N_p} = \frac{201-1}{2\times 1} = 100$$

Mass flow rate per channel:

$$\dot{m}_{ch,h} = \frac{30}{100} = 0.3 \quad kg/s$$

$$\dot{m}_{ch,c} = \frac{15}{100} = 0.15 \quad kg/s$$

Mass velocity, G_{ch}:

$$G_{ch,h} = \frac{0.3}{0.0036} = 83.33 \quad kg/(m^2s)$$

$$G_{ch,c} = \frac{0.15}{0.0036} = 41.67 \quad kg/(m^2s)$$

Hot and cold fluid Reynolds numbers are:

$$\mathrm{Re}_h = \frac{G_{ch}\cdot D_e}{\mu_h} = \frac{83.33\times 0.01025}{3.16\times 10^{-4}} = 2703$$

$$\mathrm{Re}_c = \frac{G_{ch}\cdot D_e}{\mu_c} = \frac{41.67\times 0.01025}{6.24\times 10^{-4}} = 684.5$$

Hot fluid heat transfer coefficient, h_h—This can be obtained by referring to Table 11.6 C_h=0.13, and n= 0.732 for β = 50 °C:

$$Nu_h = \frac{h_h D_e}{k} = 0.13\cdot\left(\mathrm{Re}_h\right)^{0.732}(\mathrm{Pr})^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.17}$$

By assuming $\mu_b = \mu_w$, then:

$$Nu_h = 0.13 \cdot (2703)^{0.732} (1.96)^{1/3} = 52.9$$

$$h_h = \frac{Nu_h k}{D_e} = \frac{52.9 \times 0.675}{0.01025} = 3484 \text{ W/(m}^2 \cdot \text{K)}$$

$$Nu_c = 0.13 \cdot (684.5)^{0.732} (4.14)^{1/3} = 24.84$$

$$h_c = \frac{Nu_c k}{D_e} = \frac{24.84 \times 0.630}{0.01025} = 1526 \text{ W/(m}^2 \cdot \text{K)}$$

<u>Overall heat transfer coefficient:</u>

$$\frac{1}{U_c} = \frac{1}{h_c} + \frac{1}{h_h} + \frac{0.0006}{20}$$

$$U_c = \left[\frac{1}{h_c} + \frac{1}{h_h} + \frac{0.0006}{20}\right]^{-1} = \left[\frac{1}{3484} + \frac{1}{1526} + \frac{0.0006}{20}\right]^{-1} = 1028.5 \text{ W/(m}^2\text{.K)}$$

<u>Fouled overall heat transfer coefficient:</u>

Assume the R_{fc} = 0.00006 (m^2.K/W) for city water, and R_{fh} = 0.000086(m^2.K/W) for the wastewater side.

$$\frac{1}{U_f} = \frac{1}{U_c} + 0.00006 + 0.000086$$

$$U_f = \left[\frac{1}{U_c} + 0.00006 + 0.000086\right]^{-1} = \left[\frac{1}{1028.5} + 0.000146\right]^{-1} = 894 \text{ W/(m}^2\text{.K)}$$

<u>The corresponding cleanliness factor is:</u>

$$CF = \frac{U_f}{U_c} = \frac{894}{1028.5} = 0.87 \text{ m}^2$$

<u>Actual heat duties for clean surface are:</u>

$$Q_c = U_c A_e \Delta T_m = 1028.5 \times 199 \times 1.2987 \times 21.7 = 5462.3 \text{ kW}$$

$$Q_f = U_f A_e \Delta T_m = 894 \times 199 \times 1.2987 \times 21.7 = 5013.7 \text{ kW}$$

<u>The safety factor is:</u>

$$C_s = \frac{Q_f}{Q_r} = \frac{5013.7 \times 10^3}{4074525} = 1.23$$

<u>The percentage over surface design is:</u>

$$OS = 100 U_c R_{ft} = 100 \times 1028.5 \times 0.000146 = 15\%$$

This heat exchanger is NOT suitable for this purpose since it is too large.

Problem 11.10

In Example 11.1 a new mass flow rate is provided as 90 kg/s. Assume that the inlet temperature of hot and cold fluid are changed to 120 and 20 °C, respectively. If the flow arrangement is single pass for both streams, use the ε-NTU method to calculate the outlet temperatures for the specific heat exchanger. There may be a constraint on the outlet temperature of the cold stream; then this type of analysis is important to see if the required outlet temperature of the cold stream is satisfied.

SOLUTION:

Cold water properties at $T_{C,in}$ = 20 °C (using this as approximation):

$$\rho = 999 \ \ \text{kg/m}^3$$
$$\mu = 10.67 \times 10^{-4} \ \ \text{kg/m.s}$$
$$c_p = 4183 \ \ \text{J/kg.K}$$
$$\text{Pr} = 8.20$$
$$k = 0.591 \ \ \text{W/(m.K)}$$

Hot water properties at $T_{h,in}$ = 120 °C:

$$\rho = 943 \ \ \text{kg/m}^3$$
$$\mu = 2.295 \times 10^{-4} \ \ \text{kg/m.s}$$
$$c_p = 4245 \ \ \text{J/kg.K}$$
$$\text{Pr} = 1.419$$
$$k = 0.687 \ \ \text{W/(m.K)}$$

The effective number of plate is:

$$N_e = N_t - 2 = 103$$

Plate pitch:

$$p = \frac{L_c}{N_t} = \frac{0.38}{105} = 0.0036 \ \text{m}$$

Mean channel flow gap:

$$b = 0.0036 - 0.0006 = 0.003 \ \text{m}$$

One channel flow area:

$$A_{ch} = b \times L_w = 0.003 \times 0.63 = 0.00189 \quad \text{m}^2$$

The single-plate heat transfer area:

$$A_1 = \frac{A_e}{N_e} = \frac{110}{103} = 1.067 \quad \text{m}^2$$

Enlargement factor:

$$\phi = \frac{1.067}{0.85} = 1.255$$

Channel equivalent diameter:

$$D_e = \frac{4(b \cdot L_w)}{2(b + L_w \phi)} = \frac{2b}{\phi} = \frac{2 \times 0.003}{1.255} = 0.00478 \quad \text{m}$$

The number of channel per pass:

$$N_{cp} = \frac{N_t - 1}{2N_p} = \frac{105-1}{2\times 1} = 52$$

Mass flow rate per channel:

$$\dot{m}_{ch} = \frac{90}{52} = 1.73 \quad \text{kg/s}$$

Mass velocity, G_{ch}:

$$G_{ch} = \frac{1.73}{0.00189} = 915.3 \quad \text{kg/(m}^2\text{s)}$$

Hot and cold fluid Reynolds numbers are:

$$Re_h = \frac{G_{ch}\cdot D_e}{\mu_h} = \frac{915.3\times 0.00478}{2.295\times 10^{-4}} = 19064$$

$$Re_c = \frac{G_{ch}\cdot D_e}{\mu_c} = \frac{915.3\times 0.00478}{10.67\times 10^{-4}} = 4100$$

Hot fluid heat transfer coefficient, h_h—This can be obtained by referring to Table 11.6 C_h=0.3, and n= 0.663 for β = 45 ^{o}C:

$$Nu_h = \frac{h_h D_e}{k} = 0.3\cdot(\text{Re}_h)^{0.663}(\text{Pr})^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.17}$$

By assuming $\mu_b = \mu_w$, then:

$$Nu_h = 0.3\cdot(19064)^{0.663}(1.419)^{1/3} = 231.8$$

$$h_h = \frac{Nu_h k}{D_e} = \frac{231.8\times 0.687}{0.00478} = 33311 \ \text{W/(m}^2\cdot\text{K)}$$

$$Nu_c = 0.3\cdot(4100)^{0.663}(8.2)^{1/3} = 149.3$$

$$h_c = \frac{Nu_c k}{D_e} = \frac{149.3\times 0.591}{0.00478} = 18455 \ \text{W/(m}^2\cdot\text{K)}$$

Overall heat transfer coefficient:

$$\frac{1}{U_c} = \frac{1}{h_c} + \frac{1}{h_h} + \frac{0.0006}{17.5}$$

$$U_c = \left[\frac{1}{h_c} + \frac{1}{h_h} + \frac{0.0006}{17.5}\right]^{-1} = 8439 \ \text{W/(m}^2\text{.K)}$$

Fouled overall heat transfer coefficient:

$$\frac{1}{U_f} = \frac{1}{U_c} + 0.00006$$

$$U_f = \left[\frac{1}{U_c} + 0.00006\right]^{-1} = \left[\frac{1}{8439} + 0.00006\right]^{-1} = 5602 \ \text{W/(m}^2\text{.K)}$$

The corresponding cleanliness factor is:

$$CF = \frac{U_f}{U_c} = \frac{5602}{8439} = 0.66$$

NTU:

$$C_c = \left(\dot{m}c_p\right)_c = 90 \times 4183 = 376470$$

$$C_h = \left(\dot{m}c_p\right)_h = 90 \times 4245 = 382050$$

$$\therefore\ C_{min} = C_c$$

$$NTU = \frac{AU}{C_{min}} = \frac{5602 \times 110}{376470} = 1.637$$

<u>ε for counterflow:</u>

$$\varepsilon = \frac{\exp\left[\left(1 - \frac{C_{min}}{C_{max}}\right)NTU\right] - 1}{\exp\left[\left(1 - \frac{C_{min}}{C_{max}}\right)NTU\right] - \frac{C_{min}}{C_{max}}} = 0.60$$

<u>Actual heat duties:</u>

$$Q = \varepsilon \cdot C_{min}\left(T_{h1} - T_{c1}\right) = 0.60 \times 376470 \times \left(120 - 20\right) = 22588 \ \text{ kW}$$

<u>Outlet temperature:</u>

$$T_{h2} = T_{h1} - \frac{Q}{\left(\dot{m}c_p\right)_h} = 120 - \frac{22588 \times 10^3}{90 \times 4245} = 61 \ {}^{o}\text{C}$$

$$T_{c2} = T_{c1} + \frac{Q}{\left(\dot{m}c_p\right)_h} = 20 + \frac{22588 \times 10^3}{90 \times 4183} = 80 \ {}^{o}\text{C}$$

Problem 12.1

(See also Problem 8.1) Design a shell-and-tube type power condenser for a 250 MW(e) coal-fired power station. Steam enters the turbine at 500°C and 4 MPa. The thermodynamic efficiency of the turbine is 0.85. Assume that the condenser pressure is 10 kPa. Cooling water for the operation is available at 15°C. Assume a laminar film condensation on the shell side.

SOLUTION:

Tube geometry:

$$\text{O.D. of the tubes, } d_o = 0.00254 \text{ m}$$
$$\text{I.D. of the tubes, } d_i = 0.02291 \text{ m}$$
$$\text{Tube wall thermal conductivity, } k = 111 \text{ W/(m.K)}$$

Cooling water properties at 15 ^{o}C:

$$\rho_c = 999 \text{ kg/m}^3$$
$$\mu_c = 11.54 \times 10^{-4} \text{ kg/m.s}$$
$$c_{p,c} = 4186 \text{ J/kg.K}$$
$$Pr_c = 8.2$$
$$k_c = 0.591 \text{ W/m.K}$$

Properties of the condensed liquid at 10 kPa, T_s = 45.8 ^{o}C:

$$i_{lg} = 2392 \text{ kJ/kg}$$
$$\rho_l = 990 \text{ kg/m}^3$$
$$\mu_l = 5.88 \times 10^{-4} \text{ kg/m.s}$$
$$k_l = 0.635 \text{ W/m.K}$$
$$i_l = 191.8 \text{ kJ/kg}$$

Thermodynamic properties of steam enter and exit the turbine:

$$h_1 = 3445.2 \text{ kJ/kg}, \qquad s_1 = 7.09 \text{ kJ/kg.K}$$

Since the expansion process in the turbine is almost the adiabatic, the state of the exit steam can be determined as:

$$h_2 = 2246 \text{ kJ/kg}, \qquad s_2 = 7.09 \text{ kJ/kg.K}$$
$$x_2 = 0.8587$$

As the turbine proficiency is 0.85, the actual exit enthalpy should be:

$$h_2' = 3445 - 0.85 \times (3445 - 2246) = 2425.85 \text{ kJ/kg}$$

Mass flow rate of wet steam entering the condenser:

$$\dot{m}_s = \frac{W_T}{\Delta i} = \frac{250 \times 10^3}{3445 - 2425.85} = 245.3 \text{ kg/s}$$

Fouling resistances chosen from TEMA tables in Chapter 6 are:

$$R_{fi} = 0.00018 \text{ m}^2\text{.K/W}$$
$$R_{fo} = 0.00009 \text{ m}^2\text{.K/W}$$

Condenser heat load:

$$Q = \dot{m}_c (i_{in} - i_l) = 245.3 \times (2425.85 - 191.8) = 549 \text{ MW}$$

By assume the temperature increase of cooling water to be 10 ^{o}C, the cooling water mass flow rate is calculated by:

$$\dot{m}_c = \frac{Q}{(T_{c2} - T_{c1}) c_{p,c}} = \frac{549 \times 10^6}{10 \times 4186}$$
$$= 13115 \text{ kg/s}$$

Number of tubes:

$$N_T = \frac{4\dot{m}_c}{u_c \rho_c \pi d_i^{\,2}} = \frac{4 \times 13115}{2 \times 999 \times \pi \times 0.02291^2} = 15924 \text{ tubes}$$

To calculate the condensing-side heat transfer coefficient, an estimate of the average number of tubes rows in a vertical column is needed. For typical condenser tube layouts, this was estimated as 70.

Tube-side Reynolds number:

$$Re = \frac{u_c \rho_c d_i}{\mu_c} = \frac{2 \times 999 \times 0.02291}{11.54 \times 10^{-4}} = 39666$$

Tube-side heat transfer coefficient:

$$f = (1.58 \ln Re - 3.28)^{-2} = \left(1.58 \times \ln(39666) - 3.28\right)^{-2} = 0.0055$$

$$Nu = \frac{(f/2)\, Re\, Pr}{1.07 + 12.7 (f/2)^{1/2}\left(Pr^{2/3} - 1\right)}$$

$$= \frac{(0.0055/2) \times 39666 \times 8.2}{1.07 + 12.7 (0.0055/2)^{1/2}\left(8.2^{2/3} - 1\right)}$$

$$= 287.4$$

$$h_i = \frac{Nu \cdot k_c}{d_i} = \frac{287.4 \times 0.591}{0.02291} = 7414 \;\; W/\left(m^2 K\right)$$

Assume the condenser is to be designed without subcooling, so that:

$$\Delta T_{in} = (45.8 - 15) = 30.8 \;\; ^oC$$

and

$$\Delta T_{out} = (45.8 - 25) = 20.8 \;\; ^oC$$

Hence the LMTD is:

$$\Delta T_{lm} = \frac{\Delta T_{in} - \Delta T_{out}}{\ln\left(\Delta T_{in} / \Delta T_{out}\right)} = \frac{30.8 - 20.8}{\ln(30.8/20.8)} = 25.5 \;\; ^oC$$

The shell-side heat transfer coefficient depends on the local heat flux and hence an iteration are necessary. The overall heat transfer coefficient, U, based on the tube O.D., is given by:

$$\frac{1}{U} = R_t + \frac{1}{h_o}$$

where h_o is the coefficient outside the tubes (shell side) and R_t is the sum of all other thermal resistances given by:

$$R_t = R_{fo} + \left[\frac{1}{h_i} + R_{fi}\right]\frac{d_o}{d_i} + \frac{t_w}{k_w}\frac{d_o}{D_m}$$

where D_m is approximated as:

$$D_m = \frac{d_0 - d_i}{\ln\left(d_o / d_i\right)} \approx \frac{1}{2}\left(d_o + d_i\right)$$

where t_w is the wall thickness.

Hence

$$R_t = 0.00009 + \left[\frac{1}{7414} + 0.00018\right]\frac{0.0254}{0.0229} + \frac{0.0013}{111}\frac{0.0254}{0.0229}$$

$$= 4.515 \times 10^{-4}$$

and

$$\frac{1}{U} = 4.515 \times 10^{-4} + \frac{1}{h_o}$$

The condensing-side heat transfer coefficient, h_o, may be calculated by the Nusselt method with the Kern correction for condensate inundation, hence

$$h_o = 0.728\left[\frac{\rho_l^2 g \cdot i_{lg} k_l^3}{\mu_l \Delta T_w d_o}\right]^{1/4} \frac{1}{N^{1/6}}$$

where ΔT_w is the difference between the saturation temperature and the temperature at the surface of the fouling. Because $\rho_l >> \rho_g$, the above equation has been simplified. By inserting the values, we get:

$$h_o = 0.728\left[\frac{(990)^2 (9.81)\cdot(2392\times10^3)(0.635)^3}{(5.66\times10^{-4})\Delta T_w (0.0254)}\right]^{1/4} \frac{1}{70^{1/6}}$$

$$= 8986 / (\Delta T_w)^{1/4}$$

The temperature difference, $\Delta T_{w,}$ is given by:

$$\Delta T_w = \Delta T - R_t q''$$

where ΔT is the local temperature difference between the streams; R_t is the sum of all other resistances; and q'' is the local heat flux, which is given by:

$$q'' = U\Delta T$$

Hence

$$\Delta T_w = \Delta T(1 - R_t U) = \Delta T(1 - 4.515\times10^{-4} U)$$

Table 12.1.1 and 12.1.2 summarize the results of this iteration for the inlet and outlet of the condenser when ΔT_{in} = 30.8 °C and ΔT_{out} = 20.8 °C, respectively. The initial guess of ΔT_w is 10°C.

Table 12.1.1 Iteration for Overall Coefficient at the Inlet of the Power Condenser

$\Delta T_{w,}$ °C	$h_{o,}$ W/m^2.K	U, W/m^2.K
10	5053	1540
25.46	4000	1425
25.56	3996	1425

Table 12.1.2 Iteration for Overall Coefficient at the Outlet of the Power Condenser

$\Delta T_{w,}$ °C	$h_{o,}$ W/m^2.K	U, W/m^2.K
10	5053	1540
17.19	4413	1474
17.23	4410	1474

The mean overall heat transfer coefficient can then be determined by taking the average of the inlet and outlet overall heat transfer coefficients:

$$U_m = \frac{1425 + 1474}{2} = 1450 \ \ W/m^2.K$$

The required surface area is therefore calculated as:

$$Q = U_m A_o \Delta T_m$$

$$A_o = \frac{Q}{U_m \Delta T_m} = \frac{549\times10^6}{1450\times25.5} = 14848 \ \ m^2$$

The required length is found by the following equation:

$$A_o = N_T \pi d_o L$$

$$L = \frac{A_o}{N_T \pi d_o} = \frac{14848}{15923\times\pi\times0.0254} = 11.7 \ \ m$$

Now that the length of the power condenser has been determined, the shell size of the condenser, the tube-side pressure drop, and the tube-side pumping power can all the calculated.

The expression of the shell diameter as a function of heat transfer area, A_o; tube length, L; and tube layout dimensions, P_T, PR, and d_o as parameters can be estimated from equation:

$$D_s = 0.637\sqrt{\frac{CL}{CPT}}\left[\frac{A_o(PR)^2 d_o}{L}\right]^{1/2}$$

where CL is the tube layout constant, CL = 1.00 for 90^o and 45^o; and CL = 0.87 for 30^o and 60^o.

CPT accounts fro the incomplete coverage of the shell diameter by the tubes:

CPT = 0.93 for 1-tube pass

CPT = 0.9 for 2-tube pass

CPT = 0.85 for 3-tube pass

P_T (the tube pitch) is 0.0381 m.

PR (the tube pitch ratio) is P_T/d_o = 0.0381/0.0254 = 1.501.

$$D_s = 0.637\sqrt{\frac{1}{0.93}}\left[\frac{14848\times(1.501)^2\times 0.0254}{11.7}\right]^{1/2} = 5.6 \text{ m}$$

The pressure drop on the tube side is calculated as:

$$\Delta p_{total} = \Delta p_t + \Delta p_r$$

The pressure drop through the tubes:

$$\Delta p_t = 4f\frac{LN_p}{D_e}\frac{G^2}{2\rho}$$

where

$$f = 0.046\,Re^{-0.2} = 0.046\times 39666^{-0.2} = 0.0055$$

$$G = u_c\rho$$

Therefore,

$$\Delta p_t = 4\times 0.0055\times\frac{11.7\times 1}{0.02291}\times\frac{(2\times 999)^2}{2\times 999}$$
$$= 22448 \text{ Pa}$$

The pressure drop due to the return is given by:

$$\Delta p_r = 4N_p\frac{G^2}{2\rho} = 4\times 1\times\frac{(2\times 999)^2}{2\times 999} = 7992 \text{ Pa}$$

Therefore, the total pressure drop on the tube side is determined by:

$$\Delta p_{total} = \Delta p_t + \Delta p_r$$
$$= 22448 + 7992$$
$$= 30440 \text{ Pa}$$

The pumping power is proportional to the pressure drop across the condenser:

$$P = \frac{\dot{m}\Delta p_{tot}}{\rho\eta_p} = \frac{13115\times 30440}{999\times 0.85} = 470141 \text{ W}$$

Problem 12.2

In a power plant, a shell-and-tube type heat exchanger is used as a condenser. This heat exchanger consist of 20,000 tubes and the fluid velocity through the tubes is 2 m/s. The tubes are made of Admiralty metal and have 18 BWG, 7/8-in. O.D. The cooling water enters at 20°C and exits at 30 °C. The average temperature of the tube walls is 55 °C and the shell side heat transfer coefficient is 4000W/m^2.K. Fouling on both sides is neglected. By making acceptable engineering assumptions, perform the thermal design of this condenser.

SOLUTION:

- Tube Geometry;

From Table 9.1,

The tube is 18 BWG,7/8 in. OD

$$\mathrm{OD} = {}^{7}/_{8}\ \mathrm{in.} = {}^{7}/_{8}\ \mathrm{x}\ 0.0254 = 0.022225\ \mathrm{m}$$

$$\mathrm{ID} = 0.777\ \mathrm{in.} = 0.777\ \mathrm{x}\ 0.0254 = 0.0197358\ \mathrm{m}$$

$$A_c = 0.4742\ \mathrm{in.}^2 = 0.4742\ \mathrm{x}\ (0.0254)^2 = 0.000305934\ \mathrm{m}^2$$

The mean temperature of the water can be calculated as below.

$$T_m = \frac{20 + 30}{2} = 25°C$$

Thermal properties of cooling water @ $T_m = 25°C$

$$\rho_c = 997\ {}^{kg}/_{m^3}$$

$$\mu_c = 0.000909\ {}^{kg}/_{ms}$$

$$c_{p_c} = 4180\ {}^{J}/_{kgK}$$

$$Pr_c = 6.29$$

$$k_c = 0.605\ {}^{W}/_{mK}$$

- Admiralty metal is known as Admiralty Brass (%70-73 Cu). [1]

Thermal conductivity of tube wall;

$$k_w = 110.784\ {}^{W}/_{mK}$$

- Mass flow rate of cooling water can be found as below;

$$\dot{m}_c = \rho u (A_c N_T) = 997\ \mathrm{x}\ 2\ \mathrm{x}\ (0.000305934\ \mathrm{x}\ 20000) = 12200.65\ {}^{kg}/_{s}$$

- Reynolds number;

$$Re = \frac{\rho u d_i}{\mu} = \frac{997\ \mathrm{x}\ 2\ \mathrm{x}\ 0.0197358}{0.000909} = 43292.8$$

- Heat load;

$$Q = \dot{m} c_{p_c} (T_{c_2} - T_{c_1}) = 12200.65\ \mathrm{x}\ 4180\ \mathrm{x}\ (30 - 20) = 509987170\ \mathrm{W}$$

$$Q = 510\ \mathrm{MW}$$

- Tube side heat transfer coefficient and friction factor can be calculated with 2 different correlations. So that two thermal designs can be defined.

Table 3.3,

$$f = (1.58\ \ln Re - 3.28)^{-2}$$

$$Nu = \frac{(f/2) Re Pr}{1.07 + 12.7 (f/2)^{1/2} (Pr^{2/3} - 1)}$$

$$f = (1.58\ \ln\ (43292.8) - 3.28)^{-2}$$

$$f = 0.0054164$$

$$Nu = \frac{(0.0054164/2)\ \mathrm{x}\ 43292.8\ \mathrm{x}\ 6.29}{1.07 + 12.7 (0.0054164/2)^{1/2} (6.29^{2/3} - 1)}$$

$$Nu = 277.13$$

$$h_i = \frac{Nu k}{d_i} = \frac{277.13\ \mathrm{x}\ 0.605}{0.0197358}$$

$$h_i = 8495.4\ W/m^2K$$

- Overall heat transfer coefficient can be calculated form Eq. 12.6.

$$\frac{1}{U_o} = \frac{1}{h_i} + \frac{1}{h_o} + \frac{d_o \ln(d_o/d_i)}{2k_w}$$

$$\frac{1}{U_o} = \frac{1}{8495.4} + \frac{1}{4000} + \frac{0.022225 \text{ x } \ln(1.126)}{2 \text{ x } 110.784}$$

$$U_o = 2634.25\ W/m^2K$$

Table 3.3,

$$f = (1.58 \ln Re - 3.28)^{-2}$$

$$Nu = \frac{(f/2)RePr}{1 + 8.7(f/2)^{1/2}(Pr - 1)}$$

$$f = (1.58 \ln (43292.8) - 3.28)^{-2}$$

$$f = 0.0054164$$

$$Nu = \frac{(0.0054164/2) \text{ x } 43292.8 \text{ x } 6.29}{1 + 8.7(0.0054164/2)^{1/2}(6.29 - 1)}$$

$$Nu = 217.22$$

$$h_i = \frac{Nuk}{d_i} = \frac{217.22 \text{ x } 0.605}{0.0197358}$$

$$h_i = 6658.87\ W/m^2K$$

- Overall heat transfer coefficient can be calculated form Eq. 12.6.

$$\frac{1}{U_o} = \frac{1}{h_i} + \frac{1}{h_o} + \frac{d_o \ln(d_o/d_i)}{2k_w}$$

$$\frac{1}{U_o} = \frac{1}{6658.87} + \frac{1}{4000} + \frac{0.022225 \text{ x } \ln(1.126)}{2 \text{ x } 110.784}$$

$$U_o = 2426.72\ W/m^2K$$

Problem 12.3

A surface condenser is designed as a two-tube-pass, shell-and-tube type of heat exchanger at 10 kPa (h_{fg} = 2007.5 kJ/kg, Ts = 45 ^oC). Coolant water enters at 15 ^oC and leaves at 25 ^oC inside the tubes. Heat transfer coefficient is 300 $W/m^2.K$. The tubing is thin walled, has a 5-cm I.D. and is made of carbon steel; and the length of the heat exchanger is 2 m.
Calculate:
a. LMTD
b. Steam mass flow rate
c. Surface area of the condenser
d. Number tubes
e. Effectiveness, ε

SOLUTION:

Cooling water properties at 20 ^oC:

$$\rho_c = 998 \ \ kg/m^3$$
$$\mu_c = 10.07 \times 10^{-4} \ \ kg/m.s$$
$$c_{p,c} = 4182 \ \ J/kg.K$$
$$Pr_c = 7.05$$
$$k_c = 0.599 \ \ W/m.K$$

a.

Assume the condenser is to be designed without subcooling, so that:

$$\Delta T_{in} = (45 - 15) = 30 \ \ ^oC$$

and

$$\Delta T_{out} = (45 - 25) = 20 \ \ ^oC$$

Hence the LMTD is:

$$\Delta T_{lm} = \frac{\Delta T_{in} - \Delta T_{out}}{\ln(\Delta T_{in}/\Delta T_{out})} = \frac{30 - 20}{\ln(30/20)} = 24.7 \ \ ^oC$$

b.

$$A_c = \pi (D/2) L = \pi (0.05/2) 2 = 0.157 \text{ m}^2$$

$$Q = UA_c \times N_t \Delta T_m = 300 \times 0.157 \times 2 \times 24.7 = 2328.857 \text{ W}$$

$$Q = \dot{m}_s h_{fg} = \dot{m}_s \times 2007.5 \times 10^3 = 2328.857 \text{ W}$$

$$\dot{m}_s = 0.00116 \text{ kg/s}$$

c.

$$A_c = \pi (D/2) L = \pi (0.05/2) 2 = 0.157 \text{ m}^2$$

d.

$N_t = 2$

e.

from steam propertiy; c_{ps} = 1894 J/kg K

$\dot{m}_s c_{ps} = 2.197 \text{ W/K}$

$Q = \dot{m}_c c_{pc} \Delta T_c = \dot{m}_c \times 4182 \times 10 = 2328.857 \text{ W}$

$\dot{m}_c = 0.056 \text{ kg/s}$

$\dot{m}_c c_{pc} = 232.886 \text{ W/K}$

$C_{\min} = 2.197 \text{ W/K}$

$C_{\max} = 232.886 \text{ W/K}$

$C_{\min}/C_{\max} = 0.0094$

$AU/C_{\min} = 21.456$

From figure 2.15;

$\varepsilon = 100\%$

Problem 12.4

A shell-and-tube steam condenser is to be constructed of 18 BWG Admiralty tubes with 7/8-in. O.D. Steam will condense outside these single-pass horizontal tubes at Ts = 60 °C. Cooling water enters each tube at T_i = 17 °C, with a flow rate of m = 0.7 kg/s per tube and leaves at To = 29 °C. The heat transfer coefficient for the condensation of steam is h_s = 4500 W/m^2.K. Steam-side fouling is neglected. The fouling factor for the cooling water is 1.76×10^{-4} m^2.K/W. Assume that the tube wall temperature is 50°C. The tube material is carbon steel.

SOLUTION:

Cold fluid properties at $T_m = \frac{17+29}{2} = 23℃$:

$$\rho_c = 997.45\ {kg}/{m^3}$$
$$\mu_c = 9.4\ x\ 10^{-4}\ {kg}/{ms}$$
$$k_c = 0.603\ {W}/{mK}$$
$$c_{p_c} = 4180\ {J}/{kgK}$$
$$Pr_c = 6.52$$

For 18 BWG and O.D. 7/8 in tubes at Table 9.1:

$$O.D. = {7}/{8}\ in = 0.02223\ m$$
$$I.D. = 0.777\ in = 0.01974\ m$$
$$A_c = 0.4742\ in^2 = 3.0593\ x\ 10^{-4}\ m^2$$
$$k_w = 42\ {W}/{m.K}$$

Velocity of cooling water in tube:

$$u = \frac{\dot{m}_c}{\rho A_c} = \frac{0.7}{997.45\ x\ 3.0593\ x\ 10^{-4}} = 2.29\ {m}/{s}$$
$$Re = \frac{u_c d_i \rho_c}{\mu_c} = \frac{2.29\ x\ 997.45\ x\ 0.01974}{9.4\ x\ 10^{-4}} = 47967.4$$

Flow is turbulent so Nusselt number can be calculated from Table 3.3 by using Petukhov correlation:

$$Nu = \frac{\left({f}/{2}\right) x\ Re\ x\ Pr}{1.07 + 12.7\left({f}/{2}\right)^{1/2}\left(Pr^{2/3} - 1\right)}$$
$$f = (1.58 lnRe - 3.28)^{-2} = (1.58\ x \ln 47967.4 - 3.28)^{-2} = 5.29\ x\ 10^{-3}$$
$$Nu = \frac{\left({5.29\ x\ 10^{-3}}/{2}\right) x\ 47967.4\ x\ 6.52}{1.07 + 12.7\left({5.29\ x\ 10^{-3}}/{2}\right)^{1/2}\left(6.52^{2/3} - 1\right)} = 306.8$$
$$h_{tube} = \frac{Nu\ x\ k_c}{d_i} = \frac{306.8\ x\ 0.603}{0.01974} = 9371.85\frac{W}{m^2K}$$

The overall heat transfer coefficient:

$$U = 1790.3\frac{W}{m^2K}$$

Assume that condenser will be designed without subcooling:

$$\Delta T_{in} = 60 - 17 = 43℃$$

$$\Delta T_{out} = 60 - 29 = 31℃$$

LMTD can be calculated as follows:

$$\Delta T_{lm} = \frac{\Delta T_{in} - \Delta T_{out}}{\ln\left(\Delta T_{in}/\Delta T_{out}\right)} = \frac{43 - 31}{\ln\left(\frac{43}{31}\right)} = 36.7℃$$

Assume that heat capacity of condenser is 100 kW:

$$A_o = \frac{Q}{U\Delta T_{lm}} = \frac{100 \ x \ 10^3}{1790.3 \ x \ 36.7} = 1.52 \ m^2$$

Number of tubes can be calculated as follows:

$$\dot{Q} = N_t \dot{m}(T_{out} - T_{in})c_{p_c}$$

$$N_t = 2.85 \approx 3 \ tubes$$

Hence, length of the tubes can be calculated from A_o:

$$A_o = N_t d_o \pi L$$

$$L = \frac{1.52}{3 \ x \ \pi \ x \ 0.02223} = 7.25 \ m$$

Shell diameter can be calculated as follows:

$$D_s = 0.637\sqrt{\frac{CL}{CPT}}\left[\frac{A_o (PR)^2 d_o}{L}\right]^{1/2}$$

For single pass CPT = 0.93

Select tube layout as 90° and CL becomes 1 (CL=1)

Assume that tube pitch ratio as 1.5 (PR=1.5)

$$D_s = 0.637\sqrt{\frac{1}{0.93}}\left[\frac{1.52 \ x \ (1.5)^2 \ x \ 0.02223}{7.25}\right]^{1/2} = 0.0676 \ m$$

Total pressure drop on the tube side;

$$\Delta p_t = 4f\frac{LN_p}{D_s}\frac{G^2}{2\rho}$$

$$f = 0.046Re^{-0.2} = 0.046 \ x \ (47967.4)^{-0.2} = 5.33 \ x \ 10^{-3}$$

$$G = u_c \ x \ \rho = 2.29 \ x \ 997.45 = 2284.16 \ {}^{kg}/_{m^2 s}$$

$$\Delta p_t = 4 \ x \ 5.33 \ x \ 10^{-3} \ x \ \frac{7.25}{0.01974} \ x \ \frac{2284.16^2}{2 \ x \ 997.45} = 20479 \ Pa$$

Pump Power:

$$P = \frac{\dot{m}\Delta p_{total}}{\rho \eta_p} = \frac{0.7 \ x \ 3 \ x \ 20479}{997.45 \ x \ 0.85} = 50.7 \ W$$

Problem 12.5

(See also Problem 8.2) A shell-and-tube type condenser will be designed for a refrigerant-22 condenser for a refrigeration system that provides a capacity of 100 KW for air-conditioning. The condensing temperature is 47°C at design conditions. The tubes are copper (386 W/m · K) and are 14 mm ID and 16 mm OD. Cooling water enters the condenser tubes at 30°C with a velocity of 1.5 m/s and leaves at 35°C. The condenser will have two tube passes. Inline and triangular pitch arrangements will be considered.

<u>SOLUTION:</u>

Tube geometry:

$$\text{O.D. of the tubes, } d_o = 0.016 \text{ m}$$
$$\text{I.D. of the tubes, } d_i = 0.014 \text{ m}$$
$$\text{Tube wall thermal conductivity, } k = 386 \text{ W/(m.K)}$$

Cooling water properties at 32.5 ^oC:

$$\rho_c = 995 \text{ kg/m}^3$$
$$\mu_c = 7.57\times10^{-4} \text{ kg/m.s}$$
$$c_{p,c} = 4178 \text{ J/kg.K}$$
$$Pr_c = 5.13$$
$$k_c = 0.617 \text{ W/m.K}$$

Properties of the condensed liquid at 47 ^oC, from Appendix B.8:

$$i_{lg} = 158.1 \text{ kJ/kg}$$
$$\rho_l = 1099 \text{ kg/m}^3$$
$$\mu_l = 1.76\times10^{-4} \text{ kg/m.s}$$
$$c_{p,l} = 1372 \text{ J/kg.K}$$
$$Pr_l = 3.14$$
$$k_l = 0.077 \text{ W/m.K}$$

Mass flow rate of wet steam entering the condenser:

$$\dot{m}_s = \frac{W_T}{\Delta i} = \frac{100}{158.1} = 0.63 \text{ kg/s}$$

Fouling resistances chosen from TEMA tables in Chapter 6 are:

$$R_{fi} = 0.000176 \text{ m}^2\text{.K/W}$$
$$R_{fo} = 0.000352 \text{ m}^2\text{.K/W}$$

The cooling water mass flow rate is calculated by:

$$\dot{m}_c = \frac{Q}{(T_{c2}-T_{c1})c_{p,c}} = \frac{100}{5\times4.178}$$
$$= 4.79 \text{ kg/s}$$

Number of tubes:

$$N_T = \frac{4\dot{m}_c}{u_c\rho_c\pi d_i^2} = \frac{4\times4.79}{1.5\times995\times\pi\times0.014^2} = 21 \text{ tubes}$$

Since the condenser is two tube passes, the tube number is rounded to 22 tubes.

To calculate the condensing-side heat transfer coefficient, an estimate of the average number of tubes rows in a vertical column is needed. For two-tube passes, assume the tube rows in vertical column to be 3.

Tube-side Reynolds number:

$$Re = \frac{u_c \rho_c d_i}{\mu_c} = \frac{1.5 \times 995 \times 0.014}{7.57 \times 10^{-4}} = 27602$$

Tube-side heat transfer coefficient:

$$f = (1.58 \ln Re - 3.28)^{-2} = \left(1.58 \times \ln(27602) - 3.28\right)^{-2} = 0.0060$$

$$Nu = \frac{(f/2)\, Re\, Pr}{1.07 + 12.7(f/2)^{1/2}\left(Pr^{2/3} - 1\right)}$$

$$= \frac{(0.0060/2) \times 27602 \times 5.13}{1.07 + 12.7(0.0060/2)^{1/2}\left(5.13^{2/3} - 1\right)}$$

$$= 173.8$$

$$h_i = \frac{Nu \cdot k_c}{d_i} = \frac{173.8 \times 0.617}{0.014} = 7662 \;\; W/\left(m^2 K\right)$$

Assume the condenser is to be designed without subcooling, so that:

$$\Delta T_{in} = (47 - 30) = 17 \;\; ^oC$$

and

$$\Delta T_{out} = (47 - 35) = 12 \;\; ^oC$$

Hence the LMTD is:

$$\Delta T_{lm} = \frac{\Delta T_{in} - \Delta T_{out}}{\ln\left(\Delta T_{in}/\Delta T_{out}\right)} = \frac{17 - 12}{\ln(17/12)} = 14.4 \;\; ^oC$$

The shell-side heat transfer coefficient depends on the local heat flux and hence an iteration are necessary. The overall heat transfer coefficient, U, based on the tube O.D., is given by:

$$\frac{1}{U} = R_t + \frac{1}{h_o}$$

where h_o is the coefficient outside the tubes (shell side) and R_t is the sum of all other thermal resistances given by:

$$R_t = R_{fo} + \left[\frac{1}{h_i} + R_{fi}\right]\frac{d_o}{d_i} + \frac{t_w}{k_w}\frac{d_o}{D_m}$$

where D_m is approximated as:

$$D_m = \frac{d_0 - d_i}{\ln\left(d_o/d_i\right)} = \frac{0.016 - 0.014}{\ln(0.016/0.014)} = 0.015$$

where t_w is the wall thickness.

Hence

$$R_t = 0.000352 + \left[\frac{1}{7662} + 0.000176\right]\frac{0.016}{0.014} + \frac{0.001}{386}\frac{0.016}{0.015}$$

$$= 5.737 \times 10^{-4}$$

and

$$\frac{1}{U} = 5.737 \times 10^{-4} + \frac{1}{h_o}$$

The condensing-side heat transfer coefficient, h_o, may be calculated by the Nusselt method with the Kern correction for condensate inundation, hence

$$h_o = 0.728 \left[\frac{\rho_l^{\,2} g \cdot i_{lg} k_l^{\,3}}{\mu_l \Delta T_w d_o} \right]^{1/4} \frac{1}{N^{1/6}}$$

where ΔT_w is the difference between the saturation temperature and the temperature at the surface of the fouling. Because $\rho_l >> \rho_g$, the above equation has been simplified. By inserting the values, we get:

$$h_o = 0.728 \left[\frac{(1099)^2 (9.81) \cdot (158.1 \times 10^3)(0.077)^3}{\left(1.76 \times 10^{-4}\right) \Delta T_w (0.016)} \right]^{1/4} \frac{1}{3^{1/6}}$$
$$= 2530 \big/ \left(\Delta T_w\right)^{1/4}$$

The temperature difference, ΔT_w, is given by:

$$\Delta T_w = \Delta T - R_t q''$$

where ΔT is the local temperature difference between the streams; R_t is the sum of all other resistances; and q'' is the local heat flux, which is given by:

$$q'' = U \Delta T$$

Hence

$$\Delta T_w = \Delta T\left(1 - R_t U\right) = \Delta T\left(1 - 5.737 \times 10^{-4}\, U\right)$$

Table 12.1.1 and 12.1.2 summarize the results of this iteration for the inlet and outlet of the condenser when ΔT_{in} = 17 ^oC and ΔT_{out} = 12 ^oC, respectively. The initial guess of ΔT_w is 10^{o}C.

Table 12.1.1 Iteration for Overall Coefficient at the Inlet of the Power Condenser

ΔT_w, ^oC	h_o, W/m^2.K	U, W/m^2.K
10	1422	783
11	1389	773
11.06	1387	772
11.07	1387	772
11.07	1387	772

Table 12.1.2 Iteration for Overall Coefficient at the Outlet of the Power Condenser

ΔT_w, ^oC	h_o, W/m^2.K	U, W/m^2.K
10	1422	783
7.77	1515	810
7.65	1521	812
7.64	1521	812

7.64	1521	812

The mean overall heat transfer coefficient can then be determined by taking the average of the inlet and outlet overall heat transfer coefficients:

$$U_m = \frac{1521 + 1387}{2} = 1454 \ \ W/m^2.K$$

The required surface area is therefore calculated as:

$$Q = U_m A_o \Delta T_m$$

$$A_o = \frac{Q}{U_m \Delta T_m} = \frac{100 \times 10^3}{1454 \times 14.4} = 4.78 \quad m^2$$

The required length is found by the following equation:

$$A_o = N_T \pi d_o L$$

$$L = \frac{A_o}{N_T \pi d_o} = \frac{4.78}{22 \times \pi \times 0.016} = 4.32 \ \ m$$

Now that the length of the power condenser has been determined, the shell size of the condenser, the tube-side pressure drop, and the tube-side pumping power can all the calculated.

The expression of the shell diameter as a function of heat transfer area, A_o; tube length, L; and tube layout dimensions, P_T, PR, and d_o as parameters can be estimated from equation:

$$D_s = 0.637 \sqrt{\frac{CL}{CPT}} \left[\frac{A_o (PR)^2 d_o}{L} \right]^{1/2}$$

where CL is the tube layout constant, CL = 1.00 for 90^o and 45^o; and CL = 0.87 for 30^o and 60^o.

CPT accounts fro the incomplete coverage of the shell diameter by the tubes:

CPT = 0.93 for 1-tube pass

CPT = 0.9 for 2-tube pass

CPT = 0.85 for 3-tube pass

P_T (the tube pitch) is 0.0381 m.

PR (the tube pitch ratio) is P_T/d_o = 1.5, P_T = 1.5×0.016 = 0.024

$$D_s = 0.637 \sqrt{\frac{0.87}{0.90}} \left[\frac{4.78 \times (1.5)^2 \times 0.016}{4.32} \right]^{1/2} = 0.125 \ \ m$$

The pressure drop on the tube side is calculated as:

$$\Delta p_{total} = \Delta p_t + \Delta p_r$$

The pressure drop through the tubes:

$$\Delta p_t = 4f \frac{L N_p}{D_e} \frac{G^2}{2\rho}$$

where

$$f = 0.046 \, Re^{-0.2} = 0.046 \times 27602^{-0.2} = 0.0060$$

$$G = u_c \rho$$

Therefore,

$$\Delta p_t = 4 \times 0.0060 \times \frac{4.32 \times 2}{0.014} \times \frac{(1.5 \times 995)^2}{12 \times 995}$$
$$= 16580 \text{ Pa}$$

The pressure drop due to the return is given by:

$$\Delta p_r = 4N_p \frac{G^2}{2\rho} = 4 \times 2 \times \frac{(1.5 \times 995)^2}{2 \times 995} = 8955 \text{ Pa}$$

Therefore, the total pressure drop on the tube side is determined by:

$$\Delta p_{total} = \Delta p_t + \Delta p_r$$
$$= 16580 + 8955$$
$$= 25535 \text{ Pa}$$

The pumping power is proportional to the pressure drop across the condenser:

$$P = \frac{\dot{m}\Delta p_{tot}}{\rho \eta_p} = \frac{4.79 \times 25535}{995 \times 0.85} = 144 \quad \text{W}$$

The pumping power is proportional to the pressure drop across the condenser:

$$P = \frac{\dot{m}\Delta p_{tot}}{\rho \eta_p} = \frac{4.79 \times 25535}{995 \times 0.85} = 144 \quad \text{W}$$

Problem 12.6

Repeat Problem 12.5 for the refrigerant-134A.

SOLUTION:

Tube geometry:

O.D. of the tubes, $d_o = 0.016$ m

I.D. of the tubes, $d_i = 0.014$ m

Tube wall thermal conductivity, $k = 386$ W/(m.K)

Cooling water properties at 32.5 °C:

$\rho_c = 995 \ \ kg/m^3$

$\mu_c = 7.57 \times 10^{-4} \ \ kg/m.s$

$c_{p,c} = 4178 \ \ J/kg.K$

$Pr_c = 5.13$

$k_c = 0.617 \ \ W/m.K$

Properties of the condensed liquid at 47 °C, from Appendix B.8:

$i_{lg} = 154.6 \ \ kJ/kg$

$\rho_l = 1117 \ \ kg/m^3$

$\mu_l = 1.72 \times 10^{-4} \ \ kg/m.s$

$c_{p,l} = 1559 \ \ J/kg.K$

$Pr_l = 3.98$

$k_l = 0.068 \ \ W/m.K$

Mass flow rate of wet steam entering the condenser:

$$\dot{m}_s = \frac{W_T}{\Delta i} = \frac{100}{154.6} = 0.65 \ \ kg/s$$

Fouling resistances chosen from TEMA tables in Chapter 6 are:

$R_{fi} = 0.000176 \ \ m^2.K/W$

$R_{fo} = 0.000352 \ \ m^2.K/W$

The cooling water mass flow rate is calculated by:

$$\dot{m}_c = \frac{Q}{(T_{c2} - T_{c1})c_{p,c}} = \frac{100}{5 \times 4.178}$$

$$= 4.79 \ \ kg/s$$

Number of tubes:

$$N_T = \frac{4\dot{m}_c}{u_c \rho_c \pi d_i^2} = \frac{4 \times 4.79}{1.5 \times 995 \times \pi \times 0.014^2} = 21 \text{ tubes}$$

Since the condenser is two tube passes, the tube number is rounded to 22 tubes.

To calculate the condensing-side heat transfer coefficient, an estimate of the average number of tubes rows in a vertical column is needed. For two-tube passes, assume the tube rows in vertical column to be 3.

Tube-side Reynolds number:

$$\text{Re} = \frac{u_c \rho_c d_i}{\mu_c} = \frac{1.5 \times 995 \times 0.014}{7.57 \times 10^{-4}} = 27602$$

Tube-side heat transfer coefficient:

$$f = (1.58 \ln \text{Re} - 3.28)^{-2} = \left(1.58 \times \ln(27602) - 3.28\right)^{-2} = 0.0060$$

$$\text{Nu} = \frac{(f/2)\,\text{Re}\,\text{Pr}}{1.07 + 12.7(f/2)^{1/2}\left(\text{Pr}^{2/3} - 1\right)}$$

$$= \frac{(0.0060/2) \times 27602 \times 5.13}{1.07 + 12.7(0.0060/2)^{1/2}\left(5.13^{2/3} - 1\right)}$$

$$= 173.8$$

$$h_i = \frac{\text{Nu} \cdot k_c}{d_i} = \frac{173.8 \times 0.617}{0.014} = 7662 \ \ \text{W}/\left(\text{m}^2\text{K}\right)$$

Assume the condenser is to be designed without subcooling, so that:

$$\Delta T_{in} = (47 - 30) = 17 \ \ ^{o}\text{C}$$

and

$$\Delta T_{out} = (47 - 35) = 12 \ \ ^{o}\text{C}$$

Hence the LMTD is:

$$\Delta T_{lm} = \frac{\Delta T_{in} - \Delta T_{out}}{\ln\left(\Delta T_{in}/\Delta T_{out}\right)} = \frac{17 - 12}{\ln(17/12)} = 14.4 \ \ ^{o}\text{C}$$

The shell-side heat transfer coefficient depends on the local heat flux and hence an iteration are necessary. The overall heat transfer coefficient, U, based on the tube O.D., is given by:

$$\frac{1}{U} = R_t + \frac{1}{h_o}$$

where h_o is the coefficient outside the tubes (shell side) and R_t is the sum of all other thermal resistances given by:

$$R_t = R_{fo} + \left[\frac{1}{h_i} + R_{fi}\right]\frac{d_o}{d_i} + \frac{t_w}{k_w}\frac{d_o}{D_m}$$

where D_m is approximated as:

$$D_m = \frac{d_0 - d_i}{\ln(d_o/d_i)} = \frac{0.016 - 0.014}{\ln(0.016/0.014)} = 0.015$$

where t_w is the wall thickness.

Hence

$$R_t = 0.000352 + \left[\frac{1}{7662} + 0.000176\right]\frac{0.016}{0.014} + \frac{0.001}{386}\frac{0.016}{0.015}$$

$$= 5.737 \times 10^{-4}$$

and

$$\frac{1}{U} = 5.737 \times 10^{-4} + \frac{1}{h_o}$$

The condensing-side heat transfer coefficient, h_o, may be calculated by the Nusselt method with the Kern correction for condensate inundation, hence

$$h_o = 0.728\left[\frac{\rho_l{}^2 g\cdot i_{lg} k_l{}^3}{\mu_l \Delta T_w d_o}\right]^{1/4}\frac{1}{N^{1/6}}$$

where ΔT_w is the difference between the saturation temperature and the temperature at the surface of the fouling. Because $\rho_l >> \rho_g$, the above equation has been simplified. By inserting the values, we get:

$$h_o = 0.728\left[\frac{(1117)^2(9.81)\cdot(154.6\times10^3)(0.068)^3}{\left(1.72\times10^{-4}\right)\Delta T_w(0.016)}\right]^{1/4}\frac{1}{3^{1/6}}$$

$$= 2324/\left(\Delta T_w\right)^{1/4}$$

The temperature difference, $\Delta T_{w,}$ is given by:

$$\Delta T_w = \Delta T - R_t q''$$

where ΔT is the local temperature difference between the streams; R_t is the sum of all other resistances; and q'' is the local heat flux, which is given by:

$$q'' = U\Delta T$$

Hence

$$\Delta T_w = \Delta T\left(1 - R_t U\right) = \Delta T\left(1 - 5.737\times10^{-4}\,U\right)$$

Table 12.1.1 and 12.1.2 summarize the results of this iteration for the inlet and outlet of the condenser when ΔT_{in} = 17 ^oC and ΔT_{out} = 12 ^oC, respectively. The initial guess of ΔT_w is 10^oC.

Table 12.1.1 Iteration for Overall Coefficient at the Inlet of the Power Condenser

$\Delta T_{w,}$ ^oC	$h_{o,}$ W/m^2.K	U, W/m^2.K
10	1307	747
11.22	1270	734
11.3	1268	734
11.3	1268	734

Table 12.1.2 Iteration for Overall Coefficient at the Outlet of the Power Condenser

$\Delta T_{w,}$ ^oC	$h_{o,}$ W/m^2.K	U, W/m^2.K
10	1307	747
7.92	1385	771
7.81	1390	773
7.81	1390	773

The mean overall heat transfer coefficient can then be determined by taking the average of the inlet and outlet overall heat transfer coefficients:

$$U_m = \frac{734+773}{2} = 754 \;\; W/m^2.K$$

The required surface area is therefore calculated as:

$$Q = U_m A_o \Delta T_m$$

$$A_o = \frac{Q}{U_m \Delta T_m} = \frac{100 \times 10^3}{754 \times 14.4} = 9.21 \quad m^2$$

The required length is found by the following equation:

$$A_o = N_T \pi d_o L$$

$$L = \frac{A_o}{N_T \pi d_o} = \frac{9.21}{22 \times \pi \times 0.016} = 8.33 \ \ m$$

Now that the length of the power condenser has been determined, the shell size of the condenser, the tube-side pressure drop, and the tube-side pumping power can all the calculated.

The expression of the shell diameter as a function of heat transfer area, A_o; tube length, L; and tube layout dimensions, P_T, PR, and d_o as parameters can be estimated from equation:

$$D_s = 0.637\sqrt{\frac{CL}{CPT}}\left[\frac{A_o (PR)^2 d_o}{L}\right]^{1/2}$$

where CL is the tube layout constant, CL = 1.00 for 90^o and 45^o; and CL = 0.87 for 30^o and 60^o.

CPT accounts fro the incomplete coverage of the shell diameter by the tubes:

CPT = 0.93 for 1-tube pass

CPT = 0.9 for 2-tube pass

CPT = 0.85 for 3-tube pass

P_T (the tube pitch) is 0.0381 m.

PR (the tube pitch ratio) is P_T/d_o = 1.5, P_T = 1.5×0.016 = 0.024

$$D_s = 0.637\sqrt{\frac{0.87}{0.90}}\left[\frac{9.21 \times (1.5)^2 \times 0.016}{8.33}\right]^{1/2} = 0.125 \ \ m$$

The pressure drop on the tube side is calculated as:

$$\Delta p_{total} = \Delta p_t + \Delta p_r$$

The pressure drop through the tubes:

$$\Delta p_t = 4f \frac{LN_p}{D_e}\frac{G^2}{2\rho}$$

where

$$f = 0.046\,Re^{-0.2} = 0.046 \times 27602^{-0.2} = 0.0060$$

$$G = u_c \rho$$

Therefore,

$$\Delta p_t = 4 \times 0.0060 \times \frac{4.32 \times 2}{0.014} \times \frac{(1.5 \times 995)^2}{12 \times 995}$$
$$= 16580 \ \ Pa$$

The pressure drop due to the return is given by:

$$\Delta p_r = 4N_p \frac{G^2}{2\rho} = 4 \times 2 \times \frac{(1.5 \times 995)^2}{2 \times 995} = 8955 \ \ Pa$$

Therefore, the total pressure drop on the tube side is determined by:

$$\Delta p_{total} = \Delta p_t + \Delta p_r$$
$$= 16580 + 8955$$
$$= 25535 \text{ Pa}$$

The pumping power is proportional to the pressure drop across the condenser:

$$P = \frac{\dot{m}\Delta p_{tot}}{\rho\eta_p} = \frac{4.79 \times 25535}{995 \times 0.85} = 144 \quad \text{W}$$

The pumping power is proportional to the pressure drop across the condenser:

$$P = \frac{\dot{m}\Delta p_{tot}}{\rho\eta_p} = \frac{4.79 \times 25535}{995 \times 0.85} = 144 \quad \text{W}$$

Problem 12.7

A shell-and-tube type condenser will be designed for a refrigerant-22 condenser for a refrigeration system that provides a capacity of 100 KW for air conditioning. The condensing temperature is 47°C at design conditions. The tubes are copper and are 14 mm ID and 16 mm OD. Cooling water enters the condenser tubes at 30°C with a velocity of 1.5 m/s and leaves at 35°C. The condenser will have two tube passes.

SOLUTION:

Tube geometry:

$$\text{O.D. of the tubes, } d_o = 0.016 \text{ m}$$

$$\text{I.D. of the tubes, } d_i = 0.014 \text{ m}$$

$$\text{Tube wall thermal conductivity, } k = 386 \text{ W/(m.K)}$$

Cooling water properties at 23 ^{o}C:

$$\rho_c = 997 \text{ kg/m}^3$$

$$\mu_c = 9.35 \times 10^{-4} \text{ kg/m.s}$$

$$c_{p,c} = 4180 \text{ J/kg.K}$$

$$Pr_c = 6.49$$

$$k_c = 0.603 \text{ W/m.K}$$

Properties of the condensed liquid at 47 ^{o}C, from Appendix B.8:

$$i_{lg} = 154.6 \text{ kJ/kg}$$

$$\rho_l = 1117 \text{ kg/m}^3$$

$$\mu_l = 1.72 \times 10^{-4} \text{ kg/m.s}$$

$$c_{p,l} = 1559 \text{ J/kg.K}$$

$$Pr_l = 3.98$$

$$k_l = 0.068 \text{ W/m.K}$$

Mass flow rate of wet steam entering the condenser:

$$\dot{m}_s = \frac{W_T}{\Delta i} = \frac{100}{154.6} = 0.65 \text{ kg/s}$$

Fouling resistances chosen from TEMA tables in Chapter 6 are:

$$R_{fi} = 0.000176 \text{ m}^2.\text{K/W}$$

$$R_{fo} = 0.000352 \text{ m}^2.\text{K/W}$$

The cooling water mass flow rate is calculated by:

$$\dot{m}_c = \frac{Q}{(T_{c2} - T_{c1}) c_{p,c}} = \frac{100}{5 \times 4.180}$$

$$= 4.78 \text{ kg/s}$$

Number of tubes:

$$N_T = \frac{4\dot{m}_c}{u_c \rho_c \pi d_i^2} = \frac{4 \times 4.78}{2 \times 997 \times \pi \times 0.014^2} = 16 \text{ tubes}$$

To calculate the condensing-side heat transfer coefficient, an estimate of the average number of tubes rows in a vertical column is needed. For two-tube passes, assume the tube rows in vertical column to be 3.

Tube-side Reynolds number:

$$Re = \frac{u_c \rho_c d_i}{\mu_c} = \frac{2 \times 997 \times 0.014}{9.35 \times 10^{-4}} = 29857$$

Tube-side heat transfer coefficient:

$$f = (1.58 \ln Re - 3.28)^{-2} = \left(1.58 \times \ln(29857) - 3.28\right)^{-2} = 0.0059$$

$$Nu = \frac{(f/2)\, Re\, Pr}{1.07 + 12.7 (f/2)^{1/2} \left(Pr^{2/3} - 1\right)}$$

$$= \frac{(0.0059/2) \times 29857 \times 6.49}{1.07 + 12.7 (0.0059/2)^{1/2} \left(6.49^{2/3} - 1\right)}$$

$$= 205.6$$

$$h_i = \frac{Nu \cdot k_c}{d_i} = \frac{205.6 \times 0.603}{0.014} = 8855 \ \ W/\left(m^2 K\right)$$

Assume the condenser is to be designed without subcooling, so that:

$$\Delta T_{in} = (47 - 20) = 27 \ {}^{o}C$$

and

$$\Delta T_{out} = (47 - 26) = 21 \ {}^{o}C$$

Hence the LMTD is:

$$\Delta T_{lm} = \frac{\Delta T_{in} - \Delta T_{out}}{\ln\left(\Delta T_{in}/\Delta T_{out}\right)} = \frac{27 - 21}{\ln(27/21)} = 23.9 \ {}^{o}C$$

The shell-side heat transfer coefficient depends on the local heat flux and hence an iteration are necessary. The overall heat transfer coefficient, U, based on the tube O.D., is given by:

$$\frac{1}{U} = R_t + \frac{1}{h_o}$$

where h_o is the coefficient outside the tubes (shell side) and R_t is the sum of all other thermal resistances given by:

$$R_t = R_{fo} + \left[\frac{1}{h_i} + R_{fi}\right]\frac{d_o}{d_i} + \frac{t_w}{k_w}\frac{d_o}{D_m}$$

where D_m is approximated as:

$$D_m = \frac{d_0 - d_i}{\ln\left(d_o/d_i\right)} = \frac{0.016 - 0.014}{\ln(0.016/0.014)} = 0.015$$

where t_w is the wall thickness.

Hence

$$R_t = 0.000352 + \left[\frac{1}{8855} + 0.000176\right]\frac{0.016}{0.014} + \frac{0.001}{386}\frac{0.016}{0.015}$$

$$= 6.85 \times 10^{-4}$$

and

$$\frac{1}{U} = 6.85 \times 10^{-4} + \frac{1}{h_o}$$

The condensing-side heat transfer coefficient, h_o, may be calculated by the Nusselt method with the Kern correction for condensate inundation, hence

$$h_o = 0.728 \left[\frac{\rho_l{}^2 g \cdot i_{lg} k_l{}^3}{\mu_l \Delta T_w d_o} \right]^{1/4} \frac{1}{N^{1/6}}$$

where ΔT_w is the difference between the saturation temperature and the temperature at the surface of the fouling. Because $\rho_l >> \rho_g$, the above equation has been simplified. By inserting the values, we get:

$$h_o = 0.728 \left[\frac{(1117)^2 (9.81) \cdot (154.6 \times 10^3)(0.068)^3}{\left(1.72 \times 10^{-4}\right) \Delta T_w (0.016)} \right]^{1/4} \frac{1}{3^{1/6}}$$

$$= 2324 \big/ \left(\Delta T_w\right)^{1/4}$$

The temperature difference, $\Delta T_{w,}$ is given by:

$$\Delta T_w = \Delta T - R_t q''$$

where ΔT is the local temperature difference between the streams; R_t is the sum of all other resistances; and q'' is the local heat flux, which is given by:

$$q'' = U \Delta T$$

Hence

$$\Delta T_w = \Delta T\left(1 - R_t U\right) = \Delta T\left(1 - 6.8 \times 10^{-4}\, U\right)$$

Table 12.1.1 and 12.1.2 summarize the results of this iteration for the inlet and outlet of the condenser when ΔT_{in} = 17 ^oC and ΔT_{out} = 12 ^oC, respectively. The initial guess of ΔT_w is 10^oC.

Table 12.1.1 Iteration for Overall Coefficient at the Inlet of the Power Condenser

$\Delta T_{w,}$ ^oC	$h_{o,}$ W/m^2.K	U, W/m^2.K
10	1307	690
11.83	1253	674
12.23	1243	671
12.30	1240	671
12.32	1240	671
12.33	1240	671
12.33	1240	671

Table 12.1.2 Iteration for Overall Coefficient at the Outlet of the Power Condenser

$\Delta T_{w,}$ ^oC	$h_{o,}$ W/m^2.K	U, W/m^2.K
10	1307	690
9.2	1334	697

9.0	1342	699
9.0	1342	699

The mean overall heat transfer coefficient can then be determined by taking the average of the inlet and outlet overall heat transfer coefficients:

$$U_m = \frac{671+699}{2} = 685 \ \ W/m^2.K$$

The required surface area is therefore calculated as:

$$Q = U_m A_o \Delta T_m$$

$$A_o = \frac{Q}{U_m \Delta T_m} = \frac{100\times 10^3}{685\times 23.9} = 6.108 \quad m^2$$

The required length is found by the following equation:

$$A_o = N_T \pi d_o L$$

$$L = \frac{A_o}{N_T \pi d_o} = \frac{6.108}{16\times\pi\times 0.016} = 7.6 \ \ m$$

Now that the length of the power condenser has been determined, the shell size of the condenser, the tube-side pressure drop, and the tube-side pumping power can all the calculated.

The expression of the shell diameter as a function of heat transfer area, A_o; tube length, L; and tube layout dimensions, P_T, PR, and d_o as parameters can be estimated from equation:

$$D_s = 0.637\sqrt{\frac{CL}{CPT}}\left[\frac{A_o (PR)^2 d_o}{L}\right]^{1/2}$$

where CL is the tube layout constant, CL = 1.00 for 90^o and 45^o; and CL = 0.87 for 30^o and 60^o.

CPT accounts fro the incomplete coverage of the shell diameter by the tubes:

CPT = 0.93 for 1-tube pass

CPT = 0.9 for 2-tube pass

CPT = 0.85 for 3-tube pass

P_T (the tube pitch) is 0.0381 m.

PR (the tube pitch ratio) is P_T/d_o = 1.5, P_T = 1.5×0.016 = 0.024

$$D_s = 0.637\sqrt{\frac{0.87}{0.90}}\left[\frac{6.108\times(1.5)^2\times 0.016}{7.6}\right]^{1/2} = 0.106 \ \ m$$

The pressure drop on the tube side is calculated as:

$$\Delta p_{total} = \Delta p_t + \Delta p_r$$

The pressure drop through the tubes:

$$\Delta p_t = 4f\frac{LN_p}{D_e}\frac{G^2}{2\rho}$$

where

$$f = 0.046\,Re^{-0.2} = 0.046\times 29857^{-0.2} = 0.0058$$

$$G = u_c \rho$$

Therefore,

$$\Delta p_t = 4 \times 0.0058 \times \frac{7.6 \times 2}{0.014} \times \frac{(2 \times 997)^2}{12 \times 997}$$
$$= 50226 \ \ \text{Pa}$$

The pressure drop due to the return is given by:

$$\Delta p_r = 4N_p \frac{G^2}{2\rho} = 4 \times 2 \times \frac{\left(2 \times 997\right)^2}{2 \times 997} = 15952 \ \ \text{Pa}$$

Therefore, the total pressure drop on the tube side is determined by:

$$\Delta p_{total} = \Delta p_t + \Delta p_r$$
$$= 50226 + 15952$$
$$= 66178 \ \ \text{Pa}$$

The pumping power is proportional to the pressure drop across the condenser:

$$P = \frac{\dot{m}\Delta p_{tot}}{\rho \eta_p} = \frac{4.78 \times 66178}{997 \times 0.85} = 373 \ \ \text{W}$$

Problem 13.1

Water, c_p = 4182 J/kg· K, at a flow rate of 2 kg/hr is heated from 7°C to 25°C in a double-pipe PPS oil cooler by engine oil, c_p = 2072 J/kg· K, with an inlet temperature of 65°C and a flow rate of 10 kg/hr. The tubes with an outside diameter of 1 cm and the wall thickness is 0.2 cm. Take the thermal conductivity to be 0.3 W/m^2· K and the heat transfer coefficient of water to be 4911 W/m· K, what are the areas required for counterflow.

SOLUTION:

The hot water outlet temperature can be obtained from the heat balance;

$$Q = \left(\dot{m}c_p\right)_h \left(T_{in} - T_{out}\right)_h = \left(\dot{m}c_p\right)_c \left(T_{out} - T_{in}\right)_c$$

$$Q = \frac{2}{3600}(4182)(25-7) = 41.82 \text{ W}$$

$$T_{out,h} = T_{in,h} + \frac{Q}{\left(\dot{m}c_p\right)_h} = 65 + \frac{41.82}{(10/3600)\times 2072} = 57.73°C$$

Properties of unused engine oil at 61.37^{0}C, from Appendix B (Table B.12), are

$$\rho = 863.111 \text{ kg/m}^3$$
$$\mu = 0.07 \text{ kg/(m·s)}$$
$$k = 0.14 \text{ W/(m·K)}$$
$$Pr = 1030.77$$

$$\text{Re} = \frac{u_m d_i \rho}{\mu} = \frac{4\dot{m}_i}{\mu \pi d_i} = \frac{4\times 10}{0.07\,\pi(0.008)} = 9740.63$$

The flow is turbulent.

The heat transfer coefficient inside the tube can be calculated by use of the correlation given for turbulent flow in Chapter 3. The Gnielinski correlation for constant properties, Eq. (3.31), is used here:

$$f = \left(1.58 \ln Re - 3.28\right)^{-2}$$

$$f = \left(1.58 \ln 9740.63 - 3.28\right)^{-2} = 0.11$$

$$Nu_b = \frac{(f/2)(Re - 1000)Pr}{1 + 12.7(f/2)^{1/2}\left(Pr^{2/3} - 1\right)}$$

$$Nu_b = \frac{0.11/2\ (9740.63 - 1000)1030.77}{1 + 12.7\ (0.11/2)^{1/2}\ \left(1030.77^{2/3} - 1\right)} = 1637.4$$

$$h_i = \frac{Nu\,k}{d_i} = \frac{1637.4 \times 0.140}{0.008} = 28562.03\ \mathrm{W/m^2 \cdot K}$$

The overall heat transfer coefficient for the clean PPS surface based on the outside surface area of the tube is

$$U_c = \frac{1}{\dfrac{d_o}{d_i}\dfrac{1}{h_i} + \dfrac{d_o \ln\left(d_o/d_i\right)}{2k} + \dfrac{1}{h_o}}$$

$$U_{c,\,PPS} = \frac{1}{\dfrac{0.01}{0.008}\dfrac{1}{4911} + \dfrac{0.01 \ln\left(0.01/0.008\right)}{2 \times 0.3} + \dfrac{1}{28562.03}} = 525.015$$

and the mean temperature difference is

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln \dfrac{\Delta T_1}{\Delta T_2}} = \frac{(57.73 - 7) - (65 - 25)}{\ln \dfrac{(57.73 - 7)}{(65 - 25)}} = 103.98°\mathrm{C}$$

The surface areas are calculated from Eqs. (6.1).

$$Q = U_c A_o \Delta T_{lm}$$

$$41.82 = 525.015 \times A_{o,\,PPS} \times 103.97$$

$$A_{o,\,PPS} = \frac{41.82}{525.015 \times 103.97}$$

$$A_{o,\,PPS} = 0.766 \quad \text{m}^2$$

Problem 13.2

Refrigerant-22 (R-22) boiling at 250 K flows through a tubular heat exchanger. Tubes are arranged horizontally with an inner diameter of 0.0172 m. The mass velocity of the R-22 is 200 kg/m^2· s. Heat is supplied by condensing the fluid outside at –12°C. The hot fluid-side heat transfer coefficient, h_h, is 5400 W/m^2 · K. The tube is made of PVDF polymer, and fouling is neglected. By the use of both Chen and Shah's methods, calculate the convection heat transfer coefficient for R-22 for various values of quality assuming no nucleate boiling and the heat transfer rate per length.

SOLUTION:

The properties of refrigerant-22, from Appendix A, at 250 K (–23°C) are:

$$\begin{aligned}
P_s &= 217.4 \text{ kPa } (2.174\,\text{bars}) \\
\Delta h_v &= 221900 \text{ J/kg} \\
\rho_v &= 9.6432 \text{ kg/m}^3 \\
\rho_l &= 1360 \text{ kg/m}^3 \\
c_{pl} &= 1122 \text{ J/kg}\cdot\text{K} \\
\text{k}_l &= 0.112 \text{ W/m}\cdot\text{K} \\
\sigma &= 0.0155 \text{ N/m} \\
\mu_{\text{gn}} &= 0.110\times10^{-4} \text{ Pa}\cdot\text{s} \\
\mu_l &= 0.282\times10^{-3} \text{ Pa}\cdot\text{s}
\end{aligned}$$

The Prandtl number can be calculated as follows:

$$Pr = \frac{\mu_l c_{pl}}{k_l} = \frac{\left(0.282\times10^{-3}\right)\left(1122\right)}{0.112} = 2.825$$

The Reynolds number for the liquid phase is

$$Re_{LO} = \frac{Gd_i}{\mu_l} = \frac{\left(200\right)\left(0.0172\right)}{0.000282} = 12.200$$

The Nusselt number at $x = 0$ can be calculated by Gnielinski's correlation:

$$Nu_{LO} = \frac{(f/2)(Re_{LO} - 1000)Pr_l}{1 + 12.7(f/2)^{1/2}(Pr_l^{2/3} - 1)}$$

where

$$f = (1.58 \ln Re_L - 3.28)^{-2}$$

$$f = [1.58 \ln(12200) - 3.28]^{-2}$$

$$Nu_{LO} = \frac{(0.00372)(12200 - 1000)(2.825)}{1 + 12.7(0.00372)^{1/2}(2.825^{2/3} - 1)} = 66.43$$

The heat transfer coefficient is then

$$h_{LO} = \frac{Nu_{LO} k_l}{d_i} = \frac{(66.43)(0.112)}{(0.0172)} = 432.55 \text{ W/m}^2 \cdot \text{K}$$

Using the Shah method,[45,46] define whether the effect of stratification is important or not. For this, calculate the Froude number:

$$Fr_{LO} = \frac{G^2}{\rho_l^2 g d_i} = \frac{(200)^2}{(1360)^2(9.81)(0.0172)} = 0.1282$$

$Fr > 0.04$, therefore stratification effects are negligible and both Shah and Chen's methods may be used.

Chen's method, at $x = 0.05$, employs the Martinelli Parameter:

$$\frac{1}{X_{tt}} = \left(\frac{x}{(1-x)}\right)^{0.9}\left(\frac{\rho_l}{\rho_v}\right)^{0.5}\left(\frac{\mu_v}{\mu_l}\right)^{0.1}$$

$$= \left(\frac{0.05}{0.95}\right)^{0.9}\left(\frac{1360}{9.64}\right)^{0.5}\left(\frac{110}{2820}\right)^{0.1} = 0.6069$$

The enhancement factor can be calculated from Eq. (8.55):

$$F_o = F\left(1 - x_v\right)^{0.8}$$

where F, from Eq. (8.54), is

$$F = 2.35\left(0.213 + \frac{1}{X_{tt}}\right)^{0.736} = 2.031$$

Therefore,

$$F_o = 2.031\left(1 - 0.05\right)^{0.8} = 1.949$$

and h_{cb} can be calculated from Eq. (8.72):

$$h_{cb} = F_o\left(h_{LO}\right) = 1.949\left(432.55\right) = 842.7 \text{ W/m}^2 \cdot \text{K}$$

The overall heat transfer coefficient, with negligible wall resistance and fouling, is

$$U_c = \frac{1}{\dfrac{d_o}{d_i}\dfrac{1}{h_{cb}} + \dfrac{d_o \ln\left(d_o/d_i\right)}{2k_{PVDF}} + \dfrac{1}{h_h}}$$

$$U_c = \frac{1}{\dfrac{0.019}{0.0172}\dfrac{1}{842.7} + \dfrac{d_o \ln\left(d_o/d_i\right)}{2 \times 0.19} + \dfrac{1}{5400}}$$
$$= 273.43 \text{ W/m}^2 \cdot \text{K}$$

To calculate heat flux,

$$Q/L = 2\pi d_o U\left(T_H - T_s\right) = 2 \times \pi \times 0.019 \times 273.43\left(-12 - \left(-23\right)\right) = 838.14 \text{ W/m}$$

Problem 13.3

A heat exchanger is to be designed to heat raw water by the use of condensed water at 67°C and 0.2 bar, which will flow in the shell side with a mass flow rate of 50,000 kg/hr. The heat will be transferred to 30,000 kg/hr of city water coming from a supply at 17°C (c_p = 4184 J/kg · K). A single shell and a single tube pass without baffles is preferable. A fouling resistance of 0.000176 m^2 · K/W is suggested and the surface over design should not be over 35%. A maximum coolant velocity of 1.5 m/s is suggested to prevent erosion. A maximum tube length of 5 m is required because of space limitations. The tube material is PPSU polymer. Raw water will flow inside of 3/4in. straight tubes (19 mm OD with 16 mm ID). Tubes are laid out on a square pitch with a pitch ratio of 1.25. The permissible maximum pressure drop on the shell side is 5.0 psi. The water outlet temperature should not be less than 40°C. Perform the preliminary analysis whether the surface over design acceptable.

SOLUTION:

***Preliminary Analysis* —** The cold water outlet temperature of at least 40°C determines the exchanger configuration to be considered. Heat duty can be calculated from the fully specified cold stream:

$$Q = \left(\dot{m}c_p\right)_c \left(T_{c_2} - T_{c_1}\right)$$

$$Q = \frac{30{,}000}{3600} \times 4179\left(40 - 17\right) = 801 \text{ kW}$$

The hot water outlet temperature becomes

$$T_{h_2} = T_{h_1} - \frac{Q}{\left(\dot{m}c_p\right)_h} = 67 - \frac{801 \times 10^3}{\frac{50{,}000}{3600} \times 4184} = 53.2°\text{C}$$

First, we have to estimate the individual heat transfer coefficients from Table 9.4. We can assume the shell side heat transfer coefficient and the tube side heat transfer coefficient as 5000 W/m^2 · K and 5000 W/m^2 · K, respectively. Assuming bare tubes, one can estimate the overall heat transfer coefficient from Eq. (9.2) as

$$\frac{1}{U_f} = \frac{1}{h_o} + \frac{r_o}{r_i}\frac{1}{h_i} + R_{ft} + r_o \frac{\ln(r_o/r_i)}{k}$$

$$U_f = \left[\frac{1}{5,000} + \frac{19}{16}\frac{1}{5,000} + 0.000176 + \frac{0.019}{2}\frac{\ln(19/16)}{0.35}\right]^{-1} = 378.89 \ \text{W/m}^2 \cdot \text{K}$$

where k_{ppsu} = 0.35 W/mK and

$$\frac{1}{U_c} = \frac{1}{h_o} + \frac{r_o}{r_i}\frac{1}{h_i} + r_o \frac{\ln(r_o/r_i)}{k}$$

$$U_c = \left[\frac{1}{5,000} + \frac{19}{16}\frac{1}{5,000} + \frac{0.019}{2}\frac{\ln(19/16)}{0.35}\right]^{-1} = 405.96 \ \text{W/m}^2 \cdot \text{K}$$

We need to calculate ΔT_m from the four inlet and outlet temperatures.

$$\Delta T_{lm,cf} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = \frac{27 - 36.2}{\ln\left(\frac{27}{36.2}\right)} = 31.4°\text{C}$$

Assuming F = 0.90, then

$$\Delta T_m = 0.90\,\Delta T_{lm,cf} = 0.90 \times 31.4 \approx 28°\text{C}$$

Next we can estimate the required areas A_f and A_c:

$$A_f = \frac{Q}{U_f \Delta T_m} = \frac{801.93 \times 10^3}{378.89 \times 28} = 75.50 \ \text{m}^2$$

$$A_c = \frac{Q}{U_c \Delta T_m} = \frac{801.93 \times 10^3}{405.96 \times 28} = 70.47 \ \text{m}^2$$

The surface over design is A_f/A_c = 1.07 (7%), which is acceptable.

Problem 13.4

Air at 2 atm and 500 K with a velocity of u_∞ = 20 m/s flows across a PVDF compact heat exchanger matrix studied by Cheng and Van Der Geld[44] having the configuration shown in Figure 13.5. By assuming that the PVDF heat exchange surface is similar to the surface 9.68-0.870 and using the length of the matrix as 48 plates with 4 mm per plate, calculate the heat transfer coefficient and the frictional pressure drop.

<u>SOLUTION:</u>

For air at 500 K and 2 atm, from the appendix,

$$\rho = 1.41 \text{ kg/m}^3$$
$$c_p = 1030 \text{ J/kg}\cdot\text{K}$$
$$\mu = 2.69\times10^{-5} \text{ kg/m}\cdot\text{s}$$
$$Pr = 0.718$$

The mass flux G is

$$G = \frac{\dot{m}}{A_{\min}} = \frac{\rho u_\infty A_{fr}}{A_{\min}} = \frac{\rho u_\infty}{\sigma} = \frac{1.41\times 20}{0.78} = 36.15 \text{ kg/m}^2\cdot\text{s}$$

From Figure 13.5, the hydraulic diameter can be obtained as

$$D_h = \frac{4A_c}{P_w} = \frac{4\times 0.00147\times 0.00137}{(0.00147\times 2)+(0.00137\times 2)} = 0.00142 \text{ m}$$

The Reynolds number becomes

$$Re = \frac{GD_h}{\mu} = \frac{36.15\times 0.00142}{2.69\times10^{-5}} = 1732.07$$

From Figure 10.7, for *Re* = 1732.07, we get

$$\frac{h}{Gc_p}\cdot Pr^{2/3} = 0.0045$$

$$h = 0.0045\frac{Gc_p}{\mathrm{Pr}^{2/3}}$$

$$h = 0.0045\times\frac{36.15\times 1030}{(0.718)^{2/3}} = 208.97\ \mathrm{W/m^2\cdot K}$$

$$\Delta p_f = f\,\frac{A_t}{A_{\min}}\frac{\rho_i}{\rho}\frac{G^2}{2\rho_i}$$

$$\frac{A_t}{A_{\min}} = \frac{4\times L}{D_h} = \frac{4\times\left(48\times 4\times 10^{-3}\right)}{0.00142} = 541.52$$

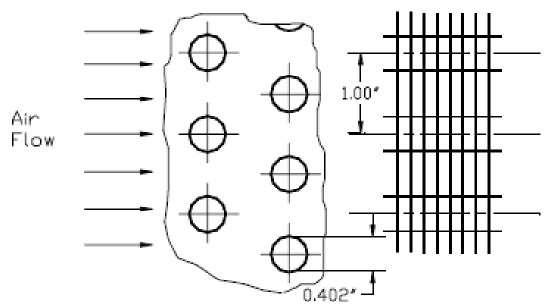
For *Re* = 1732.07, from Figure 10.7, *f* = 0.0275 and $\rho_i/\rho_o \approx 1$:

$$\Delta p_f = 0.0275\times 541.52\times\frac{(36.15)^2}{2\times 1.41} = 6901\ \mathrm{N/m^2}$$

Problem 13.5

Air enters the core of a finned tube PTFE heat exchanger of the type shown in Figure 10.4 at 2 atm and 150°C. The air mass flow rate is 10 kg/s and flows perpendicular to the tubes. The core is 0.5 m long with a 0.30 m^2 frontal area. The height of the core is 0.5 m. Water at 15°C and at a flow rate of 50 kg/s flows inside the tubes. Airside data are given on Figure 10.4. For water side data, assume that σ_w = 0.129, D_h = 0.373 cm, and waterside heat transfer area/total volume = 138 m^2/m^3. When consider this problem as a rating problem, calculate:
a. The airside and waterside heat transfer coefficients
b. Overall heat transfer coefficient based on the outer (airside) surface area
c. Total heat transfer rate and outlet temperatures of air and water

SOLUTION:



Fin thickness, t = 0.033 cm
Fin Area/ Total area = 0.839
Fin Length L:

$$L = \frac{(1.00 - 0.402)}{2} = 0.299'' = 0.7595 \text{ cm}$$

Air passage hydraulic diameter, D_h=0.3633 m
Free flow area/Frontal area, σ = 0.534
Heat transfer area/ Total volume = 587 m^2/m^3

For the air side, we assume a mean temperature 80°C and determine the physical properties of air at p_a = 2 atm

$$\rho_a = 1.9816 \text{ kg/m}^3$$
$$\mu_a = 2.1\times10^{-5} \text{ Pa}\cdot\text{s}$$
$$c_{p,a} = 1009 \text{ kJ/(kg}\cdot\text{K)}$$
$$Pr_a = 0.708$$

Assuming an average temperature of 15°C for the water, we take the physical properties as:

$$\mu_w = 11.54\times10^{-4} \text{ Pa}\cdot\text{s}$$
$$c_{p,w} = 4.186 \text{ kJ/(kg}\cdot\text{K)}$$
$$Pr_w = 8.20$$
$$k_w = 0.591 \text{ W/(m}\cdot\text{K)}$$

(a) Heat transfer coefficient:

Air side:

$$G_a = \frac{\dot{m}_a}{A_{\min}} = \frac{\dot{m}_a}{\sigma_a A_{\text{fr}}} = \frac{10}{0.534 \times 0.30} = 62.42\ \text{kg}\Big/\left(\text{m}^2 \cdot \text{s}\right)$$

$$Re = \frac{G_a D_{h,a}}{\mu_a} = \frac{62.42 \times 0.003633}{2.1 \times 10^{-5}} = 10799$$

From Fig. 10.4, for Re = 10799, we have:

$$\frac{h_a}{G_a c_{p,a}} Pr_a^{2/3} = 0.004$$

$$h_a = 0.004 \times \frac{G_a c_{p,a}}{Pr_a^{2/3}} = 0.004 \times \frac{62.42 \times 1009}{0.708^{2/3}} = 317.1\ \text{W}\Big/\left(\text{m}^2 \cdot \text{K}\right)$$

Water side:

$$A_{\text{fr,w}} = \frac{0.3}{0.5} \times 0.5 = 0.3\ \text{m}^2$$

$$G_w = \frac{\dot{m}_w}{A_{\min}} = \frac{\dot{m}_w}{\sigma_w A_{\text{fr,w}}} = \frac{50}{0.129 \times 0.25} = 1550.39\ \text{kg}\Big/\left(\text{m}^2 \cdot \text{s}\right)$$

$$Re_w = \frac{G_w D_{h,w}}{\mu_w} = \frac{1292 \times 0.003733}{11.54 \times 10^{-5}} = 5011$$

Using Gnielinski's correlation, we have:

$$f = \left(1.58 \ln Re - 3.28\right)^{-2} = 0.00965$$

$$Nu = \frac{\left(f/2\right)\left(Re - 1000\right) Pr}{1 + 12.7\left(f/2\right)^{1/2}\left(Pr^{2/3} - 1\right)} = \frac{\left(0.01022/2\right)\left(4179 - 1000\right) 8.2}{1 + 12.7\left(0.01022/2\right)^{1/2}\left(8.2^{2/3} - 1\right)} = 37$$

So

$$h_h = \frac{k_w Nu}{D_e} = \frac{0.591 \times 37}{0.00373} = 5862.5\ \text{W/m}^2 \cdot \text{K}$$

(b) Overall heat transfer coefficient U_a:

To determine U_{air}, based on the air-side surface, the fin efficiency η should be determined. The reason is that the effective temperature difference between the fluid and the fin surface is lower than that between the fluid and the fin base. Only the air side has extended surfaces which can be regarded as plate fins, the thermal conductivity of fin material, k is given as:

$$k = 0.27\ \text{W/m} \cdot \text{K} \quad \text{for PTFE}$$

The fin efficiency η for a plate fin can be calculated as:

$$\eta_f = \frac{\tanh(\text{mL})}{(\text{mL})}$$

Where

$$\text{mL} = \sqrt{\frac{2 \times 2317.1}{0.27 \times 0.033 \times 10^{-2}}} \times 0.7595 \times 10^{-2} = 54.77$$

$$\eta_f = \frac{\tanh(\text{mL})}{\text{mL}} = \frac{\tanh(54.77)}{54.77} = 0.018$$

The air-weighted fin efficiency is determined by following Eq.:

$$\eta = \left[1 - \frac{A_f}{A}(1-\eta_f)\right] = \left[1 - 0.839 \times (1 - 0.018)\right] = 0.176$$

The overall heat transfer coefficient, U_a, based on the air-side surface, is:

$$\frac{1}{U_a} = \frac{1}{\eta h_a} + \frac{1}{(A_w/A_a)h_w}$$

Here the ration of the water-side to air-side heat transfer surfaces is found to be

$$\frac{A_w}{A_a} = \frac{\text{water - side heat transfer area / total volume}}{\text{air - side heat transfer area / total volume}} = \frac{138}{587} = 0.235$$

Then

$$\frac{1}{U_a} = \frac{1}{0.176 \times 317.1} + \frac{1}{0.235 \times 5862.5}$$

$$U_a = 53.64 \text{ W/m}^2 \cdot \text{K}$$

(c) Outlet temperature of air and water:

$$C_a = \dot{m}_a c_{p,a} = 10 \times 1009 = 10090 \text{ W/K}$$
$$C_w = \dot{m}_w c_{p,w} = 50 \times 4186 = 209300 \text{ W/K}$$
$$C_a = 10090 \text{ W/K} = C_{\min}$$
$$\frac{C_{\min}}{C_{\min}} = \frac{10090}{209300} = 0.048$$

Total volume of matrix, V = A_{fr}×(core length) = 0.3×0.5 = 0.15 m^3

$$\frac{A_a}{V} = 587 \text{ (from Fig. 9.4)}$$
$$A_a = 587 \times V = 587 \times 0.15 = 88.05 \text{ m}^2$$

$$NTU = \frac{A_a U_a}{C_{\min}} = \frac{88.06 \times 53.64}{10090} = 0.47$$

From Fig. 2.15, for a cross-flow heat exchanger with both fluids unmixed, for NTU = 0.47, and C_{min}/C_{max} = 0.048, we obtain:

$$\varepsilon = 0.40$$

Then the total heat transfer rate, Q becomes:

$$\begin{aligned} Q &= \varepsilon C_{\min}\left(T_{in,a} - T_{in,w}\right) \\ &= 0.40 \times 10090 \times (150 - 15) \\ &= 544860 \quad \text{W} \end{aligned}$$

From the heat balance

$$Q = \dot{m}_a c_{p,a}\left(T_{in,a} - T_{out,a}\right) = \dot{m}_w c_{p,w}\left(T_{out,w} - T_{in,w}\right)$$

the outlet temperature can be obtained as:

$$T_{out,a} = T_{in,a} - \frac{Q}{\dot{m}_a c_{p,a}} = 150 - \frac{544860}{10090} = 96°\text{C}$$
$$T_{out,w} = T_{in,w} + \frac{Q}{\dot{m}_w c_{p,w}} = 15 + \frac{544860}{209300} = 17.6°\text{C}$$

Problem A.1: Additional problem for Chapter 11

A gasketed-plate heat exchanger will be used for heating city water (R_{fc} = 0.0006 m^2.K/W) using the wastewater available at 90 ^oC. The vertical distance between the ports of the plate is 1.60 m and the width of the plate is 0.50 m with a gap of 6 mm between the plates. The enhancement factor is provided by the manufacturer as 1.17 and the Chevron angle is 60^o. The plates are made of titanium (k = 20 W/m.K) with a thickness of 0.0006m. The port diameter is 0.15 m. The cold water enters to the plate heat exchanger at 15 ^oC and leaves at 45 ^oC at a rate of 6 kg/s; and it will be heated by the hot water available at 90 ^oC, flowing at a rate of 12 kg/s. By considering single-pass arrangements for both streams, calculate a, b with using correlation proposed by Kumar and correlation proposed by Muley and Manglik than compare these results:

a. The effective surface area and the number of plates of this heat exchangers

b. The pressure drop for both streams

GIVEN:

-Inlet temperature of hot wastewater (T_{h1}) = 90 ^oC
-Mass flow rate of hot water ($\dot{m}_h$) = 12 kg/s
-Inlet temperature of cold water (T_{c1}) = 15 ^oC
-Outlet temperature of cold water (T_{c2}) = 45 ^oC
-Mass flow rate of cold water ($\dot{m}_c$) = 6 kg/s
-Fouling resistance (R_{fc}) = 0.00006 m^2.K/W
-Plate thermal conductivity (Titanium , k_w) = 20 W/(m.K)
-Gasket heat exchanger configuration:

L_v = 1.60 m L_w = 0.50 m
b = 6 mm ϕ = 1.17
$\beta = 60^o$ t = 0.0006 m
D_p = 0.15 m

SOLUTION:

Cold water properties at $T_m = \dfrac{T_{in} + T_{out}}{2} = \dfrac{15 + 45}{2} = 30\ ^oC$:

$$\rho = 996\ \text{kg/m}^3$$
$$\mu = 8.15\times10^{-4}\ \text{kg/m.s}$$
$$c_p = 4179\ \text{J/kg.K}$$
$$Pr = 5.58$$
$$k = 0.612\ \text{W/(m.K)}$$

Hot water properties at T_{h1} = 90 ^oC:

$$\rho = 965\ \text{kg/m}^3$$
$$\mu = 3.16\times10^{-4}\ \text{kg/m.s}$$
$$c_p = 4205\ \text{J/kg.K}$$
$$Pr = 1.96$$
$$k = 0.675\ \text{W/(m.K)}$$

a.

Heat duty:

$$Q = (\dot{m}c_p)_c (T_{c2} - T_{c1}) = (\dot{m}c_p)_h (T_{h1} - T_{h2})$$

$$= 6 \times 4179 \times (45 - 15) = 752220 \text{ W}$$

$$T_{h2} = T_{h1} - \frac{(\dot{m}c_p)_c (T_{c2} - T_{c1})}{(\dot{m}c_p)_h} = 90 - \frac{6 \times 4179 \times (45 - 15)}{12 \times 4205} = 75 \ ^{o}\text{C}$$

Log-mean temperature difference:

$$\Delta T_{lm,cf} = \frac{(T_{h1} - T_{c2}) - (T_{h2} - T_{c1})}{\ln \frac{(T_{h1} - T_{c2})}{(T_{h2} - T_{c1})}} = \frac{(90 - 45) - (75 - 15)}{\ln \frac{90 - 45}{75 - 15}} = 52.14 \ ^{o}\text{C}$$

To solve for the number of plates needed, a trial-and-error method has to be used. First, an estimated heat transfer coefficient is assumed, which is verified and iterated later until the assumed value matches the value resulted from it.

Assume the overall heat transfer coefficient U_c to be 5000 W/(m^2.K) (From table 11.2).

For safety factor C_s = 1.4,

$$Q_f = C_s \cdot Q_r = 1.4 \times 752220 = 1053108 \text{ W}$$

where $Q_f = U_f \cdot A_e F \Delta T_{lm,cf}$, then:

$$A_e = \frac{Q_f}{U_f \cdot F \Delta T_{lm,cf}} = \frac{1053108}{5000 \times 1 \times 52.14} = 4.0395 \text{ m}^2$$

The single-plate projected area:

$$A_{1p} = L_w \cdot L_p = (L_v - D_p) \cdot L_w = (1.6 - 0.15)(0.50) = 0.725 \text{ m}^2$$

The single-plate heat transfer area:

$$A_1 = A_{1p} \cdot \phi = 0.725 \times 1.17 = 0.8483 \text{ m}^2$$

The number of plates:

$$N_e = \frac{A_e}{A_1} = \frac{4.0395}{0.8483} \approx 5 \text{ plates}$$

$$\therefore N_t = N_e + 2 = 5 + 2 = 7 \text{ plates}$$

Verify the assumed heat transfer coefficient:

One channel flow area:

$$A_{ch} = b \times L_w = 0.006 \times 0.5 = 0.003 \text{ m}^2$$

Channel equivalent diameter:

$$D_e = \frac{4(b \cdot L_w)}{2(b + L_w \phi)} = \frac{2b}{\phi} = \frac{2 \times 0.006}{1.17} = 0.01026 \text{ m}$$

The number of channels per pass, N_{cp}:

$$N_{cp} = \frac{N_t - 1}{2 \cdot N_p} = \frac{7 - 1}{2 \times 1} = 3$$

Mass flow rate per channel:

$$\dot{m}_{ch,h} = \frac{12}{3} = 4 \quad \text{kg/s}$$

$$\dot{m}_{ch,c} = \frac{6}{3} = 2 \quad \text{kg/s}$$

Mass velocity, G_{ch}:

$$G_{ch,h} = \frac{4}{0.003} = 1333.34 \quad \text{kg/(m}^2\text{s)}$$

$$G_{ch,c} = \frac{2}{0.003} = 666.67 \quad \text{kg/(m}^2\text{s)}$$

Hot and cold fluid Reynolds numbers for correlation proposed by Kumar are:

$$Re_h = \frac{G_{ch,h} \cdot D_e}{\mu_h} = \frac{1333.34 \times 0.01026}{3.16 \times 10^{-4}} = 43291$$

$$Re_c = \frac{G_{ch,c} \cdot D_e}{\mu_c} = \frac{666.67 \times 0.01026}{8.15 \times 10^{-4}} = 8393$$

Hot and cold fluid Reynolds numbers for correlation proposed by Muley and Manglik are:

$$Re_h = \frac{G_{ch,h} \cdot 2b}{\mu_h} = \frac{1333.34 \times 0.012}{3.16 \times 10^{-4}} = 50633$$

$$Re_c = \frac{G_{ch,c} \cdot 2b}{\mu_c} = \frac{666.67 \times 0.012}{8.15 \times 10^{-4}} = 9816$$

Heat transfer coefficient :

1) This can be obtained by referring to Table 11.6 C_h=0.108, and n= 0.703 for $\beta = 60\,^o$ with using the correlation proposed by Kumar:

$$Nu_h = \frac{h_h D_e}{k} = 0.108 \cdot (Re_h)^{0.703} (Pr)^{1/3} \left(\frac{\mu_b}{\mu_w}\right)^{0.17}$$

By assuming $\mu_b = \mu_w$, then:

$$Nu_h = 0.108 \cdot (43291)^{0.703} (1.96)^{1/3} = 245.6$$

$$h_h = \frac{Nu_h k}{D_e} = \frac{245.6 \times 0.675}{0.01026} = 16158 \ \text{W/(m}^2 \cdot \text{K)}$$

$$Nu_c = 0.108 \cdot (8393)^{0.703} (5.58)^{1/3} = 109.9$$

$$h_c = \frac{Nu_c k}{D_e} = \frac{109.9 \times 0.612}{0.01026} = 6555.4 \ \text{W/(m}^2 \cdot \text{K)}$$

Overall heat transfer coefficient:

$$\frac{1}{U_c} = \frac{1}{h_c} + \frac{1}{h_h} + \frac{0.0006}{20}$$

$$U_c = \left[\frac{1}{h_c} + \frac{1}{h_h} + \frac{0.0006}{20}\right]^{-1} = \left[\frac{1}{6555.4} + \frac{1}{16158} + \frac{0.0006}{20}\right]^{-1} = 4091 \ \text{W/(m}^2\text{.K)}$$

Fouled overall heat transfer coefficient:

$$\frac{1}{U_f} = \frac{1}{U_c} + 0.00006$$

$$U_f = \left[\frac{1}{U_c} + 0.00006\right]^{-1} = \left[\frac{1}{4091} + 0.00006\right]^{-1} = 3284.7 \ \text{W/(m}^2\text{.K)}$$

So, the resulting heat transfer coeffcient is less than the assumed 5000 W/(m^2.K), the procedure has to be repeated, assuming the U_f to be 3284.7 W/(m^2.K) as follows:

$$A_e = \frac{Q_f}{U_f \cdot F\Delta T_{lm,cf}} = \frac{1053108}{3284.7 \times 1 \times 52.14} = 6.1489 \ \text{m}^2$$

The number of plates:

$$N_e = \frac{A_e}{A_1} = \frac{6.1489}{0.8483} = 7.25 \approx 9$$ plates (round off to odd number to make equal number of channel passes for cold and hot fluid)

So the number of plates we get is:

$$\therefore N_t = N_e + 2 = 9 + 2 = 11 \ \text{plates}$$

and the effective surface area is:

$$A_e = N_e \cdot A_1 = (11 - 2) \times 0.8483 = 7.6347 \ \text{m}^2$$

However, this problem cannot be solved without iterations. Reynolds numbers were calculated according to mass velocities for 7 plates. But, number of plates is determined to be 11 for those Reynolds numbers. Therefore, iterations should be performed until same number of plates is obtained for consecutive iterations.

2) Heat transfer coefficient can be obtained for $\beta = 60$, $Re_h = 38947$, $Re_c = 7550$ by referring to the correlation proposed by Muley et al. and Muley Manglik (Eq 11.12) :

$$Nu = \frac{2h_h b}{k} = \left[0.2668 - 0.006967\beta + 7.244x10^{-5}\beta^2\right] x \left[20.78 - 50.94\Phi + 41.1\Phi^2 - 10.51\Phi^3\right]$$
$$x(Re)^{[0.728 + 0.0543\sin\{(\pi\beta/45) + 3.7\}]} x (Pr)^{1/3} x \left(\frac{\mu_b}{\mu_w}\right)^{0.17}$$

By assuming $\mu b = \mu w$, then:

$$Nu_h = \left[0.2668 - 0.00696x(60) + 7.244x10^{-5}x(60)^2\right] x \left[20.78 - 50.94x(1.17) + 41.1x(1.17)^2 - 10.51x(1.17)^3\right]$$
$$x(50633)^{[0.728 + 0.0543\sin\{(\pi 60/45) + 3.7\}]} x (1.96)^{1/3} = 240.78$$

$$h_h = \frac{Nu_h k}{2b} = \frac{240.78x0.675}{2x0.006} = 13543.8$$

$$Nuc = \left[0.2668 - 0.00696x(60) + 7.244x10^{-5}x(60)^2\right]x\left[20.78 - 50.94x(1.17) + 41.1x(1.17)^2 - 10.51x(1.17)^3\right]$$

$$x(9816)^{\left[0.728+0.0543\sin\{(\pi 60/45)+3.7\}\right]}x(5.58)^{1/3} = 102.1$$

$$hc = \frac{Nu_c k}{2b} = \frac{102.1x0.612}{2x0.006} = 5207.6$$

Overall heat transfer coefficient:

$$\frac{1}{U_c} = \frac{1}{h_c} + \frac{1}{h_h} + \frac{0.0006}{20}$$

$$U_c = \left[\frac{1}{h_c} + \frac{1}{h_h} + \frac{0.0006}{20}\right]^{-1} = \left[\frac{1}{5207.6} + \frac{1}{13543.8} + \frac{0.0006}{20}\right]^{-1} = 3380 \text{ W/(m}^2\text{.K)}$$

Fouled overall heat transfer coefficient:

$$\frac{1}{U_f} = \frac{1}{U_c} + 0.00006$$

$$U_f = \left[\frac{1}{U_c} + 0.00006\right]^{-1} = \left[\frac{1}{3380} + 0.00006\right]^{-1} = 2810 \text{ W/(m}^2\text{.K)}$$

So, the heat transfer coefficient is less than the assumed 5000 W/(m^2.K), the procedure has to be repeated, assuming the U_f to be 2810 W/(m^2.K) as follows:

$$A_e = \frac{Q_f}{U_f \cdot F\Delta T_{lm,cf}} = \frac{1053108}{2810 \times 1 \times 52.14} = 7.1878 \text{ m}^2$$

The number of plates:

$$N_e = \frac{A_e}{A_1} = \frac{7.1878}{0.8483} = 8.47 \approx 9$$ plates (round off to odd number to make equal number of channel passes for cold and hot fluid)

So the number of plates we get is:

$$\therefore N_t = N_e + 2 = 9 + 2 = 11 \text{ plates}$$

and the effective surface area is:

$$A_e = N_e \cdot A_1 = (11-2) \times 0.8483 = 7.6347 \text{ m}^2$$

However, this problem cannot be solved without iterations. Reynolds numbers were calculated according to mass velocities for 7 plates. But, number of plates is determined to be 11 for those Reynolds numbers. Therefore, iterations should be performed until same number of plates is obtained for consecutive iterations.

b.

Friction coefficients and Pressure Drop :

1) Friction coefficients can be obtained by referring to Table 11.6 K_p=0.760, and m= 0.215 for $\beta = 60^o$, Re_h=33300 and Re_c=6456;

$$f_h = \frac{0.760}{(Re_h)^{0.215}} = \frac{0.760}{43291^{0.215}} = 0.077$$

$$f_c = \frac{0.760}{(Re_c)^{0.215}} = \frac{0.760}{8393^{0.215}} = 0.109$$

<u>Hot and cold fluid frictional pressure drop:</u>

$$\Delta p_c = 4f \frac{L_{eff} N_p}{D_e} \frac{G_c^{\ 2}}{2\rho} \left(\frac{\mu_b}{\mu_w} \right)^{-0.17}$$

$$(\Delta p_c)_h = 4 \times 0.077 \times \frac{1.6 \times 1}{0.01026} \times \frac{1333.34^2}{2 \times 965} = 44243.3 \text{ Pa}$$

$$(\Delta p_c)_c = 4 \times 0.109 \times \frac{1.6 \times 1}{0.01026} \times \frac{666.67^2}{2 \times 996} = 15170 \text{ Pa}$$

<u>Port mass velocity:</u>

$$G_{p,h} = \frac{\dot{m}}{\pi \left(\frac{D_p^{\ 2}}{4} \right)} = \frac{12}{\pi \times \left(\frac{0.15^2}{4} \right)} = 679.06 \text{ kg/(m}^2\text{s)}$$

$$G_{p,c} = \frac{\dot{m}_c}{\pi \left(\frac{D_p^{\ 2}}{4} \right)} = \frac{6}{\pi \times \left(\frac{0.15^2}{4} \right)} = 339.53 \text{ kg/(m}^2\text{s)}$$

<u>Pressure drop in port ducts:</u>

$$\Delta p_p = 1.4 N_p \frac{G_p^{\ 2}}{2\rho}$$

$$(\Delta p_c)_h = 1.4 \times 1 \times \frac{679.06^2}{2 \times 965} = 334.49 \text{ Pa}$$

$$(\Delta p_c)_c = 1.4 \times 1 \times \frac{339.53^2}{2 \times 996} = 81.02 \text{ Pa}$$

<u>Total pressure drop for hot and cold fluids:</u>

$$\Delta p_t = \Delta p_C + \Delta p_P$$

$$(\Delta p_t)_h = (\Delta p_c)_h + (\Delta p_p)_h = 44243.3 + 334.49 = 44577.8 \text{ Pa}$$

$$(\Delta p_t)_c = (\Delta p_c)_c + (\Delta p_p)_c = 15170 + 81.02 = 15251 \text{ Pa}$$

2) <u>Friction coefficients can also be obtained for β = 60 º for β = 60, Re_h=3e3300 and Re_c=6456 by referring to the correlation proposed by Muley et al. and Muley Manglik (Eq 11.13):</u>

$$f = \left[2.917 - 0.1277\beta + 2.016x10^{-3}\beta^2\right]x\left[5.474 - 19.02\Phi + 18.93\Phi^2 - 5.341\Phi^3\right]x\mathrm{Re}^{-[0.2+0.0557\sin\{(\pi\beta/45)+2.1\}]}$$

$$f_h = \left[2.917 - 0.1277x(60) + 2.016x10^{-3}x(60)^2\right]x\left[5.474 - 19.02x(1.17) + 18.93(1.17)^2 - 5.341x(1.17)^3\right]$$
$$x50633^{-[0.2+0.0557\sin\{(\pi 60/45)+2.1\}]} = 0.156$$

$$f_C = \left[2.917 - 0.1277x(60) + 2.016x10^{-3}x(60)^2\right]x\left[5.474 - 19.02x(1.17) + 18.93(1.17)^2 - 5.341x(1.17)^3\right]$$
$$x9816^{-[0.2+0.0557\sin\{(\pi 60/45)+2.1\}]} = 0.219$$

Muley and Manglik defined the Fanning friction factor as given below :

$$f = \frac{\rho 2b\Delta P}{2LG^2} \rightarrow \Delta P = f\,\frac{L}{b}\,\frac{G^2}{\rho}$$

$$(\Delta p_c)_h = 0.156 \times \frac{1.6}{0.012} \times \frac{1333.34^2}{965} = 38319 \text{ Pa}$$

$$(\Delta p_c)_c = 0.219 \times \frac{1.6}{0.012} \times \frac{666.67^2}{996} = 13030 \text{ Pa}$$

Usage of this correlation does not change the way that pressure drop in port ducts are calculated. Pressure drop in port ducts can be used which have already been calculated above.

Total pressure drop for hot and cold fluids:

$$\Delta p_t = \Delta p_C + \Delta p_P$$
$$(\Delta p_t)_h = (\Delta p_c)_h + (\Delta p_p)_h = 38319 + 334.49 = 38653.5 \text{ Pa}$$
$$(\Delta p_t)_c = (\Delta p_c)_c + (\Delta p_p)_c = 13030 + 81.02 = 13111.02 \text{ Pa}$$

Table 1. Comparison of results for different correlations

	Correlations proposed by Kumar	Correlations proposed by Muley and Manglik
Hot water heat transfer coefficient (h_h)	16158 (W/m^2K)	13543.8 (W/m^2K)
Cold water heat transfer coefficient (h_c)	6555.4(W/m^2K)	5207.6 (W/m^2K)
Overall heat transfer coefficient (U_f)	3284.7 (W/m^2K)	2810 (W/m^2K)
Number of plates	11	11
Pressure drop in hot water due to friction$(\Delta P_c)_h$	44243 (Pa)	38319 (Pa)
Pressure drop in cold water due to friction$(\Delta P_c)_c$	15170 (Pa)	13030 (Pa)
Total pressure drop in hot water$(\Delta P_{total})_h$	44577.8 (Pa)	38653.5 (Pa)
Total pressure drop in cold water$(\Delta P_{total})_c$	15251 (Pa)	13111.02(Pa)

Problem A.2: Additional problem for Chapter 11

A one-pass countercurrent flow heat exchanger has 201 plates. The exchanger has a vertical port distance of 2 m and is 0.6 m wide, with a gap between the plates of 6 mm. The heat exchanger will be used for the following process: cold water from the city supply with an inlet temperature of 10 °C is fed to the heat exchanger at a rate of 15 kg/s that will be heated to 75°C with a wastewater entering at an inlet temperature of 90 °C. The flow rate of hot water is 30 kg/s, which is a distilled water. The other construction parameters are given as in Problem 11.2 but using the enlargement factor as 1.25. There is no limitation on the pressure drop. Is this heat exchanger suitable for this purpose (larger or smaller)?

GIVEN:

-Inlet temperature of hot wastewater (T_{h1}) = 90 °C
-Mass flow rate of hot water ($\dot{m}_h$) = 30 kg/s
-Inlet temperature of cold water (T_{c1}) = 10 °C
-Outlet temperature of cold water (T_{c2}) = 75 °C
-Mass flow rate of cold water ($\dot{m}_c$) = 15 kg/s
-Plate thermal conductivity (Titanium , k_w) = 20 W/(m.K)
-Gasket heat exchanger configuration:

$L_v = 2.0$ m	$L_w = 0.60$ m
b = 6 mm	$\phi = 1.25$
$\beta = 50^o$	t = 0.0006 m
$D_p = 0.15$ m	

SOLUTION:

Cold water properties at $T_m = \dfrac{T_{in} + T_{out}}{2} = \dfrac{10+75}{2} = 42.5\,^oC$:

$$\rho = 991 \ \ kg/m^3$$
$$\mu = 6.24\times10^{-4} \ \ kg/m.s$$
$$c_p = 4179 \ \ J/kg.K$$
$$Pr = 4.14$$
$$k = 0.630 \ \ W/(m.K)$$

Hot water properties at T_{h1} = 90 °C:

$$\rho = 965 \ \ kg/m^3$$
$$\mu = 3.16\times10^{-4} \ \ kg/m.s$$
$$c_p = 4205 \ \ J/kg.K$$
$$Pr = 1.96$$
$$k = 0.675 \ \ W/(m.K)$$

Heat duty:

$$Q = \left(\dot{m}c_p\right)_c\left(T_{c2} - T_{c1}\right) = \left(\dot{m}c_p\right)_h\left(T_{h1} - T_{h2}\right)$$
$$= 15\times4179\times\left(75-10\right) = 4074525 \ \ W$$

$$T_{h2} = T_{h1} - \frac{\left(\dot{m}c_p\right)_c\left(T_{c2} - T_{c1}\right)}{\left(\dot{m}c_p\right)_h} = 90 - \frac{15\times4179\times\left(75-10\right)}{30\times4205} = 57.7 \ \ ^oC$$

Log-mean temperature difference:

$$\Delta T_{lm,cf} = \frac{(T_{h1} - T_{c2}) - (T_{h2} - T_{c1})}{\ln \frac{(T_{h1} - T_{c2})}{(T_{h2} - T_{c1})}} = \frac{(90-75)-(57.7-10)}{\ln \frac{90-75}{57.7-10}} = 21.7 \ ^{o}C$$

The effective number of plate is:

$$N_e = N_t - 2 = 199$$

One channel flow area:

$$A_{ch} = b \times L_w = 0.006 \times 0.60 = 0.0036 \quad m^2$$

The single-plate projected area:

$$A_{1p} = L_w \cdot L_p = (L_v - D_p) \cdot L_w = (2.0 - 0.15) \times 0.6 = 1.11 \ m^2$$

The single-plate heat transfer area:

$$A_1 = A_{1p} \cdot \phi = 1.11 \times 1.25 = 1.3875 \quad m^2$$

Channel equivalent diameter at ϕ = 1.17:

$$D_e = \frac{4(b \cdot L_w)}{2(b + L_w \phi)} = \frac{2b}{\phi} = \frac{2 \times 0.006}{1.25} = 0.0096 \quad m$$

The number of channel per pass:

$$N_{cp} = \frac{N_t - 1}{2N_p} = \frac{201-1}{2 \times 1} = 100$$

Mass flow rate per channel:

$$\dot{m}_{ch,h} = \frac{30}{100} = 0.3 \quad kg/s$$

$$\dot{m}_{ch,c} = \frac{15}{100} = 0.15 \quad kg/s$$

Mass velocity, G_{ch}:

$$G_{ch,h} = \frac{0.3}{0.0036} = 83.33 \quad kg/(m^2 s)$$

$$G_{ch,c} = \frac{0.15}{0.0036} = 41.67 \quad kg/(m^2 s)$$

Hot and cold fluid Reynolds numbers are:

$$Re_h = \frac{G_{ch} \cdot D_e}{\mu_h} = \frac{83.33 \times 0.0096}{3.16 \times 10^{-4}} = 2531.544$$

$$Re_c = \frac{G_{ch} \cdot D_e}{\mu_c} = \frac{41.67 \times 0.01025}{6.24 \times 10^{-4}} = 641.077$$

Hot fluid heat transfer coefficient, h_h—This can be obtained by referring to Table 11.6 C_h=0.13, and n= 0.732 for β = 50 °C:

$$Nu_h = \frac{h_h D_e}{k} = 0.13 \cdot (Re_h)^{0.732} (Pr)^{1/3} \left(\frac{\mu_b}{\mu_w} \right)^{0.17}$$

By assuming $\mu_b = \mu_w$, then:

$$Nu_h = 0.13\cdot\left(2531.6\right)^{0.732}\left(1.96\right)^{1/3} = 50.17$$

$$h_h = \frac{Nu_h k}{D_e} = \frac{50.17\times 0.675}{0.0096} = 3527.34 \text{ W/(m}^2\cdot\text{K)}$$

$$Nu_c = 0.13\cdot\left(641.077\right)^{0.732}\left(4.14\right)^{1/3} = 23.67$$

$$h_c = \frac{Nu_c k}{D_e} = \frac{23.67\times 0.630}{0.0096} = 1553.594 \text{ W/(m}^2\cdot\text{K)}$$

Overall heat transfer coefficient:

$$\frac{1}{U_c} = \frac{1}{h_c} + \frac{1}{h_h} + \frac{0.0006}{20}$$

$$U_c = \left[\frac{1}{h_c} + \frac{1}{h_h} + \frac{0.0006}{20}\right]^{-1} = \left[\frac{1}{3527.34} + \frac{1}{1553} + \frac{0.0006}{20}\right]^{-1} = 1044.5 \text{ W/(m}^2\text{.K)}$$

Fouled overall heat transfer coefficient:

Assume the R_{fc} = 0.00006 (m^2.K/W) for city water, and R_{fh} = 0.000086(m^2.K/W) for the wastewater side.

$$\frac{1}{U_f} = \frac{1}{U_c} + 0.00006 + 0.000086$$

$$U_f = \left[\frac{1}{U_c} + 0.00006 + 0.000086\right]^{-1} = \left[\frac{1}{1044.5} + 0.000146\right]^{-1} = 906.3 \text{ W/(m}^2\text{.K)}$$

The corresponding cleanliness factor is:

$$CF = \frac{U_f}{U_c} = \frac{906.3}{1044.5} = 0.87 \text{ m}^2$$

Actual heat duties for clean surface are:

$$Q_c = U_c A_e \Delta T_m = 1044.5\times 199\times 1.3875\times 28.3 = 8161.706 \text{ kW}$$

$$Q_f = U_f A_e \Delta T_m = 906.3\times 199\times 1.3875\times 28.3 = 7083.376 \text{ kW}$$

The safety factor is:

$$C_s = \frac{Q_f}{Q_r} = \frac{7083.376\times 10^3}{4074525} = 1.738$$

The percentage over surface design is:

$$OS = \frac{U_c}{U_f} = \frac{1131.5}{971.1} = 1.1525 \approx 15\%$$

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CRC Press
Taylor & Francis Group
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